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Operational Resul s

Colorado Springs, Colorado
Nov 27-30, 1979

SERI

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SOLAR HEATING AND COOLING SYSTEMS: OPERATIONAL RESULTS
~~CONFERENCE PROCEEDINGS~~

NOVEMBER 27-30, 1979

THE BROADMOOR
COLORADO SPRINGS, COLORADO

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Session I: Overview

SECOND ANNUAL
SOLAR HEATING AND COOLING SYSTEMS OPERATIONAL RESULTS CONFERENCE

INTRODUCTION

DAN S. WARD
Conference Chairman
and
Research Associate Professor
Solar Energy Applications Laboratory
Colorado State University

The Second Annual Solar Heating and Cooling Systems Operational Results Conference constitutes the second major conference on solar heating and cooling systems for buildings. Its purpose is to present results from operating solar heating and cooling systems to architects, engineers, manufacturers, contractors, and other groups interested in practical solar applications. In addressing this purpose, the conference includes information from the federally funded National Solar Heating and Cooling Demonstration Program, the National Solar Data Program, numerous state and local projects, and privately owned operating solar systems. The principal objectives of the conference are:

(1) To present actual performance data and operating experiences that demonstrate the practicality of solar heating and cooling systems,

(2) To quantify the thermal performance of solar hot water, heating, cooling, and passive installations and to discuss design and maintenance improvements, costs, and reliability,

(3) To summarize and evaluate experiences of operating systems, and

(4) To provide a forum for interaction among people involved in solar heating and cooling.

The purpose and objectives of this second annual conference are, in many respects, a continuation of the first Operational Results Conference, held in November 1978. Just as the first conference demonstrated the essential requirement of a systems engineering approach to the design and installation of solar heating and cooling systems, the second annual conference establishes the thesis that the engineering and economic performance of solar systems are more than adequate when all the engineering details are accounted for in the design and installation of the complete solar system. The theme of the second annual conference may well be: "It works, when you do it right."

The contents of this Proceedings provide the justification for the above statements. When combined with the Proceedings of the first annual conference (Conference Proceedings - Solar Heating and Cooling Systems Operational Results - SERI/TP-49-063; Solar/0500-79-00), these publications provide a wealth of information, detail, and experience in the design and installation of solar heating and cooling systems. The authors of both volumes are to be commended on the high quality and exceptional usefulness of the results of these project(s).

In assessing the contents of the papers to follow this introduction, it is important to consider the establishment of certain minimum performance evaluation criteria, and unambiguous formats for the presentation of performance results. Emphasis is therefore placed on the practical evaluation of the results, effective improvements in the operation of the system (and their effect on performance), comparisons of different operating modes, comparative analysis of the operating performance of different solar heating and cooling systems, and other operating experience which demonstrates the practicality of solar heating and cooling systems. To aid in this effort, a Performance Evaluation Criteria was developed for use by the authors of the conference papers and follows this introduction.

In addition to introducing the thesis of this conference, this introductory note also recognizes several people for their efforts which contributed significantly to the success of this conference. As Conference Co-Chairman, H. Jackson Hale of the Department of Energy conceived and initiated the planning of the conference and was instrumental in its progress throughout the planning and execution stages. The Conference Group at the Solar Energy Research Institute, under the leadership of Vicky Curry, deserves high praise for their efforts. Special thanks are due to many people who contributed their time generously and enthusiastically, including: Donna Post, Kathi McKenna, Zo Milne, and Kate Blattenbauer. The considerable efforts of the session Chairman; Jack Meeker (Northeast Solar Energy Center), Robert G. Mills (The Mills Company), J. Douglas Balcomb (Los Alamos Scientific Laboratory), and Lynn Hall (Carrier Corporation) contributed immeasurably to the conference, particularly in their attempts to summarize the essential elements of the developing solar energy industry.

To all of these people, I am indebted.

Dan S. Ward
October 1979

Session II:
Solar Domestic Hot Water Systems

PERFORMANCE EVALUATION CRITERIA

In order to adequately evaluate the operating performance of solar heating and cooling systems, it is not possible to limit our "figure of merit" to a single number. While the COP of an absorption chiller is indicative of the overall performance of a particular unit design, no such simple number is available in solar systems. Rather, it is necessary to utilize several parameters to categorize solar cooling system designs in order to account for variations in (1) individual component variations in performance within a complete solar system, (2) solar and climatological factors (i.e., the total amount, relative direct/diffuse/reflective components, and time-dependence of solar radiation as well as temperature and weather conditions affecting the heating/cooling load), (3) specific building designs and/or type of applications of solar cooling units, (4) quality of overall system design, specifications and installation, and (5) control modes, maintenance, reliability of equipment, and other associated factors.

Performance parameters which meet these and other requirements include:

(1) System overall efficiency
(2) System thermal efficiency
(3) Solar coefficient of performance
(4) Solar controlled coefficient of performance
(5) Utilizability/solar operating threshold
(6) Solar heat delivered to load

and, the "bottom line",

(7) Actual savings in non-renewable energy resources.

In order to quantify these concepts, we make the following general definitions:

η_s = Average daily solar heating and cooling system efficiency over a specified period (typically one month), dimensionless

η_{ST} = Average daily solar heating and cooling system thermal efficiency over a specified period (typically one month), dimensionless

SCOP' = Solar coefficient of performance, dimensionless

$$= \frac{\text{Total useful heating and/or cooling by solar (daily average)}}{\text{Total electrical power usage to operate the solar system (daily average)}}$$

SCOP = Solar controlled coefficient of performance, dimensionless

$$= \frac{\text{Total, controlled useful heating and/or cooling by solar (daily average)}}{\text{Total electrical power usage to operate the solar system (daily average)}}$$

$\bar{\phi}$ = Average daily utilizability over a specified period (typically one month), dimensionless

$\bar{I}_c$ = Average daily solar operating threshold, i.e., the minimum solar radiation intensity over a specified period (typically one month), at which the solar system can collect useful energy, kJ/m^2·hr

$\bar{Q}_c$ = Average daily savings in non-renewable energy resources over a specified period (typically one month) by the solar cooling system, kJ/day

$\bar{Q}_H$ = Average daily savings in non-renewable energy resources over a specified period (typically one month) by the solar heating system, kJ/day

$\bar{Q}_u^*$ = Average daily solar heat delivered to load over a specified period (typically one month), kJ/day

These parameters can be defined quantitatively with:

$$\eta_s = \frac{\bar{Q}_u^A - \bar{Q}_L^{Ext} - \beta\bar{Q}_L^{Int} - \xi\bar{E}}{A_c \bar{I}_T} (COP)_{HC} \qquad (1)$$

where:

$\bar{Q}_u^A$ = Average daily useful heat collection by the solar collector array over a specified period (typically one month), kJ/day [see equations 13a and 13b]

$\bar{Q}_L^{Ext}$ = Average daily solar system heat losses to the exterior of the conditioned space over a specified period (typically one month), kJ/day

$\bar{Q}_L^{Int}$ = Average daily solar system heat losses to the interior of the conditioned space over a specified period (typically one month), kJ/day

β = Factor to account for the degree of usefulness/non-usefulness of heat losses to the interior of the conditioned space, dimensionless
[see equations 17h and 17c]

ξ = Factor to convert the electrical energy usage to its thermal energy equivalent, dimensionless
[see equation 18]

$\bar{E}$ = Average daily electrical power required to operate the solar system over a specified period (typically one month), kJ(elec)/day

A_c = Area of the solar collector array, m^2

$\bar{I}_T$ = Average daily integrated solar radiation per unit area on the tilted surface of the solar collector array over a specified period (typically one month), kJ/day·m^2

$(COP)_{HC}$ = Coefficient of performance of the heating or cooling unit (e.g., COP of an absorption chiller), dimensionless

$$\eta_{ST} = \frac{\bar{Q}_u^A - \bar{Q}_L^{Ext} - \bar{Q}_L^{Int}\beta}{A_c \bar{I}_T}(COP)_{HC} \quad (2)$$

$$SCOP' = \frac{\bar{Q}_u^A - \bar{Q}_L^{Ext} - \bar{Q}_L^{Int}\beta}{\bar{E}}(COP)_{HC} \quad (3)$$

$$SCOP = SCOP' \text{ for the condition } \beta \geq 1 \quad (4a)$$

$$= \frac{\bar{Q}_u^A - \bar{Q}_L^{Ext} - \bar{Q}_L^{Int}}{\bar{E}}(COP)_{HC} \text{ for } \beta < 1 \quad (4b)$$

$\bar{\phi}$ is defined by Klein*, however we will utilize for the definition of Klein's $I_{c,min}$ the following:

$$\bar{I}_c = \frac{F_R U_L}{F_R(\overline{\tau\alpha})}(\bar{T}_{f,i} - \bar{T}_a + \Delta T_{control}) \quad (5)$$

where:

F_R = Solar collector heat removal factor, dimensionless

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

$(\overline{\tau\alpha})$ = Average daily value of the collector transmissivity-absorptivity product over a specific period (typically one month), dimensionless

$\bar{T}_{f,i}$ = Average daily inlet fluid temperature to the solar collector array when operating over a specified period (typically one month), °C

T_a = Average daily ambient temperature when the solar system is actively collecting solar energy over a specified period (typically one month), °C

*Klein, S.A., "Calculation of Flat-Plate Collector Utilizability", Solar Energy, Vol. 21, pp. 393-402, 1978.

$\Delta T_{control}$ = Differential set point between the solar collector outlet and thermal storage temperatures which is used to control the on/off functions of the solar collector array heat transfer medium pump and/or blower, °C

$$\bar{Q}_c = [\bar{Q}_u^A - \bar{Q}_L^{Ext} - \bar{Q}_L^{Int}\beta - \xi\bar{E}](COP)_c \quad (6a)$$

$$\bar{Q}_H = [\bar{Q}_u^A - \bar{Q}_L^{Ext} - \bar{Q}_L^{Int}\beta - \xi\bar{E}](COP)_H \quad (6b)$$

Based on these defining equations, we note several specific relationships:

$$\eta_S = \eta_{ST}\{1 - \frac{\xi(COP)_{HC}}{SCOP'}\} \quad (7)$$

$$SCOP' = \eta_{ST}\left(\frac{A_c\bar{I}_T}{\bar{E}}\right) = \frac{\bar{Q}_u^*}{\bar{E}}(COP)_c \quad (8)$$

$$\eta_{s\,(cooling)} = \frac{\bar{Q}_c}{A_c\bar{I}_T} \quad (9)$$

$$\eta_{s\,(heating)} = \frac{\bar{Q}_H}{A_c\bar{I}_T} \quad (10)$$

$$\eta_{ST\,(cooling)} = \frac{\bar{Q}_c + \xi\bar{E}(COP)_c}{A_c\bar{I}_T} \quad (11)$$

$$\eta_{ST\,(heating)} = \frac{\bar{Q}_H + \xi\bar{E}(COP)_H}{A_c\bar{I}_T} \quad (12)$$

As supporting definitions we define:

$$\bar{Q}_u^A = \begin{cases} A_c F_R(\overline{\tau\alpha})\bar{I}_T\bar{\phi}C_n & \text{(Design)} \quad (13a) \\ \dot{m}C_p(\bar{T}_{f,o} - \bar{T}_{f,i}) & \text{(Experimental)} \quad (13b) \end{cases}$$

where

C_n = Factor to account for multiple solar collector modules in series flow configurations, dimensionless

$\dot{m}$ = Average daily mass flow rate through solar collector array over a specified period (typically one month), kg/hr

C_p = Average daily specific heat of solar collector heat transfer medium over a specified period (typically one month), kJ/kg·°C

$\bar{T}_{f,o}$ = Average daily outlet fluid temperature of the solar collecting array when operating, °C

$$\bar{Q}_u^* = \bar{Q}_u^A - \bar{Q}_L^{Ext} - \bar{Q}_L^{Int}\beta \quad (14)$$

i.e.,

$$\eta_{ST} = \frac{\bar{Q}_u^*}{A_c \bar{I}_T} (COP)_{HC} \quad (15)$$

$$\bar{Q}_{HC} = (\bar{Q}_u^* - \xi \bar{E}) (COP)_{HC} \quad (16)$$

$\beta_{(\text{heating season})}$ = Fraction of heat lost to the conditioned space which is not useful (17h)

Define the building heating load, Q_{Load} as:

$$Q_{load} = (UA)_L (T_r - T_a)$$

where T_r is the thermostat room temperature setting and $(UA)_L$ is the building heat loss coefficient. The building internal heat generation, Q_I, is:

$$Q_I = Q_I(\text{solar}) + Q_I(\text{non-solar})$$

where:

$$Q_I(\text{solar}) = \bar{Q}_L^{Int}$$

when:

(1) $Q_{Load} \leq Q_I(\text{non-solar})$, $\beta = 1$

(2) $Q_{Load} \geq Q_I$, $\beta = 0$

(3) $Q_I(\text{non-solar}) < Q_{Load} < Q_I$, $0 < \beta < 1$

$$\beta_{(\text{cooling season})} = \left[1 + \frac{1}{(COP)_C}\right] \quad (17c)$$

$$\xi = \left\{ \frac{\eta_f/\eta_e}{(COP)_{CONV}} \right\} \text{ whichever is greater} \quad (18)$$

where

η_f = Average daily conventional unit efficiency of converting non-renewable energy resources into useful heating and/or cooling, dimensionless

η_e = Electrical power generation, distribution and transmission efficiency, dimensionless

$(COP)_{CONV}$ = Coefficient of performance of conventional heating and/or cooling unit, dimensionless

Finally we note that $\bar{Q}_L^{Ext}$ and $\bar{Q}_L^{Int}$ include both operating heat losses [i.e., of the form $(UA)_L(T_{op} - T_a)$] and heat capacitance losses when not operating [i.e., of the form $(cV\rho)(T_{op} - T_{non\text{-}op})$].

With these criteria, we can now evaluate the operating performance of a solar heating and/or cooling system.

NOTES

THERMAL PERFORMANCE OF DHW SOLAR ENERGY SYSTEMS IN THE NATIONAL SOLAR DATA NETWORK DURING THE 1978-1979 HEATING SEASON

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ABSTRACT

This paper summarizes the thermal performance of nine solar energy systems designed to provide heating for domestic hot water (DHW) for the period October 1978 through March 1979. The systems were selected as representative of DHW heating systems being monitored by the National Solar Data Network. Three general types of DHW heating systems are included in this presentation. These types are described as systems providing only DHW heating, systems providing DHW and space heating without a recirculation loop between the main solar storage and the DHW subsystem, and systems providing DHW and space heating with a recirculation loop. For the period reported, the average DHW solar coefficient of performance (COP) varied significantly for each system type. Further variance of COP was observed within system type as a function of single vs. multiple DHW tank design. Discussion of the significance of this variation and other thermal performance parameters is presented for each of the nine systems.

INTRODUCTION

The purpose of this paper is to present representative thermal performance data for those solar energy systems designed to provide heating for domestic hot water (DHW) which were monitored by the National Solar Data Network [1] during the period October 1978 through March 1979. Heating of domestic hot water is provided by over 90 percent of the nearly 70 systems in the network which have a means of heating water [2] and which are presently operational and producing data. The other systems provide heating for process water, laundries, and restaurants. Figure 1 illustrates the geographical distribution of the systems presently producing data.

Three general types of systems are presented in this paper. These types are described as systems providing only DHW heating, systems providing DHW and space heating without a recirculation loop between the main solar storage and the DHW subsystem, and systems providing DHW and space heating with a recirculation loop. Three systems of each type were selected to illustrate the DHW solar energy system characteristic thermal performance observed during the report period. Each of the system designs are further characterized as having



Figure 1. Hot Water Systems in the National Solar Data Network

either single or multiple DHW tanks. These DHW subsystem designs are shown in Figure 2 for typical one and two tank configurations. Further information on these nine systems can be found in References 3-11.

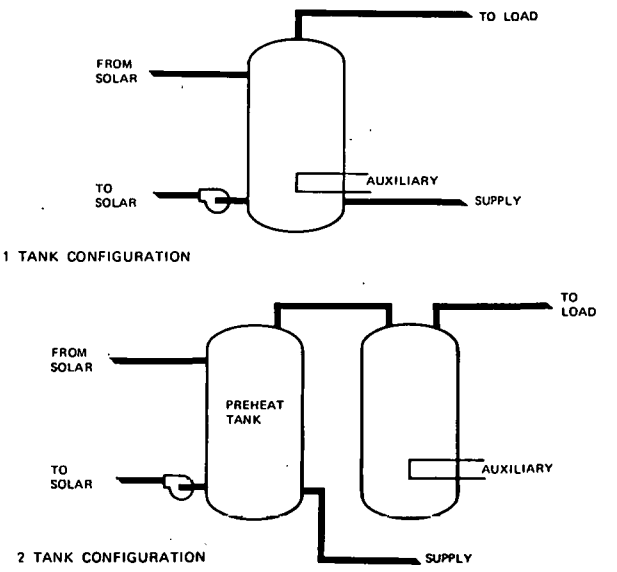


Figure 2. Typical 1- and 2-Tank DHW Configurations

SYSTEM DESCRIPTIONS

The systems selected for the first group (DHW heating only) are: A-Frame Industries; Albuquerque Western No. 1; and Colorado Sunworks. These have been numbered System 1 through 3 respectively. System 1 supports a single-family residence in Kaneohe, Hawaii. It has an array of flat-plate collectors with a gross area of 77 square feet, and uses potable water as the solar energy transport medium. Solar heated water is stored in a conventional 120 gallon DHW tank. System 2 supports a four-story, 110 unit apartment building in Albuquerque, New Mexico. It has an array of tracking collectors with a gross area of 1782 square feet, and uses water as the solar energy transport medium. Solar heated water is stored in a single 2000 gallon tank. The DHW is heated from storage via a recirculation heat exchanger and is continuously circulated throughout the building. System 3 supports a single-family residence in Longmont, California. It consists of two 30 gallon tanks which have been stripped of their insulation, painted black, and positioned inside a south wall section with a gross glazing of 44 square feet. Potable water is preheated in these tanks and transferred, on demand, to a conventional DHW tank.

The second group (space heating and DHW without recirculation) consists of: J. D. Evans, Inc., House A; Perl-Mack; and Montecito Pines. These have been numbered System 4 through 6 respectively. System 4 supports a single-family residence in Columbia, Maryland. It has an array of flat-plate collectors with a gross area of 350 square feet, and uses water as the solar energy transport medium. Solar heated water is stored in a 1000 gallon tank. The DHW is preheated via a heat exchanger in the 1000 gallon tank and transferred, on demand, to a conventional 40 gallon DHW tank. System 5 supports a single-family residence in Denver, Colorado. It has an array of flat-plate collectors with a gross area of 470 square feet and uses a glycol/water solution as the solar energy transport medium. Solar heated water is stored in an 853 gallon tank. The DHW is preheated via a 40 gallon tank immersed in the 853 gallon tank and transferred, on demand, to a conventional 40 gallon DHW tank. System 6 supports an eight apartment unit in a complex in Santa Rosa, California. It has an array of flat-plate collectors with a gross area of 950 square feet and uses water as the solar energy transport medium. Solar heated water is stored in a 2000 gallon tank. The DHW is preheated via a heat exchanger in the 2000 gallon tank and transferred, on demand, to a conventional boiler.

The final group (space heating and DHW with recirculation) consists of: Stewart-Teele-Mitchell; Ortiz and Reill Developers, Inc., House Lot 8; and Scattergood School. These have been numbered System 7 through 9 respectively. System 7 supports a single-family residence in Malta, New York. It has an array of flat-plate collectors with a gross area of 432 square feet, and uses a glycol/water solution as the solar energy transport medium. Solar heated water is stored in a 1000 gallon tank. Energy is provided from storage to a heat exchanger in the 75 gallon DHW preheat tank via a recirculation loop. Potable water is preheated and transferred, on demand, to a conventional 40 gallon DHW tank. System 8 supports a single-family residence in Escondido, California. It has an array of flat-plate collectors with a gross area of 192 square feet, and uses water as the solar energy transport medium. Solar heated water is stored in a 750 gallon tank. Energy is provided from storage to a heat exchanger in the conventional 66 gallon DHW tank via a recirculation loop. System 9 supports a gymnasium locker room at a school in West Branch, Iowa. It has an array of flat-plate collectors with a gross area of 2496 square feet, and uses air as the solar energy transport medium. Although a 64 ton rockbed provides solar energy storage for space heating, energy is provided via a heat exchanger in the collector to storage loop by recirculation of potable water from the 120 gallon DHW preheat tank. Preheated DHW is transferred, on demand, to the conventional 52 gallon DHW tank.

RESULTS

The average monthly performance of each of the nine DHW systems for the period October 1978 through March 1979 is presented in Table I. As shown by the factor TOTAL LOAD, monthly average DHW loads for the report period range from 0.44 million Btu for the school gymnasium (System 9) to over 41 million Btu for the large apartment building (System 2). The average monthly DHW load for the single-family residences was 1.08 million Btu (Systems 1, 3-5, 7, 8).

TABLE I: AVERAGE MONTHLY PERFORMANCE ($\times 10^{-6}$ BTU)

SYSTEM	1	2	3	4	5	6	7	8	9
SOL IN	0.70	13.89	0.62	0.15	1.27	3.26	0.20	0.68	1.36
AUX IN	0.17	62.85	2.03	0.46	1.68	4.81	0.50	0.26	0.96
TOTAL LOAD	0.65	41.41	1.94	0.46	2.08	6.85	0.46	0.89	0.44
% SOL	91%	24%	23%	29%	40%	48%	12%	73%	38%
LOAD-SOL	0.59	10.14	0.44	0.14	0.83	3.26	0.05	0.65	0.17
LOAD-AUX	0.06	31.27	1.50	0.32	1.25	3.60	0.41	0.24	0.27
LOSS-SOL	0.11	3.75	0.08	0.01	0.44	0.0	0.15	0.03	1.19
LOSS-AUX	0.11	31.58	0.53	0.14	0.43	1.21	0.09	0.02	0.68
S/S OP-SOL	-	0.75	-	-	-	-	0.042	0.26	0.20
S/S OP-AUX	-	0.97	-	-	-	0.68	-	-	-
COL OP-SOL	0.06	1.30	-	0.011	0.13	0.22	0.017	0.053	0.03
TOT OP-SOL	0.06	2.05	-	0.011	0.13	0.22	0.059	0.313	0.23

For the single-family residence systems, solar provided an average of 42 percent of the DHW load over the report period. This latter performance was the result of solar providing 0.59 million of the 1.44 million Btu consumed (41 percent) by the residential DHW subsystems at a solar operating expense of 0.10 million Btu for an average month during the same report period.

Review of the data presented in Table I (while not-

ing that the DHW load is defined as that energy required to heat the hot water consumed from the cold water inlet temperature to the DHW delivery temperature) provides insight into the operation of these nine DHW systems. Such insight is provided for each system by distinguishing between the delivered DHW load, the amount of energy consumed by the DHW subsystem, and the percentage of each which is solar (vs. percentage auxiliary). This distinction illustrates the impact of matching a given solar DHW system design, usually based on generalized assumptions of DHW load magnitude and frequency, to actual DHW load profile conditions. Of additional significance is the amount of energy consumed by the DHW solar energy system in providing the benefits observed in the above review. The latter is given in Table I by the factor TOT OP-SOL.

Solar provided 80 percent of the energy supplied to the DHW subsystem of System 1 while accounting for 91 percent of the DHW load. This slight increase in solar percentages is characteristic of most systems and is the consequence of lower DHW subsystem solar losses (16 percent of solar supplied to subsystem was lost) than auxiliary energy losses (65 percent lost). The overall losses from the subsystem were 25 percent of the total energy supplied to the subsystem. This percentage is considered average and can be used as a base of comparison for the other systems.

System 2 is observed to have increased the solar fraction of the DHW load to 24 percent from the 18 percent of the energy supplied to the subsystem. Solar losses from the subsystem were 27 percent of solar energy supplied while auxiliary losses were 50 percent of auxiliary energy supplied. The high overall loss (46 percent) is attributed primarily to the continuous circulation DHW loop in the apartment building.

System 3 had an increase of solar fraction of DHW load to 23 percent from the 20 percent of energy supplied to the subsystem. Solar losses were 15 percent and auxiliary losses were 26 percent for an overall loss of 24 percent from the subsystem. This loss percentage is better than average for multi-tank DHW subsystems and is a benefit of the preheat tank temperature. This temperature is lower for System 3 than most multi-tank designs (which have the preheat tank directly connected to a collector array) and thus losses from the tank are reduced.

System 4 also had a increase in solar fraction for the DHW load (29 percent) from the 25 percent of energy supplied. Solar losses were 7 percent and auxiliary 30 percent of supplied energy respectively and the overall losses were 25 percent. System 5 had a slight decrease in solar fraction from 43 percent of the energy supplied to 40 percent of the DHW load. Solar losses were 35 percent and auxiliary 26 percent of supplied energy respectively and the overall losses were 29 percent. System 6 had a solar fraction of DHW load of 48 percent which was greater than the 40 percent supplied to the subsystem. There were no solar losses and auxiliary losses were 25 percent of auxiliary energy supplied. Overall losses were 15 percent for the subsystem. These last three systems illustrate the result of providing solar energy to the subsystems only on demand. The impact is that standby losses are borne primarily by the auxiliary heating source. A greater portion of the standby loss is shifted to solar as the DHW consumption is increased. Although not shown directly in the data of Table I, System 5 provided proportionally more DHW than was provided by either of the other two systems of this type. The lack of solar loss in System 6 is the result of the system design which allows solar to only be applied directly to the DHW load.

In the final group of systems, Systems 7 and 9 experienced reductions in solar fraction of DHW load as compared to supplied energy (29 percent to 12 percent and 59 percent to 38 percent, respectively). These reductions were accompanied by the largest solar losses observed in this review. These were 75 percent of the solar supplied to the subsystem were lost for System 7 and 88 percent lost for System 9. This high solar loss is common to these two systems even though the overall losses of System 7 were 34 percent while System 9 had 81 percent overall losses. The solar fraction of DHW load for System 8 was essentially the same as the fraction of energy supplied to its DHW subsystem. The system had very low losses of 4 percent of the solar supplied and 8 percent of the auxiliary for an overall loss of 5 percent. The most significant design difference between System 8 and Systems 7 and 9 is that System 8 is a single tank DHW subsystem while the other two are multi-tank.

However valuable thermal data may be to compare system design, overall system evaluation must include the "cost" of that thermal performance. This evaluation is provided by the solar coefficient of performance (SCOP) for the DHW systems as presented in Table II. In the context of this report the DHW solar coefficient of performance SCOP is the ratio of the total amount of the DHW load provided by solar to the total amount of electrical power required to provide that solar contribution. The notation SCOP = SCOP' indicates that the amount of solar provided in a controlled manner and the total amount of solar provided are equal. Also shown in Table II are the average amounts of daily solar energy delivered to the DHW load ($\bar{Q}_u^*$), the solar DHW system thermal efficiency (η_{st}), the average daily savings in non-renewable energy resources attributed to the solar DHW system ($\bar{Q}_{DHW}$), and the solar DHW system overall efficiency (η_s).

The solar efficiency terms η_{st} and η_s are the ratios of the solar portion of the DHW load and solar portion of the DHW load minus solar operating energy respectively to the total incident solar radiation. The latter two terms were prorated to the DHW subsystem for Systems 4 through 9.

TABLE II: MONTHLY DHW SYSTEM PARAMETERS

SYSTEM 1 (A-FRAME INDUSTRIES)

MO/YR	$\bar{Q}_u^*$ ($\times 10^{-6}$ Btu/day)	η_{st}	SCOP =SCOP'	$\bar{Q}$ DHW ($\times 10^{-6}$ Btu/day)	η_s
10/78	0.019	21%	10.4	0.022	19%
11/78	0.019	23%	10.0	0.022	21%
12/78	*	*	*	*	*
1/79	0.019	27%	10.0	0.017	25%
2/79	0.017	23%	9.0	0.017	20%
3/79	0.024	19%	10.6	0.027	17%

SYSTEM 2 (ALBUQUERQUE WESTERN, NO. 1)

MO/YR	$\bar{Q}_u^*$ ($\times 10^{-6}$ Btu/day)	η_{st}	SCOP =SCOP'	$\bar{Q}$ DHW ($\times 10^{-6}$ Btu/day)	η_s
10/78	0.52	19%	5.3	0.66	15%
11/78	0.29	15%	4.1	0.27	11%
12/78	0.22	11%	4.6	0.24	8%
1/79	0.20	10%	4.0	0.22	8%
2/79	0.43	17%	5.9	0.54	14%
3/79	0.37	14%	5.5	0.43	11%

SYSTEM 3 (COLORADO SUNWORKS)

MO/YR	$\bar{Q}_u^*$ ($\times 10^{-6}$ Btu/day)	η_{st}	SCOP =SCOP'	$\bar{Q}$ DHW ($\times 10^{-6}$ Btu/day)	η_s
10/78	*	*	*	*	*
11/78	0.013	21%	∞	0.013	21%
12/78	0.018	22%	∞	0.018	22%
1/79	0.018	22%	∞	0.018	22%
2/79	0.021	28%	∞	0.021	28%
3/79	0.016	27%	∞	0.016	27%

SYSTEM 4 (J. D. EVANS, HOUSE A)

MO/YR	$\bar{Q}_u^*$ ($\times 10^{-6}$ Btu/day)	η_{st}	SCOP =SCOP'	$\bar{Q}$ DHW ($\times 10^{-6}$ Btu/day)	η_s
10/78	*	*	*	*	*
11/78	0.003	10%	9.3	0.002	9%
12/78	0.004	31%	19.1	0.004	29%
1/79	*	*	*	*	*
2/79	0.005	12%	10.0	0.004	11%
3/79	0.006	15%	11.8	0.006	14%

SYSTEM 5 (PERL MACK)

MO/YR	$\bar{Q}_u^*$ ($\times 10^{-6}$ Btu/day)	η_{st}	SCOP =SCOP'	$\bar{Q}$ DHW ($\times 10^{-6}$ Btu/day)	η_s
10/78	0.035	6%	4.7	0.051	5%
11/78	0.024	10%	3.7	0.033	7%
12/78	0.018	14%	9.8	0.029	13%
1/79	0.026	12%	9.0	0.041	11%
2/79	0.031	15%	10.0	0.045	14%
3/79	0.030	11%	9.0	0.046	10%

SYSTEM 6 (MONTECITO PINES)

MO/YR	$\bar{Q}_u^*$ ($\times 10^{-6}$ Btu/day)	η_{st}	SCOP =SCOP'	$\bar{Q}$ DHW ($\times 10^{-6}$ Btu/day)	η_s
10/78	*	*	*	*	*
11/78	0.055	20%	12.8	0.051	19%
12/78	0.083	19%	11.6	0.076	17%
1/79	0.107	23%	14.7	0.100	21%
2/79	0.143	25%	16.2	0.134	23%
3/79	0.152	24%	16.6	0.143	23%

SYSTEM 7 (STEWART-TEELE-MITCHELL)

MO/YR	$\bar{Q}_u^*$ ($\times 10^{-6}$ Btu/day)	η_{st}	SCOP =SCOP'	$\bar{Q}$ DHW ($\times 10^{-6}$ Btu/day)	η_s
10/78	*	*	*	*	*
11/78	0	-	-	0	-
12/78	0	-	-	0	-
1/79	0.002	1.4%	0.49	-0.001	-1.4%
2/79	0.003	1.9%	0.80	-0.004	-0.5%
3/79	0.005	3.9%	1.43	0.001	1.2%

SYSTEM 8 (ORTIZ & REILL DEVELOPERS, INC., HOUSE LOT 8)

MO/YR	$\bar{Q}_u^*$ ($\times 10^{-6}$ Btu/day)	η_{st}	SCOP =SCOP'	$\bar{Q}$ DHW ($\times 10^{-6}$ Btu/day)	η_s
10/78	0.023	7%	1.2	0.008	1%
11/78	0.025	19%	2.6	0.016	11%
12/78	0.016	7%	2.3	0.008	4%
1/79	0.015	13%	1.9	0.005	6%
2/79	0.024	12%	3.7	0.018	9%
3/79	0.026	11%	2.3	0.015	6%

*Unavailable data

SYSTEM 9 (SCATTERGOOD SCHOOL)

MO/YR	$\bar{Q}_u$ ($\times 10^{-6}$ Btu/day)	η_{st}	SCOP =SCOP'	$\bar{Q}$ DHW ($\times 10^{-6}$ Btu/day)	η_s
10/78	0.009	5.4%	0.80	0.029	-1.4%
11/78	0.008	3.3%	1.18	0.014	0.5%
12/78	0.004	3.2%	0.77	0.009	-0.9%
1/79	0.001	0.7%	0.19	0.005	-3.0%
2/79	0.006	2.3%	0.56	0.015	-1.8%
3/79	0.005	2.4%	0.63	0.007	-1.4%

SUMMARY AND CONCLUSIONS

Based on the data presented, the following observations can be made:

1. As a group, the DHW and space heating systems without recirculation loops had higher values of solar COP than the other system types. The DHW heating only systems were second. Two of the DHW and space heating systems with recirculation loops (Systems 7 and 9) had solar COP values less than unity. This low level of performance was also reflected in their negligible energy savings and negative overall system efficiency.
2. Solar thermal losses are higher for multi-tank DHW systems than for single DHW tank systems. This was the case for the combination system with recirculation between the main solar storage tank and the DHW subsystem (System 7) and the combination system with recirculation between the collector to storage loop and the DHW subsystem (System 9).
3. The combination of high thermal losses and increased operating energy associated with multi-tank DHW systems with recirculation loops raises serious doubt as to their economic viability.
4. The passive DHW system (System 3) demonstrated that reasonable system efficiency can be obtained without requiring operating energy.
5. Although the data reported would seem to indicate that combination DHW and space heating systems without recirculation are clearly superior performers when compared to other system types, some doubt remains as to the context of that observation. Other studies have reported preliminary indications of degraded system performance for combination systems during non-heating season operation. The relatively large operating energy requirement for a DHW only heating load coupled with the cooling penalty caused by the heat loss within the structure could revise the order of design desirability. The tendency to higher solar COP values (as shown in Table II) for mid-winter operation tend to support this caution.
6. The overall result of this study is the reinforcement of the solar energy design adages - "MATCH YOUR SYSTEM TO THE REQUIREMENTS" and "LESS IS MORE." The impact of actual DHW load magnitudes and profiles on solar DHW system performance vs. the expected performance based on assumed or "average" loads can be significant. Simple, straight forward system designs generally prove to be the best engineering solution. The performance of Systems 1, 3, 4, and 6 show these desirable characteristics.

ACKNOWLEDGEMENT

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THE PERFORMANCE OF A DAYSTAR HW2/F-B SOLAR WATER HEATER IN SOUTHERN NEW JERSEY

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ABSTRACT

A Daystar HW2/F-B solar water heater installed at the author's home (39.5°N latitude) in October, 1977 has been continuously monitored by microcomputer controlled instrumentation since January 1, 1979. During the first 7 months (Jan.-July 1979) average daily measurements were as follows: Useful heat - 13,800 kJ/day (3.83 kW-hr/day); Insolation - 13,900 kJ/M^2/day (3.86 kW-hr/M^2/day); Solar system electrical usage - 2328 kJ/day (0.65 kW-hr/day); Hot Water Usage - 173 ℓ/day (45.6 gal/day) and backup electrical heat 10,100 kJ/day (2.82 kW-hr/day). The system consists of 3.92M^2 (42.2 ft^2) net area of Daystar type 21-B collectors facing due south and tilted 45° from the horizontal, together with a single 80 gallon tank for storage. The microcomputer monitors 6 temperatures, the integrated insolation and on/off time for both the collector loop circulator and the backup electric heat once each minute. It then calculates 10-minute averages, and one hour average water usage figures are taken manually each day.

INTRODUCTION

Since January 1, 1979 a Daystar type HW2/F-B solar domestic water heater (DHW) in the author's home has been automatically monitored by a microcomputer controlled set of instruments. This paper presents the preliminary results from the first seven months of minitoring, the period from January through July, 1979. The equipment is described first, then figures detailing the SDHW system's performance are presented. The SDHW system is not unusual and hence these results should be readily reproducible by others.

Perhaps the most unique aspect of this project is the low cost microcomputer controlled instrumentation system. Although this system cost almost $3,000, currently available equipment should allow the installation of a system with similar capabilities for half this amount.

There are several advantages of a microcomputer based system such as this, besides its low cost. It is quite versatile and programmable; thus, besides automatically recording the data, it can do some analysis of the data. It is possible to write programs to display directly, in real time relevant performance parameters and, of course such programs can be easily changed as other needs become evident. Finally, having the data available in computer readable form facilitates more extensive analysis on other machines.

EQUIPMENT

This section describes both the solar water heater and the microcomputer controlled instrumentation system.

The HW2/F-B DHW System

The HW2/F-B domestic solar water heater system (see Fig. 1) consists of two Daystar Type 21-B collectors, a 301 liter (80 gallon) tank with a heat exchanger in the bottom, and a differential thermostat controling the circulator pump. The collectors have a net collecting aperture of 3.92M^2 (42.2ft^2) and: $F_r(\tau\alpha) = 0.743$; $F_r(U) = 0.770$ [1]. The collectors have the accordian folded polymer "Heat Trap" second cover and include a "Temperature Limiter" to protect the collector under stagnation conditions.

The storage tank has a 4 KW resistance heating element in the top section with its thermostat set to 43°C (110°F). The heat exchanger in the bottom is reported to have a surface area of 1.39M^2 (15ft^2) [1].

The differential thermostat is manufactured by Rho Sigma and the circulator pump is a TACO model 007 rated at 1/12 HP (0.75 amps at 110 Vac).

The collectors are mounted on a one-inch pipe rack facing due south at an angle of 45° to the horizontal. The location is about 20 miles northwest of Atlantic City, New Jersey at 39.5°N and approximately 25 M (81 feet) above sea level. There is partial shading of the collectors before 0900 solar time all year, but no shadows during the rest of the day. The collectors are located about 7.5 meters (25 feet) from the storage tank and fed with 1.90cm (3/4 inch) copper tubing insulated with 7.6cm (3 inch) diameter rigid foam. Design flow rate for the glycerine-water heat

transfer fluid in 0.63 ℓ/sec (1.0 gal/min), though no measurements of this have made.

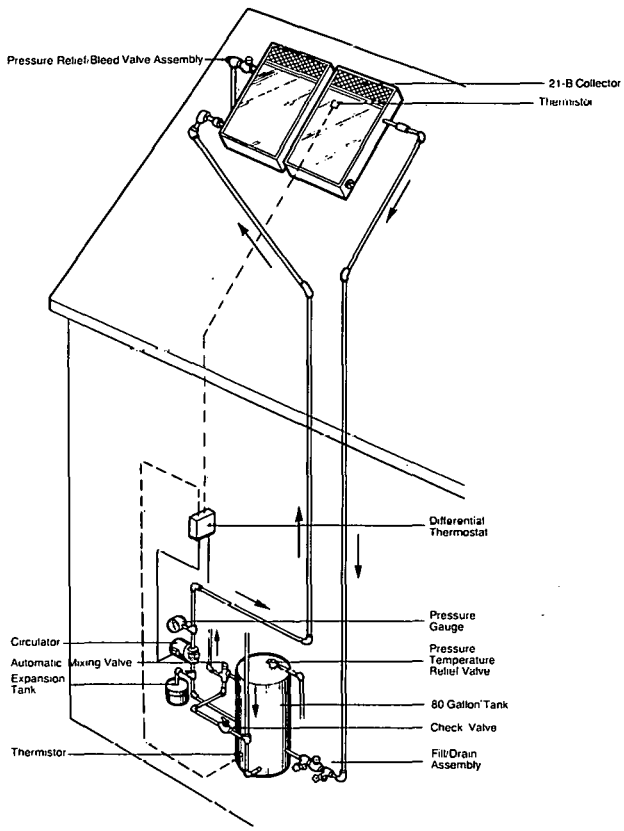


Fig. 1. Diagram of Daystar Solar DHW System. (Adapted from Daystar HW2/F-B Data Sheet by permission.)

Instrumentation

The instrumentation for the system consists of a black and white pyranometer, six separate temperature sensors, on/off indicators for the solar circulator and the backup electric heat. All of these sensors are automatically recorded by the microcomputer. In addition, a Corad water meter measures the total hot water used.

An Eppley [2] black and white pyranometer (Model 8-38), together with a model 411 electronic integrator is used to measure the total insolation on the collector surface. The integrator is read each minute by the computer and checked manually each evening. The pyranometer together with the integrator was calibrated by the Eppley Laboratory in December, 1978, and since Stockton College has no secondary standard available, this calibration is assumed to hold throughout the 7-month period reported here.

The temperature sensors are active solid state devices (Analog Devices type AD 590-K) [3]. These have been individually calibrated against a liquid in glass thermometer. The calibration is checked daily against a thermocouple thermometer and periodically against the liquid in glass one also. Accuracy of the AD590K devices is rated at $\pm$0.5°C (about $\pm$ 1°F) over the entire -55 to +150°C when these individual calibrations are made. Precision of $\pm$0.1°C or better is attained over limited temperature ranges.

Six of these temperature sensors are in use measuring 1) ambient air temperature behind the collectors; 2) cold water supply temperature on the pipe into the hot water tank initially - later moved to the cold water tank near its outlet; 3) collector outlet temperature - measured at the outlet pipe; 4) heat exchanger inlet temperature - measured on the pipe at the base of the hot water tank; 5) hot water tank bottom temperature - measured on the outside of the tank about 60° around from the cold water inlet pipe; and 6) hot water tank top temperature - measured on the outside of the tank - near the center of the top. Some experimentation with sensor placement was necessary to insure meaningful temperature readings.

The on/off indicators for the solar circulator and the backup electric heating element are simple rectifer-zener diode regulator circuits run from the 24 Vac manual readout running time meters. This circuit provides a TTL compatible +5V "on" signal and a 0V "off" signal for the microcomputer and two LED indicators.

The insolation, six temperatures, and the on/off times for the solar circulator and backup electric heating element are monitored continuously by a Motorola 6800 based microcomputer [4,5]. Minimum requirements for the computer are 4K of random access memory - four 8-line parallel input/output (I/O) ports, a printer, a timer, and a tape interface. In addition, the system in use, (manufactured by Southwest Technical Products Corp.) has a dual mini-floppy disk drive and a scientific calculator interface. The entire instrumentation package cost about $3000, with $1000 of that going for the optional floppy disk system. With currently available computing equipment a package with comparable capabilities could be set up with about $1000 worth of equipment.

The computer uses the timer to continuously drive a 24-hour clock, 4-year calendar program. Each minute as the clock is updated all the sensors are read and the results temporarily saved. Every 10 minutes averages are computed, saved in a table which holds the data for an entire day, and also listed on the line printer. Each hour - hourly averages are computed, stored in another daily table and written on the printer. At midnight each day the calendar is updated and both the 10-minute averages and the hourly averages are saved on magnetic disks where the data can be re-read later for additional processing, or plotting, etc.

A 12-bit A/D converter (AD572) [3] with 16-channel multiplexer (CD4067) is used to read the temperature sensors. This requires 12 input lines and 4 output lines or one sixteen line parallel I/O interface. The insolation integrator is read, two digits at a time, through one 8 bit parallel I/O port. The on/off sensors take an additional 2 I/O lines - the printer, calculator interface, timer, and a serial interface for the tape and terminal each require one of the I/O ports, while the disk requires 2 I/O ports. Thus, 7 of the 8 I/O ports on the SWTPC 6800 motherboard are utilized in the system as it is presently operating. Additional details on the instrumentation will be presented in a subsequent article.

The hot water usage is measured by a manual readout Corad [6] type CM-2 water meter. Accuracy of this meter is quoted in the manufacturers literature [3] as $\pm$ 2% from 1/4 GPM to 20 GPM, and is within these limits in actual checks.

To check the operation of the temperature sensors a 6-channel thermocouple thermometer (AD 2036 [3]) was also installed. These values are read manually each day. As yet no careful study of the accuracy of either the solid state sensors or of the thermocouple device has been carried out. When small differences are noticed, however, the calibration of the solid state sensors is rechecked against the liquid in glass thermometer.

RESULTS

The Measurements

Nearly continuous data have been collected beginning January 1, 1979. In this paper data are presented for the 212-day period from January 1 through July 31, 1979. During this 212-day period there are 200 days when complete data are available only 5 days from which more than 3 hours of data samples are missing during sunlight hours, and an additional 7 days with gaps of from 10 minutes to 3 hours during the day. Data for 137 of the 144 days from March 11 to July 31 are available in computer readable form on mini-floppy disks, while data taken before March 11 and on the 7 days when it was not recorded are available only in hard copy, and must be retyped for further computer processing.

Of the 10 quantities regularly measured, 6 are used in this analysis. These are: 1) the insolation, 2) cold water supply temperature, 3) tank top temperature, 4) backup electric heater "on" time, 5) solar circulator "on" time, and 6) the volume of hot water used. All of these quantities except the hot water volume are recorded automatically by the computer.

The procedure used is to calculate the energy required for the hot water and to subtract the amount of energy supplied by the backup electric heater. This method has the advantage of relying on quantities which are known quite accurately. It would, of course, be preferable to have direct measurements of the solar input via the heat exchanger and of the heat losses from the hot water tank. (Future plans call for the addition of a flowmeter in the collector loop so the solar contribution can be measured directly. The heat loss from the tank ranges from 10% to 25% of the total heat required and hence errors in this quantity contribute proportionally less to the accuracy of the overall results.)

As shown at the end of the analysis section, the uncertainty in the overall energy usage is no more than 1500 kJ/day (0.4 kW-hr/day). This quantity should be kept in mind when interpreting the figures presented in the next section.

Performance Figures

Table 1 (at the end of this paper) summarizes system performance over the seven month period. The solar contribution ranged from 7320 kJ/day (2.03 kW-hr/day) in January, 26% of total energy usage to 17,900 kJ/day (4.97 kW-hr/day) in July or 86% of total energy usage. In July the electrical backup heating was turned off so the 14% non-solar contribution was all the electrical energy required to run the solar circulator and the differential thermostat. Insolation ranged from 9480 kJ/M^2/day (2.63 kW-hr/M^2/day or 833 BTU/ft^2/day) in January to a high of 16,600 kJ/M^2/day (4.62 kW-hr/M^2/day or 1460 BTU/ft^2/day) in March. The average insolation for the entire 7 month period was 13,900 kJ/M^2/day (3.83 kW/M^2/day or 1210 BTU/ft^2/day).

Hot water usage ranged from 132 ℓ/day (34.9 gal/day) in July to 211 ℓ/day (55.8 gal/day) in April and averaged 173 ℓ/day (45.6 gal/day) over 7 months. The hot water supply temperature was 43°C (110°F) equal to the backup thermostat setting except for July when the water usage was low and collector efficiency high and the average temperature was 46°C (114°F). As noted above, the electrical backup heat was turned off for the entire month. Figures for August (the backup was still off) were 52°C (125°F) when water usage was down even more to less than 73.6 ℓ/day (19.5 gal/day) because of a 2 week vacation.

Financial savings ranged from $18.99/yr. in January to $86.40/yr. in July. The estimated actual savings for 1979 are about $50.00 and may be higher if an additional rate increase goes into effect on October 1, as requested. Savings in 1980 and subsequent years are expected to be substantially greater because the proposed new rates do not give any reduction in the cost per kW-hr for large usage.

The total system installed cost was approximately $2000, and although the system does not qualify for the HUD $400.00 grant, the tax credit of $600 makes sure that the system will pay for itself in less than 20 years.

ANALYSIS

This section describes the techniques used to compute the quantities presented in Table 1, together with the possible errors in these values.

1. Total Insolation: I_T These figures are obtained by summing the daily insolation recorded by the Eppley integrator I_D from the black and white pyranometer.

$$I_T = \Sigma\, I_D / N \qquad (1)$$

N is the number of days for which measurements were available. There were 5 days missing in January and 1 in July. Otherwise, all days' data were used. There were 7 days for which partial data, together with chart records from the pyranometer at Stockton College's Solar Energy House were used to estimate the total insolation. One day each in February, March and June, and two days were estimated in May and July. January and April had no days estimated. As indicated in the equipment section, the pyranometer was calibrated by the Eppley Labs in December, 1978. Occasional checks against Stockton's other two pryanometers and against ASHRAE's clear day values for clear days give no indication that performance was altered over the 7-month period. From the manufacturer's specifications it is estimated that the results are accurate to $\pm 3\%$ from the pryanometer itself and with a similar allowance for the integrater it is expected that the overall accuracy is better than $\pm 5\%$.

2-8. Solar Energy Delivered to Hot Water Tank; Q_U^*. This is computed by calculating the total energy required to heat the hot water used, Q_{HW}. The heat losses from the tank Q and the electrical backup heat used, E, and using the energy balance equation,

$$Q_U^* = Q_{HW} + Q_L^{tank} - E \qquad (2)$$

$$Q_{HW} = \rho\, c\, V\, \Delta T \qquad (3)$$

where ρ = density of water (1 kg/ℓ)

c = specific heat of water (4.19 kJ/kg/°C)

V = volume of water used (liters)

ΔT = hot water outlet temp. - cold water supply temp. (° C)

Q_L^{tank} = loss from the tank. This has been calculated and also measured by observing electrical energy usage when the solar system is shut down and no hot water is used. It is found to be 3600 kJ/day (1 kW-hr/day).

E = Electrical backup energy used - measured by

$$E = P\, \Delta t. \qquad (4)$$

Where P is the Power rating of the resistance heating element and Δt is the backup heat "on" time measured by both the computer and a running time meter. Checks of the line voltage at the element while the backup heat is on show voltages of 225V $\pm$ 8V.

9. Solar system daily efficiency - ηs

$$\eta s = Q_U^* / (I_T \cdot Ac) \qquad (5)$$

Ac is the collector area, 3.92 M^2 (42ft^2). The other quantities were previously defined.

10. Electrical energy used by the solar system; E_S

$$Es = P_C\, \Delta T_C + E_D \qquad (6)$$

Where P_C is the power rating of the circulating pump (80 Watts), Δt_C is the circulating pump "on" time and E_D is the differential thermostat power consumption - 260 kJ/day (.072 kW-hr/day).

11. Solar System COP;

$$COP = Q_U^* / E_S \qquad (7)$$

12. Non-renewable energy saved, Q_H

$$Q_H = Q_U^* - E_S \qquad (8)$$

13. Total Electrical usage, E

$$E_T = E_S + E_D \qquad (9)$$

14. Solar Fraction = f

$$F = 1 - (Q_{HW} + Q_L^{tank}) / E_T \qquad (10)$$

15. Financial Savings, S.
From January through May utility rates for the last kW-hr used was 3.19¢/kW-hr - for June the rate was 5.31¢/kW-hr and in July the rate went up to 5.7¢/kW-hr.

$$S = Q_H\, R \qquad (11)$$

where R is the current rate.

It is useful to close this section with a brief analysis of the overall errors in this method. The RMS error in the total energy usage comes from the following measurement of the hot water.

Hot Water Volume,	V	$\pm$ 2%
Temperature rise,	ΔT	$\pm$ 2%
Heat Losses from tank	Q_L^{tank}	$\pm$ 10% of Q_L
or 2% of Q_U^*		
Electrical Energy	E_T	$\pm$1%

The RMS error is thus approximately $\pm 5\%$ of Q_U^* or E which become approximately $\pm$1500 kJ or $\pm$0.4 kW-hr.

An experimental way to determine the error by this method is to use the energy balance relationship (Eq. 2) for times when $Q_U^* = 0$. (Times when the

solar circulator remained off.) Three such times were analyzed and the greatest error found was 0.31 kW-hr. out of 48,400 kJ for an error of only 2.3%, so perhaps the above theoretical estimate are larger than the actual errors. A proper statistical analysis of each period of more than 24 hr. for which $Q_u^* = 0$ will be made when computer analysis of the recorded data is carried out.

ACKNOWLEDGEMENT

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TABLE 1
HW2/F-B Preliminary Performance Data for 1979

			Jan	Feb	Mar	Apr	May	June	July
1.	Total Insolation I	(kJ/M^2/day)	9,480	11,800	16,600	13,700	14,900	15,300	15,300
		($kW\text{-}hr/M^2$/day)	2.63	3.27	4.62	3.81	4.14	4.24	4.25
2.	Cold Water Supply Temp	°C	11.1	10.5	12.2	13.3	13.3	13.9	14.4
		°F	52	51	54	56	56	57	58
3.	Hot Water Temp	°C	43.	43.	43.	43.	43.	43.	45.6
		°F	110	110	110	110	110	110	114
4.	ΔT (3-2)	°C	32.2	32.8	31.1	30.0	30.0	29.4	31.1
		°F	58	59	56	54	54	53	56
5.	Total Hot Water Utilized	(ℓ/day)	181	188	180	211	173	152	132
		(gal/day)	47.8	49.6	47.5	55.8	45.6	40.2	34.9
6.	Total Heat Required	(kJ/day)	28,100	29,400	29,900	31,000	25,300	22,400	20,800
		(kW-hr/day)	7.80	8.18	7.48	8.60	7.03	6.23	5.77
7.	Backup Electrical Energy Used E_m	(kW-hr/day)	19,440	19,410	9,170	13,920	6,270	4,060	0
			5.40	5.39	2.55	3.87	1.74	1.13	0
8.	Energy Delivered to Hot Water Tank Q_u^*	(kJ/day)	7,320	8,610	15,600	14,900	16,400	15,400	17,900
		(kW-hr/day)	2.03	2.39	4.32	4.13	4.55	4.29	4.97
9.	Collector Daily Efficiency, η_s ($A_C = 3.92M^2$)	(8/1A_C)	20%	19%	24%	28%	28%	26%	30%
10.	Solar System Electrical Usage, E_s	(kJ/day)	1,440	1,610	2,250	2,290	2,780	2,910	2,970
		(kW-hr/day)	.400	.446	.626	.637	.771	.810	.825
11.	Solar System COP	(8/10)	5.1	5.4	6.9	6.5	5.9	5.3	6.0
12.	Non-Renewable Energy Saved Q_H (8-10)	(kJ/day)	5,800	7,000	13,400	12,600	14,600	12,500	14,900
		(kW-hr/day)	1.63	1.95	3.71	3.50	4.06	3.47	4.15
13.	Total Electrical Usage (E_T)	(kJ/day)	20,900	21,000	11,400	16,200	9,050	6,970	2,970
		(kW-hr/day)	5.80	5.84	3.17	4.50	2.51	1.94	0.82
14.	Solar Fraction, f	(1-9/8)	.26	.29	.62	.48	.64	.69	.86
15.	Financial Savings, S	(¢/day)	5.2	6.2	11.8	11.1	13.0	18.4	23.7
		($/year)	18.99	22.72	43.23	40.78	47.30	67.30	86.40

NOTES

THERMAL PERFORMANCE OF A SOLAR HEATED HOT WATER SYSTEM

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ABSTRACT

The thermal performance analysis of the 54.3 m^2 (585 $ft.^2$) flatplate solar collector hot water heating system at the Bartlesville Energy Technology Center found the system had an extremely low summertime efficiency due to light hot water heating requirements. The collector efficiency increased from 25% in July to 41% in October due to the greater hot water heating requirements. The increase in required heat was caused by a modification to route the domestic hot water return through the solar storage tank as well as a demand increase due to the change in seasons from summer to fall. The average daily savings in nonrenewable energy resources also increased from 2500 kJ/day (2400 Btu/day) to 74,000 kJ/day (71,000 Btu/day) during the same period of time. Instrumentation used in the analysis included several thermometers, two American Meter Btu meters, and an Eppley 8-48 pyranometer.

INTRODUCTION

When a businessman or homeowner decides to go "solar" with a limited investment, he inevitably chooses a solar system to heat domestic hot water.

As a demonstration, the Bartlesville Energy Technology Center (BETC) designed and installed a solar system to heat the domestic hot water for an existing BETC laboratory-office building.[1] The system has been in continuous operation with little downtime since start up eighteen months ago.

The nominal demand was estimated at 6400 liters (1700 gallons) of water per day at 50°C (120°F). The solar heating system was designed to supply approximately 60 percent of this heating load with 55 square meters of collectors. The collectors are flat plate, single glaze, with nonselective surface, inclined at 45 degrees.

Figure 1 (see Appendix) schematically describes the flow system. It includes a loop that circulates fluid through the collector array, and passes through a heat exchanger and is common to a secondary loop. The secondary loop ciculates the solar storage fluid from the 4550 liters (1,200-gallon) storage tank through the heat exchanger and then returns to the storage tank. The supply water is heated as it passes through the solar storage tank.

The two pumps that circulate the fluid are cycled automatically by temperature sensitive switches. When the collector temperature reaches 8°C (15°F) above the solar storage temperature, the pumps are turned on. When the collector temperature drops to only 3°C (5°F) above the solar storage temperature, the pumps are switched off. On the high limit side, when the solar storage tank reaches 85°C (185°F), the pumps are also switched off.

The existing heating source (166,000 Btuh converter) brings the supply water up to temperature when the solar storage is below 50°C (120°F). Other system components include isolation valves, Btu meters, temperature sensors, thermometers, and an automatic proportional mixing valve that blends cold supply water with solar heated water when the storage is hotter than 50°C (120°F).

THERMAL PERFORMANCE

Instrumentation

Thermal instrumentation for the system involves two Btu meters, one pyranometer, and several thermometers. One Btu meter and two thermometers are used in the collection subsystem. One Btu meter and five thermometers are used in the storage subsystem.

The collector Btu meter used to measure the collected heat is an American displacement meter model 311. The meter was set at the factory with a rated accuracy of 1%. The positioning of the meter in the collector loop is in the basement three floors beneath the collectors which are mounted on the roof. Because the collector circulation fluid is 50% ethylene glycol in water a correction factor for the heat capacity was needed. Due to the differences in specific heat and density for water and 50% ethylene glycol, a correction factor of 0.886, which is equal to the actual heat transferred divided by the meter reading, was applied to obtain the correct heat flow through the collector Btu meter.

The storage Btu meter used to measure heat actually transferred to the potable water is also an American displacement meter model 311 having a

factory rated accuracy of 1%. The meter was sized so that the largest expected water usage could be maintained through the meter. In spite of this, extremely low flow rates are common where the Btu meter accuracy drops to 20%. The storage loop Btu meter is located in the potable hot water line and thus measures the heat used to heat the cold water make up.

The pyranometer is an Eppley 8-48 model and is classified a 2nd class pyranometer by the Commission for Instruments and Methods of Observation.[2] The pyranometer was first calibrated in November, 1977. The pyranometer is placed on the roof of the building near the collectors in a horizontal position. The decision to place the pyranometer in a horizontal position was based on cost considerations and the fact that horizontal data, i.e. global insolation, is generally more useful and meteorologically important.

Data

The Epply 8-48 pyranometer automatically integrates and prints the integrated radiation hourly. The Btu meters are read 6 times daily.

Incident radiation on the plane of the collectors is the basic input for any solar energy system. An expression was derived relating horizontal data with the desired radiation on a 45° titled surface. Using equations given in the ASHRAE Handbook [3] ratios ($\bar{K}_i$) of $\bar{I}_T/\bar{I}$ were derived. These values are tabulated in Table 1 (see Appendix). To find the correct $\bar{K}_i$ for any particular month the correct average diffuse sky constant ($\bar{C}$) must be known. The steps to find $\bar{C}$ and thus $\bar{K}_i$ are listed in the Appendix. The value of $\bar{I}_T$ is obtained by multiplying $\bar{K}_i$ times $\bar{I}$.

The collected energy (Q_U^A) is taken from the collector Btu meter readings. Even though there is some heat loss in the piping from the collectors on the roof to the location of the Btu meter three floors below, the estimated heat loss is only about 3%. This will lead to slightly smaller collector efficiency figures but will have a negligible effect on the overall system efficiency.

Energy used for heating hot water is designated by Q_U^*. In July, the energy used to heat hot water (Q_U^*) is taken from the storage loop Btu meter. However, in August, a modification was performed in which the hot water recirculation loop was routed through the solar storage tank. Thus for July,

$$Q_U^* = Q_U^{HW} \tag{1}$$

but for August,

$$Q_U^* = Q_U^{HW} + Q_U^{REC} \tag{2}$$

where Q_U^{HW} is the heat delivered directly to potable water, i.e. the storage loop Btu measurement, and Q_U^{REC} is the heat delivered to the hot water recirculation unit.

The energy lost is the difference in the collected energy and the energy actually used. More specifically a differentiation is established between energy lost to the outside of conditioned space (Q_L^{Ext}) and energy lost to the conditioned space (Q_L^{Int}). In our system all heat lost is in a non-conditioned mechanical service room; thus Q_L^{Int} equals zero and Q_L equals Q_L^{Ext}.

Energy to operate solar system is the electrical energy requirement of the two pumps. (See Fig. 1) The pump in the circulation loop requires 10.6 amps of current at 115 volts. The storage loop circulation pump requires 1.7 amps at 115 volts. The combined energy requirement is 1.42 kJ/sec. The length of time the solar system operates is obtained by using the known flow rate and the number of gallons passed through the collector Btu meter.

The data for the 2 months is summarized in Table 2 (see Appendix).

Results

Using a value of 2.0 for ε and

$$\bar{Q}_H = \bar{Q}_U^A - \bar{Q}_L^{Ext} - \varepsilon\bar{E} \tag{3}$$

the average daily savings in non-renewable energy resources over a month ($\bar{Q}_H$) is obtained. Other system parameters may be calculated using the following equations:

$$n_S = \frac{\bar{Q}_H}{A_C\bar{I}_T} \tag{4}$$

$$n_{ST} = \frac{\bar{Q}_H + \varepsilon\bar{E}}{A_C\bar{I}_T} \tag{5}$$

$$SCOP = \frac{\bar{Q}_U^A - \bar{Q}_L^{Ext}}{\bar{E}} \tag{6}$$

The collection subsystem efficiency may also be found by

$$n_C = \frac{\bar{Q}_U^A}{A_C\bar{I}_T} \tag{7}$$

The results for the two months of data are summarized in Table 3 (see Appendix).

A modification was incorporated into the solar system in August. As seen in Table 3, the overall efficiency and actual savings of fossil fuel energy both increased 1000% from July to August. This is due primarily to a modification of the solar system which includes a routing of the recirculated hot water through the solar storage tank. This change is shown schematically in Fig. 2 (see Appendix).

Another simple modification implemented, which does not directly affect solar system performance but was recognized to be effective by a thermal

analysis of the entire hot water system, was the installation of a timer on the hot water recirculation loop. The timer, which shuts off circulation at nights and on weekends, saves an estimated 80,000 kJ/day.

CONCLUSIONS

The overall system efficiency increased from an extremely low value in July to 8% in October. The efficiency is expected to increase further in the colder winter months ahead. The overall collection efficiency also increased during this same period of time from 31% in July to 41% in October. These trends in efficiencies are directly related to the extent of how much the solar system is being used. In the middle of an Oklahoma summer when the cold water make up temperature is 27°C (80°F) there is little demand for hot water. As the season changes hot water usage increases, and thus the need for heat to heat this hot water greater increases.

The change in the average daily savings in non-renewable energy resources over a month by the solar hot water heating system ($\bar{Q}_H$) from 2,500 kJ/day to 74,000 kJ/day also reflects the increased heat demand.

Finally, the trend in the thermal performance of the solar system indicates that up to 50% of the light impinging on the solar collector can be collected but that with our system perhaps only 15% of the incident light can actually be used to heat potable hot water.

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NOMENCLATURE

A_C = Area of the solar collector array

$\bar{E}$ = Average daily electrical power required to operate the solar system over a month

ε = Factor to convert the electrical energy usage to its thermal energy equivalent

$\bar{I}$ = Average daily integrated solar radiation per unit area on a horizontal surface over a month

$\bar{I}_T$ = Average daily integrated solar radiation per unit area on the tilted surface of the solar collector array over a month

$$\bar{K}_i = \bar{I}_T/\bar{I} \qquad (8)$$

n_C = Average daily collector subsystem efficiency over a month

S = Average daily solar heating efficiency over a month

n_{ST} = Average daily solar heating system thermal efficiency over a month

$\bar{Q}_H$ = Average daily savings in non-renewable energy resources over a month by the solar hot water heating system

$\bar{Q}_L^{Ext}$ = Average daily solar system heat losses to the exterior of the conditioned space over a month

$\bar{Q}_L^{Int}$ = Average daily solar system heat losses to the interior of the conditioned space over a month

$\bar{Q}_U$* = Average daily solar heat delivered to hot water load over a month

$\bar{Q}_U^{HW}$ = Average daily solar heat delivered to heat ground water over a month

$\bar{Q}_U^{REC}$ = Average daily solar heat delivered to heat recirculated hot water over a month

SCOP = Solar coefficient of performance

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APPENDIX

Steps to find $\bar{K}_i$

(1) Sum pyranometer daily total insolation values.

e.g. July 1979 $\quad 691300 \frac{kJ}{m^2}$ $(61032.6 \frac{Btu}{ft^2})$

(2) Divide (1) by the number of days in month to get average daily total insolation ($\bar{I}$)

e.g. $691300 \frac{kJ}{m^2} \div 31 = 22,300 \frac{kJ}{m^2\text{-day}}$ $(1968.8 \frac{Btu}{ft^2\text{-day}})$

(3) Calculate ratio $K_T = \frac{(2)}{I_o}$ where I_o values are found in Table 4

$$\text{e.g. } \frac{22,300 \frac{kJ}{m^2\text{-day}}}{40,220 \frac{kJ}{m^2\text{-day}}} = .554$$

(4) Using (3) and Fig. 3 of Lui & Jordan [4] find the corresponding ratio, Daily Diffuse Radiation on a Horizontal Surface ÷ Daily Total Radiation on a Horzontal Surface.

e.g. .554 → .385

Note that (4) will go toward 1.0 for very cloudy conditions and tends toward 0.0 for very clear conditions regardless of the position of the sun in the sky. For purposes of simplifying the calculations of C this ratio (4) was restricted to one hour from solar noon.

(5) Using Fig. 4 and (4), find the corresponding constant C

e.g. .385 → C = .62

(6) Knowing C ~ $\bar{C}$ the correct $\bar{K}_i$ is obtained from Table 1 by linear interpolation.

e.g. .62 → .8849 = $\bar{K}_i$

TABLE 1

WEIGHTED AVERAGE RATIOS OF $\bar{I}_T/\bar{I}$ ($\bar{K}_i$) FOR SEVERAL DIFFUSE SKY CONSTANTS (C)

C	$\bar{K}_i$ for	Jan.	Feb.	March	April	May	June
.05		1.9834	1.6259	1.3002	1.0340	.8705	.8223
.10		1.8597	1.5562	1.2702	1.0296	.8784	.8334
.15		1.7654	1.5013	1.2452	1.0248	.8840	.8417
.20		1.6897	1.4561	1.2238	1.0201	.8882	.8483
.25		1.6272	1.4178	1.2052	1.0156	.8914	.8537
.30		1.5745	1.3849	1.1888	1.0114	.8940	.8582
.35		1.5292	1.3562	1.1742	1.0074	.8962	.8620
.40		1.4900	1.3309	1.1611	1.0037	.8979	.8653
.45		1.4555	1.3084	1.1493	1.0002	.8994	.8682
.50		1.4250	1.2883	1.1386	.9970	.9006	.8707
.55		1.3977	1.2701	1.1289	.9940	.9017	.8730
.60		1.3733	1.2537	1.1199	.9912	.9026	.8750
.65		1.3512	1.2387	1.1117	.9886	.9034	.8768
.70		1.3312	1.2250	1.1041	.9861	.9041	.8785
.75		1.3129	1.2124	1.0971	.9838	.9047	.8800
.80		1.2961	1.2008	1.0906	.9816	.9053	.8813
.85		1.2807	1.1901	1.0845	.9795	.9058	.8826
.90		1.2665	1.1801	1.0788	.9776	.9062	.8837
.95		1.2665	1.1708	1.0735	.9758	.9066	.8848
1.00		1.2412	1.1622	1.0686	.9742	.9070	.8858

C	$\bar{K}_i$ for	July	Aug.	Sept.	Oct.	Nov.	Dec.
.05		.8386	.9691	1.1994	1.5161	1.8924	2.1159
.10		.8485	.9698	1.1800	1.4606	1.7835	1.9696
.15		.8559	.9693	1.1632	1.4163	1.6999	1.8592
.20		.8617	.9682	1.1484	1.3794	1.6324	1.7714
.25		.8663	.9669	1.1353	1.3480	1.5763	1.6994
.30		.8702	.9654	1.1236	1.3208	1.5287	1.6390
.35		.8702	.9654	1.1131	1.2970	1.4877	1.5875
.40		.8762	.9624	1.1037	1.2759	1.4520	1.5430
.45		.8786	.9609	1.0951	1.2570	1.4206	1.5041
.50		.8808	.9595	1.0872	1.2401	1.3927	1.4698
.55		.8826	.9581	1.0800	1.2248	1.3678	1.4393
.60		.8843	.9568	1.0734	1.2108	1.3453	1.4120
.65		.8857	.9555	1.0673	1.1981	1.3250	1.3874
.70		.8871	.9543	1.0616	1.1865	1.3065	1.3651
.75		.8883	.9532	1.0563	1.1757	1.2896	1.3449
.80		.8894	.9520	1.0514	1.1658	1.2741	1.3263
.85		.8904	.9510	1.0468	1.1566	1.2599	1.3093
.90		.8913	.9500	1.0426	1.1481	1.2467	1.2937
.95		.8921	.9490	1.0385	1.1401	1.2345	1.2792
1.00		.8929	.9481	1.0348	1.1326	1.2231	1.2658

TABLE 2

HEAT FLOWS FOR SOLAR SYSTEM

	units	July, 1979	Aug., 1979	Sep., 1979	Oct., 1979
$\bar{I}$	kJ/day-m^2	22,300	22,700	20,300	14,000
$\bar{K}_i$	unitless	.885	.962	1.282	1.272
$\bar{I}_T$	kJ/day-m^2	19,800	21,800	26,000	17,800
$\bar{Q}_U^A$	kJ/day	329,000	401,000	513,000	400,000
$\bar{Q}_U^{HW}$	kJ/day	47,300	35,000	40,000	42,000
$\bar{Q}_U^{REC}$	kJ/day	---	40,000	80,000	80,000
$\bar{Q}_L^{Ext}$	kJ/day	282,000	326,000	393,000	278,000
$\bar{E}$	kJ/day	22,400	24,900	31,100	24,000

TABLE 3

SYSTEM PARAMETERS

	units	July, 1979	Aug., 1979	Sep., 1979	Oct., 1979
$\bar{Q}_U^*$	kJ/day	47,300	75,000	120,000	122,000
$\bar{Q}_H$	kJ/day	2,500	25,000	58,000	74,000
n_S	unitless	0.2%	2.1%	4.7%	7.7%
n_{ST}	unitless	4.4%	6.3%	9.8%	12.6%
SCOP	unitless	2.1	3.0	3.9	5.1
n_C	unitless	30.6%	33.9%	36.3%	41.4%

TABLE 4

AVERAGE EXTRATERRESTIAL DAILY HORIZONTAL INSOLATION
IN kJ / m^2-DAY

I_o	Month
17340	January
22720	February
29180	March
34930	April
39640	May
40700	June
40220	July
36720	August
31070	September
24470	October
18430	November
15730	December

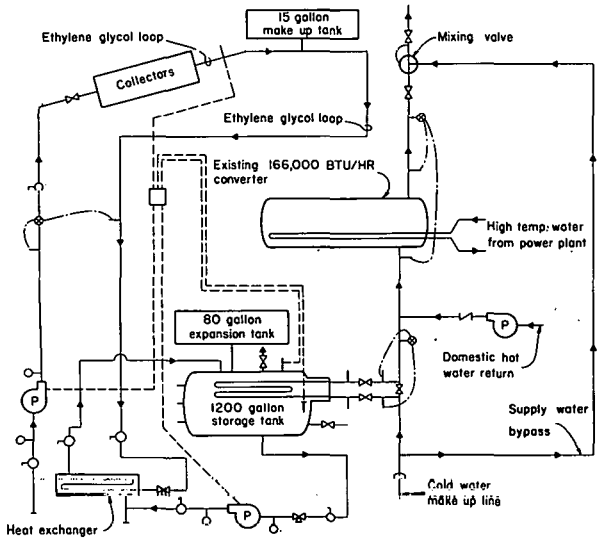


Fig. 1 Schematic of Solar System Without Domestic Hot Water Return Rerouting

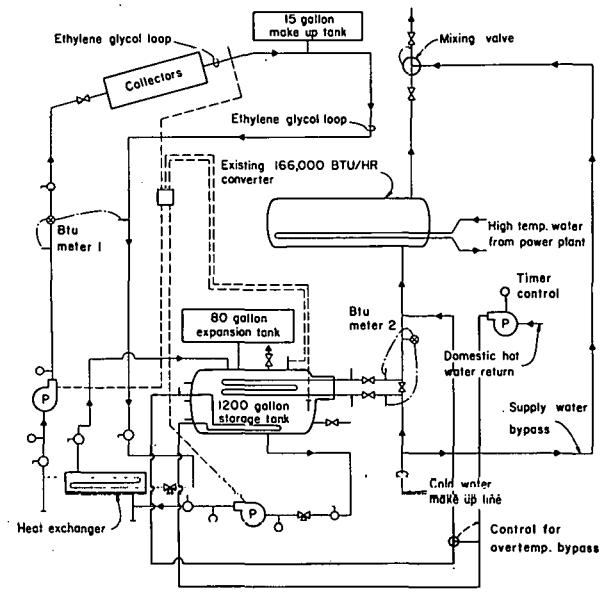


Fig. 2 A Schematic of the Solar System Which Includes the Modification - Routing the Recirculated Hot Water Through the Solar Storage Tank

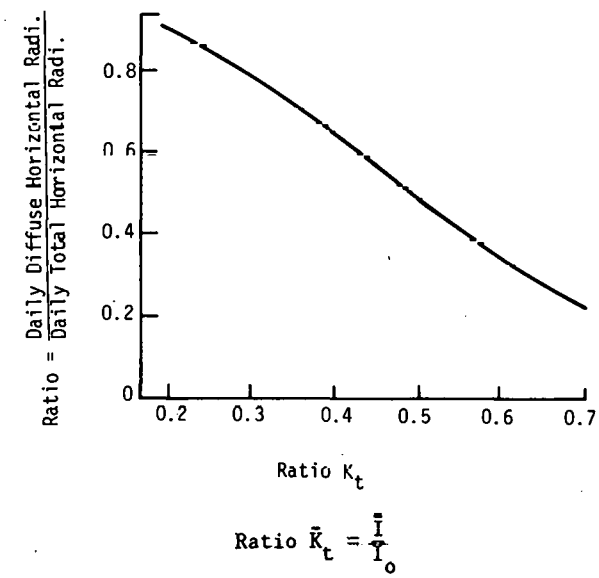


Fig. 3 The Ratio of the Daily Diffuse Radiation to the Daily Total Radiation as a Function of the Cloudiness Index K_t

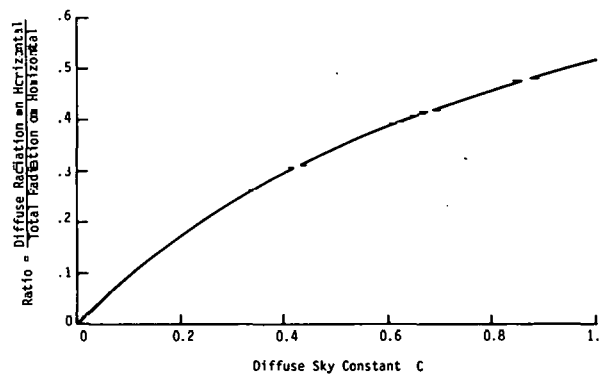


Fig. 4 The Ratio of the Diffuse Horizontal Radiation to the Total Horizontal Radiation as a Function of the Diffuse Sky Constant C

PERFORMANCE OF SIX SOLAR DOMESTIC HOT WATER SYSTEMS IN THE MID-ATLANTIC REGION

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ABSTRACT

In order to provide data required for the validation of computer programs for solar domestic hot water systems, the staff of the National Bureau of Standards (NBS) fabricated and instrumented six typical residential-size systems at its Gaithersburg, Maryland site. Three of the six systems utilize a single hot water tank and the other three utilize two tanks. Three different methods of freeze protection are being utilized among the systems: water-ethylene glycol mixture in the collector loop with an intermediate heat exchanger, drain-down using solenoid valves to dump the water from the collectors, and air-heating collectors in conjunction with an air-water heat exchanger. One system relies on the thermosyphon principle while the other five require electric power for fans or pumps. The systems have been operating since June, 1978. This paper describes the testing done and gives the experimental results for the first year of operation.

INTRODUCTION

Three computer programs, TRNSYS, [1], F-CHART [2], and SOLCOST [3], are currently being used for the design and evaluation of solar space heating and domestic hot water systems. Although widely used, the accuracy of their predictions have seldom been verified with data from controlled experiments. The staff of NBS has undertaken a project to provide the experimental data required for the validation of these computer codes for solar domestic hot water (SDHW) systems. This project involves gathering detailed performance data on six instrumented SDHW systems over a twenty-four month period. The purpose of this paper is to describe the experimental facility and discuss the experimental results for the first year of operation. The comparison between the experimental results and the computer predictions have been presented previously for the first eight months of operation [4] and will be presented for the complete first year in February, 1980 at the semi-annual meeting of ASHRAE [5].

SOLAR DOMESTIC HOT WATER TEST FACILITY

Description of Systems

The SDHW test facility consists of six independent systems. A description of each system follows.

Single Tank Direct. This system consists of two solar collectors connected in parallel, one water storage tank, flow control valves, an on-off differential temperature controller with freeze protection circuitry, a pump, piping, and insulation.

The collectors used on all five liquid systems are a single-glass cover, flat-plate, liquid collector. As noted above, this system uses two collector modules, the other four systems use three modules. The glass is tempered and has a low iron content with etched surface lines to reduce reflection. A steel absorber plate is formed around copper flow tubes and then coated with black chrome. Each collector has a gross area of 1.67 m^2 (18.0 ft^2) and a corresponding aperture area of 1.44 m^2 (15.5 ft^2). All collectors utilized at the facility are facing due south with their slope set to 39°.

The water storage tank in this system is a 310 liter (82 gal.) conventional electric water heater. Within the tank are two 4.5 kW heating elements. Only the top one is utilized in this experiment. The cold water inlet consists of a dip tube extending 0.41 m (53.0 in.) down from the upper surface of the tank. The tank is insulated with glass fiber insulation having a thickness of 5.1 cm (2.0 in.). The upper thermostat is set to maintain a temperature of 60°C (140°F). A hot water mixing valve tempers the 60°C water down to 49°C (120°F). The same thermostat setting on the auxiliary heater(s) and mixing valve are used on all the systems.

A controller is used to actuate the circulator pump when a temperature difference of 8.9°C (16°F) exists between the collector absorber plate and the storage tank temperature. A temperature difference of less than 1.7°C (3°F) stops the pump.

Collector flow rate is set at 1.7 ℓ/min (0.44 gal./min) through each collector module. This is done in the other three pumped liquid systems as well. The storage tank sensor is located on the exterior tank surface (inside the insulation) at an elevation of 15.3 cm (6.0 in.). The controller also actuates two solenoid valves to provide collector freeze protection. Freeze protection action is initiated if the absorber plate temperature falls below 2.8°C (37°F). One solenoid valve closes the supply to the collectors while the second one opens and allows drainage of the collectors onto the ground. This sequence is also employed during a power failure. An air vent and a vacuum relief valve attached to the highest point of the system allows venting of air during collector fill and eliminates a partial vacuum during drainage.

Hard copper tubing of 1.27 cm (0.50 in.) diameter is used throughout the installation except for 2.54 cm (1.00 in.) diameter headers interconnecting the two collectors. Insulation of 1.27 cm (0.50 in.) thickness, R-4, is used for the internal pipe. Exterior insulation consists of 3.18 cm (1.25 in.) thick glass fiber insulation, R-5, on the piping while 5.10 cm (2.0 in.) thick glass fiber insulation, R-8, encases the collector headers. Similar piping and insulation is used on all the systems except as noted below.

Double Tank Direct. The preheat storage tank is identical to that used with the single tank direct system. Both 4.5 kw heating elements have been disconnected. The auxiliary tank is a 159 liter (42 gal.) conventional electric water heater. Both 4.5 kw heating elements are utilized to maintain the 140°F (60°C) set point temperature. Water from the preheat tank enters the second tank through a dip tube extending 1.04 m (41.0 in.) down into the tank. The tank is insulated with glass fiber insulation having a thickness of 5.1 cm (2.0 in.).

All pumps, control components, and temperature set points are identical to those utilized with the single tank direct system.

Single Tank Indirect. The 310 liter (82 gal.) water storage tank has an integral 4.5 kw heating element located in the upper portion of the tank. A double-wall heat exchanger jacket surrounds the water tank. Heat transfer fluid is a mixture of ethylene glycol (40% by weight) and distilled water. The heat exchanger jacket has an area of 1.58 m^2 (17.0 ft^2) which is attached to the surface of the tank by mechanical bonding. Insulation surrounding the heat exchanger and tank consists of 7.62 cm (3.0 in.) thick insulation.

A controller actuates the pump when a temperature difference of 10°C (18°F) exists between the absorber plate and a tank surface temperature sensor. This sensor is located at a height of 0.74 m (29.0 in.). A 1.7°C (3°F) temperature difference stops the pump.

Double Tank Indirect. The hot water preheat tank is identical to the tank used in the single tank indirect system except it lacks an integral heating element. The auxiliary tank is identical to that used in the double tank direct system. The controller is identical to that used in the single tank indirect system.

Thermosyphon System. The water storage tank is a 250 liter (66 gal.) conventional electric water heater. The bottom heating element has been disconnected for this experiment, while the upper 4.5 kw element maintains the upper portion of the stored water at 60°C (140°F).

Freeze protection is provided in the same fashion as those utilized in the single and double tank direct system. A check valve on the collector return piping prevents reverse thermosyphoning as well as tank drainage during freeze periods.

Hard copper tubing of 2.54 cm (1.0 in.) diameter is used exclusively on the collector flow loop. All piping bends were kept to a minimum to minimize pressure drops. Insulation of 1.27 cm (0.50 in.) thickness, R-4, is used for internal pipe insulation. Exterior insulation consists of a 5.10 cm (2.0 in.) thick, R-8, glass fiber insulation.

Air System. This consists of four air collectors, a cross-flow air-water heat exchanger, two water storage tanks, a controller, pump, fan, piping, ducts, and insulation.

Each collector has a gross area of 1.77 m^2 (19.1 ft^2) and a corresponding aperture area of 1.53 m^2 (16.5 ft^2). The collector has two 0.32 cm (0.125 in.) low iron tempered glass covers with neoprene perimeter gaskets. The absorber is a 18 gauge steel sheet with a baked-on ceramic coating. Air is circulated thru channels beneath the absorber plate.

The controller simultaneously actuates the circulator and fan when a 22.2°C (40°F) temperature difference exists between the internal collector air passages and the preheat storage tank. The storage tank sensor is located on the outer surface of the tank at an elevation of 15.3 cm (6.0 in.). Circulation ceases when the temperature difference is less than 11°C (20°F).

The pre-heat tank and the auxiliary tank are identical to those used in the double tank direct system. The collector supply and return ducts are 20.32 cm (8.0 in.) flexible ducting insulated with 1.27 cm (0.50 in.) foil-faced insulation.

Inlet Water Temperature Control System

The inlet water temperature to all six SDHW systems is held constant over a given month and changed on a month-by-month basis to match the near-by Potomac River temperature. The temperature of the water is controlled by means of a 310 ℓ (82 gal.)

storage tank with one heating element in combination with a 0.75 ton chiller. The water is circulated continuously past the inlet of each system through the chiller and the 310 ℓ tank. A temperature controller controls the electric heating element in the tank which supplies the energy required to heat the water if necessary. A thermostat in the chiller actuates the chiller to remove heat, if so required. The inlet water temperature control system maintains the set point temperature to ± 2.5°C (± 4.5°F).

Automated Hot Water Draw System

The outlet of each hot water system interconnects with a main header. A solenoid valve at the center of the header releases the flow to a drain when actuated. An electronic timer combined with a stepping relay selects an interval timer range from 1.5 minutes to 10 minutes in duration. A throtting valve at the exit of each system maintains a flow rate of 3.8 ℓ/min (1.0 gal/min) when the solenoid valve is open. Thus when a given interval timer is energized for its set time interval, a corresponding amount of water is drawn from each of the six systems. Each system is providing approximately 265 ℓ (70 gal.) of hot water per day at 49°C (120°F). The load schedule used in this experiment was developed by J. Mutch of the Rand Corporation [6] and is cited in the TRNSYS User's Manual [1] as a typical hot water schedule. It consits of a slight morning peak (9 am) and an evening peak (8 pm) with water drawn throughout the 24-hour period except for 1 am - 6 am.

Instrumentation

Located within each water storage tank are Type-T thermocouples located in 15.2 cm (6.0 in) increments. Thermocouples also monitor the collector inlet and outlet temperature for each system. The inlet and exit water temperatures are measured with thermocouples and a thermopile measures the temperature difference during draw down.

A kWh meter measures the auxiliary energy consumed by the electric heating elements. A Wh meter measures the energy used by the circulators, controls, solenoid valves, etc. for each system.

Each systems' water consumption is measured by two flow totalizers. One is used to measure the total amount of water which has been drawn off, while the second one, the quantity of water which actually goes through the solar storage tanks.

A rotometer is used to measure the flow rate of the fluid through the collectors of each liquid system. A three valve bypass arrangement is included on each liquid system such that a turbine flow meter can be installed in the collector flow loop. This capability allows the flow rates to be continuously recorded if desired.

An elapsed time meter connected to each system's controller is used to measure the amount of time the circulators are in operation.

Recorded meterological information includes incident radiation on the horizontal surface, tilted surface, direct beam radiation, wind speed, wind direction, and ambient temperature.

A data acquisition system scans all channels in ten-minute intervals and is interfaced with a magnetic 9-track tape drive unit.

EXPERIMENTAL RESULTS

Detailed experimental results for two of the six SDHW systems for a period of 12 months are shown in Tables 1 through 3. Similar results for the other four systems are not shown here. The headings for the various columns of data are defined in the Nomenclature section. In general, the single tank systems performed slightly better than the double tank systems. From Tables 1 to 3, it is seen that even though the double tank system had a higher collector efficiency (η_c), the single tank system had a higher solar system efficiency (η_s) and a higher system thermal efficiency (η_{st}). The net electric energy savings when compared to a reference conventional electrical domestic hot water heater was also larger for the single tank system. This is because the large heat loss from the auxiliary tank which had to be compensated for by the electric heating elements overshadowed the advantage of a higher collector efficiency in the double tank system. The reference conventional system used in this calculation consisted of an electric 197 ℓ (52 gal.) hot water heater supplying the same quantity of hot water as the solar system it is compared with. This method of calculating energy savings was used in order that all systems could be compared against the same reference. In this paper, all the heat losses from the systems were considered lost, and electric energy was treated as a raw non-renewable energy source at the building. The source of the electric energy and the associated electrical generation efficiency were not considered.

The fractional energy savings for each of the six systems during this twelve-month period is given in Table 4. The fractional energy savings is defined as:

$$E = 1 - \frac{Q_{SHE_S}}{Q_{SHE_C}}$$

Also included in Table 4 is the fractional energy savings when the auxiliary energy needed to run the pump, controls, solenoid valves, etc are included. This quantity is defined as

$$E_{AUX} = 1 - \frac{Q_{SHE_S} + Q_{AUX}}{Q_{SHE_C}}$$

From Table 4 it is seen that on an annual basis, the single tank systems had a larger fractional

energy savings than the double tank systems, with the thermosyphon system performing the best. It should be noted that the single tank direct system had one less collector panel than the other liquid systems. It still had a better performance than the double tank systems.

ACKNOWLEDGMENTS

The authors would like to express their appreciation to Byard D. Wood of Arizona State University for his assistance in this project. Appreciation is also extended to Chuck Terlizzi who operates the SDHW test facility and Cathy Scarborough who wrote the necessary software for the reduction of the experimental data. This project was sponsored by the Active Building Systems Branch, Systems Development Division, Office of Solar Applications, U.S. Department of Energy, Washington, D.C. 20545.

NOMENCLATURE

A_C = gross area of collectors, m^2

E = fractional energy savings excluding energy consumed by pumps, controls, solenoid valves, etc., dimensionless

E_{AUX} = fractional energy savings including energy consumed by pumps, controls, solenoid valves, etc., dimensionless

f = solar fraction, dimensionless, = Q_u^*/Q_{LD}

I_T = daily average integrated solar radiation per unit area on the collector surface over the month, $mJ/(m^2 \cdot day)$

Q_{AUX} = daily average energy consumed by pumps, controls, solenoid valves, etc., mJ

Q_L = daily average heat loss from pipes and storage tank in the solar portion of the system, mJ

Q_{LD} = daily average hot water load, mJ

Q_{SAVE} = daily average electrical energy saved when compared with a reference conventional electric hot water heater, mJ

Q_{SHE_C} = daily average energy consumed by a conventional hot water system supplying the same quantity of hot water as the solar system. The conventional system consists of an electric 197 ℓ (52 gal.) hot water tank having standby losses of 11,150 kJ (10,570 Btu) per day, mJ

Q_{SHE_S} = daily average energy consumed by integral auxiliary heating elements in the solar hot water system, mJ

Q_{STO} = daily average energy delivered by the collectors to the solar storage tank, mJ

Q_u = daily average energy collected by the collectors, mJ

Q_u^* = daily average solar energy delivered to the hot water load, mJ

SCOP' = solar coefficient of performance = Q_u^*/Q_{AUX}, dimensionless

η_c = daily average solar collector efficiency = $Q_u/(A_c I_T)$, %

η_s = daily average solar hot water system efficiency $(Q_u^*-Q_{AUX})/A_c I_T)$, dimensionless

η_{ST} = daily average solar hot water system thermal efficiency = $Q_u^*/(A_c I_T)$, dimensionless

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TABLE 1. Performance Data for the Double Tank Direct SHDW System for the Period of July, 1978 thru June, 1979

MONTH	I_T	Q_u	η_c	Q_{STO}	Q_L	Q_u^*	Q_{LD}	f	Q_{AUX}	Days Operational
-	mJ/m^2	mJ	%	mJ	mJ	mJ	mJ	-	mJ	-
7	18.8	37.4	39.6	35.3	11.7	25.7	32.2	.798	4.7	20
8	17.7	33.3	37.5	31.6	11.9	21.4	26.3	.814	4.5	27
9	18.5	33.5	36.1	31.7	10.8	22.7	28.3	.804	4.5	24
10	17.0	35.8	42.0	34.4	8.3	27.5	39.5	.695	4.0	21
11	8.7	20.9	47.9	20.5	2.6	18.3	44.4	.411	3.4	22
12	11.8	26.2	44.2	25.6	4.3	21.9	43.4	.505	2.9	14
1	9.9	21.7	43.7	21.4	1.5	20.2	44.5	.453	2.2	23
2	14.9	24.3	32.5	23.9	1.7	22.6	48.9	.463	2.2	18
3	17.0	37.7	44.2	36.6	7.3	30.4	44.0	.691	3.7	29
4	15.3	30.1	39.2	29.1	6.4	23.7	41.1	.577	4.0	26
5	15.8	32.3	40.7	30.8	9.1	23.2	36.3	.640	4.5	18
6	16.4	35.9	43.7	34.4	9.9	26.0	39.6	.656	4.6	27

Note: Heat loss from the auxiliary tank was not included in Q_L.

TABLE 2. Performance Data for the Single Tank Indirect SDHW System for the Period July, 1978 thrugh June, 1979.

MONTH	I_T	Q_u	η_c	Q_{STO}	Q_L	Q_u^*	Q_{LD}	f	Q_{AUX}	Days Operational
-	mJ/m^2	mJ	%	mJ	mJ	mJ	mJ	-	mJ	-
7	18.8	32.4	34.3	30.0	2.9	29.5	36.0	.819	2.4	20
8	17.7	27.5	30.9	25.9	4.3	23.2	30.1	.771	2.1	27
9	18.5	27.8	29.9	25.9	3.7	24.1	30.7	.785	2.4	24
10	17.0	25.5	29.9	23.9	3.2	22.3	33.6	.664	1.8	21
11	8.7	7.6	17.4	7.0	0.6	7.0	39.6	.177	1.2	22
12	11.8	14.4	24.3	13.7	0.7	13.7	48.6	.282	1.3	14
1	9.9	12.8	25.8	12.2	0.6	12.2	45.0	.272	1.1	23
2	14.9	16.9	22.6	16.0	1.8	15.1	46.5	.325	1.3	18
3	17.0	28.9	33.9	27.5	2.7	26.2	49.3	.531	1.9	29
4	15.3	24.0	31.2	22.7	1.8	22.2	41.8	.531	1.7	26
5	15.8	26.1	32.9	24.6	2.6	23.5	39.1	.601	2.0	18
6	16.4	27.9	33.9	26.2	2.9	25.0	37.2	.672	2.2	27

Note: Heat loss from the part of the storage tank containing the electrical heating element was not included in Q_L.

TABLE 3. Solar System Performance Parameters for Two of the Six SDHW Systems at NBS (July, 1978 - June, 1979)

	Double Tank Direct System					Single Tank Indirect System				
Month	Q_u^*	η_s	η_{st}	SCOP'	Q_{SAVE}	Q_u^*	η_s	η_{sT}	SCOP'	Q_{SAVE}
-	mJ	-	-	-	mJ	mJ	-	-	-	mJ
7	25.7	.223	.272	5.5	23.5	29.5	.287	.313	12.3	34.9
8	21.4	.190	.241	4.8	19.2	23.2	.237	.261	11.0	28.9
9	22.7	.196	.244	5.0	21.3	24.1	.234	.260	10.0	29.6
10	27.5	.275	.322	6.9	25.7	22.3	.240	.261	12.4	28.5
11	18.3	.341	.419	5.4	16.5	7.0	.133	.160	5.8	13.7
12	21.9	.321	.370	7.6	19.5	13.7	.209	.231	10.5	20.4
1	20.2	.362	.406	9.2	18.0	12.2	.223	.245	11.1	19.1
2	22.6	.273	.302	10.3	20.1	15.1	.184	.202	11.6	21.9
3	30.4	.313	.356	8.2	27.6	26.2	.285	.307	13.8	32.3
4	23.7	.256	.309	5.9	21.2	22.2	.267	.289	13.1	28.3
5	23.2	.236	.293	5.2	20.4	23.5	.271	.296	11.8	29.5
6	26.0	.260	.316	5.7	22.8	25.0	.277	.304	11.4	30.7

TABLE 4. Fractional Energy Savings of Six SDHW Systems at NBS
%

Month	Single Tank Direct		Double Tank Direct		Single Tank Indirect		Double Tank Indirect		Double Tank Air		Single Tank Thermospyhon	
July	55.3	(46.9)	65.1	(54.3)	79.2	(74.1)	62.2	(56.8)	47.2	(36.6)	71.9	(71.9)
August	68.8	(59.2)	63.4	(51.3)	75.5	(70.0)	64.0	(58.3)	45.2	(32.4)	79.7	(79.7)
September	67.1	(57.5)	65.2	(53.9)	76.5	(70.7)	62.8	(57.6)	46.1	(34.7)	77.4	(77.4)
October	61.9	(54.4)	58.6	(50.8)	67.7	(63.7)	55.2	(51.1)	43.9	(35.5)	62.8	(62.2)
November	36.1	(31.1)	35.8	(29.7)	29.5	(27.0)	34.2	(31.8)	29.6	(25.6)	32.5	(29.3)
December	38.5	(34.1)	41.1	(35.7)	36.3	(34.2)	36.1	(33.2)	31.1	(25.5)	53.2	(50.2)
January	35.0	(31.8)	36.2	(32.3)	35.9	(34.0)	32.3	(30.3)	27.2	(23.2)	35.3	(33.6)
February	34.2	(31.4)	37.1	(33.4)	40.1	(37.9)	32.7	(30.5)	26.7	(22.1)	34.3	(33.2)
March	50.8	(44.9)	56.8	(50.1)	56.5	(53.4)	48.6	(45.5)	39.2	(32.6)	70.1	(67.1)
April	50.6	(44.3)	48.4	(40.6)	56.6	(53.4)	44.2	(40.6)	39.2	(32.2)	63.2	(58.4)
May	53.6	(46.2)	52.4	(43.0)	62.6	(58.7)	46.0	(41.8)	38.1	(29.1)	61.1	(56.6)
June	55.5	(47.2)	54.0	(45.0)	68.0	(63.5)	51.6	(46.8)	36.9	(26.4)	69.8	(68.6)
July, 1978 thru June, 1979	50.4	(44.0)	50.8	(43.1)	56.7	(53.2)	47.3	(43.6)	37.4	(29.7)	60.2	(58.2)

( ) Fractional energy savings including energy necessary to run pumps, controls, solenoid valves, etc.

NOTES

INTERIM PERFORMANCE RESULTS FROM THE NORTHEAST STATES' SOLAR WATER HEATER PERFORMANCE MONITORING PROGRAM

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ABSTRACT

The results of the Northeast States Solar Water Heater Performance Monitoring Program will be of tremendous value to the solar industry, solar-oriented government agencies and the general public. They will establish the state of the technology of Northeast solar water heating from 1978 through 1980 and assess the general effects of HUD Initiative approval, program warranty requirements, installer training and solar industry maturation on the quality of the installed products. The program will also assess the application of low-cost systems for the monitoring of solar water heater performance.

The 200 sites chosen for this program consist of solar water heaters installed under the United States Department of Housing and Urban Development (HUD) Solar Hot Water Initiative. At the present time (10/1/79), 90 sites have been monitored and 510 months of operational data have been collected. The performance results from 166 months of monitoring data are presented with performance intervals ranging from three to twelve months. A description of the monitoring technique is included with a discussion about water heating system losses and data validation.

INTRODUCTION

During 1978 the Northeast Solar Energy Center negotiated a contract with the United States Department of Energy (DOE) and was awarded funds to monitor the performance of 200 solar water heaters. These heaters were to be installed under the United States Department of Housing and Urban Development (HUD) Solar Hot Water Initiative. The HUD Initiative was introduced to the public in March of 1977 and provides 10,876 grants of $400 each to homeowners purchasing and installing state-approved systems. States participating in the program are the ten Northeast states listed below and Florida.

The Northeast States' Solar Water Heating Performance Monitoring Program was initiated with the ten Northeast state energy offices on August 11, 1978. The number of monitoring sites allocated to each state is approximately in proportion to the number of HUD grants planned. Table 1 shows the distribution of the 200 monitoring sites and the 9,217 HUD grants among the participating states.

TABLE 1

GRANT AND MONITORING SITE DISTRIBUTION

State	No. of HUD Grants	No. Monitoring Sites Allocated
Connecticut	750	20
Delaware	150	5
Maryland	950	20
Massachusetts	1,375	35
New Hampshire	200	5
New Jersey	1,725	35
New York	867	20
Pennsylvania	2,800	50
Rhode Island	250	5
Vermont	150	5
TOTAL	9,217	200

At the present time (10/1/79), a total of 90 monitoring systems have been installed in Delaware, Massachusetts, New Hampshire, Pennsylvania, Rhode Island, and Vermont. Since August 1978, 166 months of operational data have been analysed and will be presented.

The results of the NESEC monitoring program will be of tremendous value to the solar industry, solar oriented government agencies and the general public. They will establish the state of the technology of Northeast solar water heating from 1978 through 1980 and assess the general effects of HUD Initiative approval, program warranty requirements, installer training and solar industry maturation on the quality of the installed product.

The objective of this program can be summarized as follows:

1. Assess the effectiveness of the low-cost system in monitoring solar water heater performance. Develop correlation methods as necessary.

2. Assess the performance of the systems certified for the HUD Initiative and compare it to the results of various performance prediction methods (depending on the results of Objective 1).
3. Establish the general effect of equipment certification, HUD warranties, installer training, and solar industry maturation on the quality of the installed product.
4. Establish maintenance and service costs.
5. Establish the reliability, safety, and life of solar water heaters.

SOLAR WATER HEATING SYSTEM SELECTION

As specified under the HUD Solar Domestic Hot Water Initiative[1], all of the solar water heating systems approved for the program must provide at least 45% of the average yearly hot water requirement for a family of four of 70 gallons per day, heated from 55°F to 140°F. The performance of each solar water heating system has been predicted with the use of an F-CHART analysis by the Polytechnic Institute of New York (PINY) Solar Energy Applications Center. The final list of 741 systems recommended for approval in the HUD Solar Domestic Hot Water Initiative is available from PINY[2] or NESEC. As indicated in the summary of the objectives, once the performance data from the monitored sites are assessed and proven accurate, it will be compared to the results of various performance prediction methods. This comparison will provide operational data for the validation of various performance prediction methods.

MONITORING EQUIPMENT DESCRIPTION

The following procedures were established by NESEC to monitor the water heating systems. The 200 low-cost monitoring systems (LCS) for this program consist of:

- Two Bimetal Thermometers
- Three 3/4" x 1/2" x 3/4" C-F-C Tees
- Watt Hour Meter
- Meter Socket
- Elapsed Timer Indicator
- Mounting Bracket for Elapsed Time Indicator
- Water Meter with Couplings and Adapters
- Strainer for Flow Meter
- Pressure Guage
- 3/4" x 3/8" C-F Tee
- 3/8" x 1/4" Brass bushing
- 3/4" C-M Adapter
- 3/4" C-F Adapter
- 1/2" Brass Plug

The equipment is installed by the solar system installer and data are logged daily by the homeowner. The LCS provides data which include:

- the amount of hot water consumed
- the inlet and outlet water temperature of the water heating system
- the amount of electrical energy purchased to provide hot water
- the daily operational time of the solar water heating system

The data log is collected and the solar system is checked every two weeks by the state assigned data collector. All data are then copied and sent to NESEC once a month for processing.

PERFORMANCE EVALUATION TECHNIQUE

The LCS is installed on the auxiliary water heating system (see Fig. 1). The energy saved by the solar system is calculated via a subtractive method as discussed below.

With a conventional electrical water heater, the electrical power consumption equals the water heating load plus the energy losses from the pipes and tank.

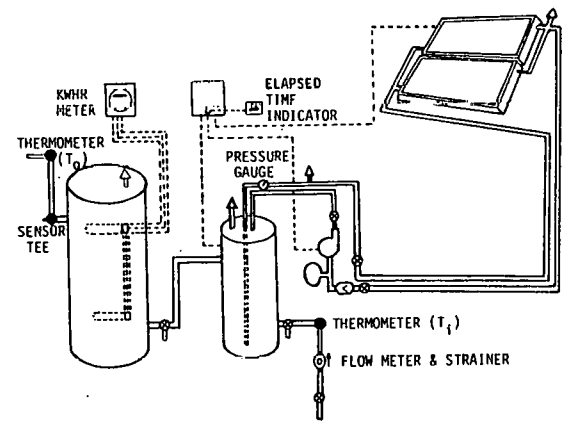


Fig. 1. Instrumentation Schematic

The total KWHR's consumed by a solar water heater with an electric backup equals the electrical consumption of the backup heating elements plus the KWHR's consumed by the solar equipment. Therefore, the KWHR's saved by the solar water heater are calculated by subtracting the solar KWHR consumption from the total conventional system load according to the following equation:

$$\text{KWHR Saved} = \frac{\text{BTU Load}}{3413} + \text{Loss KWHR} - \text{Element KWHR} - \text{Solar Equipment KWHR}$$

It is more difficult to estimate the contribution from a solar water heating system which utilizes an oil or gas backup system because of the variability of the:

- Burner efficiency
- Main burner flue loss
- Pilot flue loss

Because of this difficulty, most of the sites selected for the program have an electric backup system. Five oil or gas systems are selected for the program but for these sites we will be utilizing BTU meters to monitor the solar contribution.

A simple but more expensive and accurate method of obtaining an evaluation of an oil or gas system is by installing two BTU meters on the water heating system. The first BTU meter measures the solar storage tank inlet and outlet temperatures, multiplies the temperature differential by the flow rate through the solar storage tank, and integrates in real time the BTU's supplied to the load by the solar storage tank. A second BTU meter, measuring the outlet temperature from the backup system instead of the solar storage tank, will integrate the BTU's supplied by the complete water heating system. Once these two totals are known, the solar system's performance can be determined with the following equation:

$$\text{Solar systems \% contribution} = \frac{\text{BTU's supplied by solar storage tank}}{\text{BTU's supplied by complete water heating system}} \times 100$$

PERFORMANCE RESULTS

As of October 1, 1979, 90 solar water heating systems in six states were being monitored. The data from 26 of these systems have been analysed and exhibit monthly performances ranging from -18.4 to 94.5% solar contribution as indicated in Table 2.

Columns 1-7 in Table 2 contain raw data that are suitable for evaluation even during this early stage of the program. Columns 8-10 contain the results of preliminary calculations which are questionable at this early stage, because certain estimates had to be made for heat losses and total energy used to heat the water. Also the results do not represent annual performance since the performance intervals vary from 3-12 months.

Even though the systems installed under the HUD Initiative are supposed to provide at least 45% of the hot water for a family of four, the monitoring results indicate that some sites are either above or below expectations. This occurs since any variation from what was predicted for the hot water load, load profile, temperature difference, solar insolation, system design or installation will result in a different overall performance. The monitoring results indicate that the hot water loads vary from a minimum of 12 gallons/day to a high of 255 gallons/day. Due to this variation from the assumed heating load of 70 gallons/day, some of the performance results are above or below predicted performance expectations.

MO/YR	# OF SITES	# OF OCCUPANTS	GAL/DAY CONSUMED	OPERATION TIME/DAY	(T_o-T_i)	QPUR (KWHR)	QWAT (KWHR)	QTOT (KWHR)	%SOL
[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]
09/78	1	3.0	97.9	2.8	73.2	3.5	14.4	20.9	82.1
10/78	1	3.0	96.3	2.0	74.0	16.8	17.1	20.6	64.0
11/78	3	3.7	108.0	2.8	79.2	16.9	20.5	22.3	28.6
(min)			(73.7)	(1.3)	(72.9)				(9.0)
(max)			(153.0)	(4.4)	(86.4)				(55.2)
12/78	6	3.3	86.7	3.4	87.6	16.1	18.8	21.1	22.9
(min)			(39.5)	(0.7)	(76.7)				(12.4)
(max)			(159.7)	(4.3)	(94.8)				(51.8)
01/79	9	3.3	86.7	2.6	86.8	18.7	18.8	21.5	10.3
(min)			(36.8)	(0.9)	(76.4)				(-16.5)
(max)			(200.6)	(4.3)	(100.1)				(23.3)
02/79	13	3.5	78.2	3.5	82.7	15.8	16.5	18.7	15.0
(min)			(27.5)	(1.8)	(64.1)				(-18.4)
(max)			(255.0)	(4.8)	(101.9)				(44.9)
03/79	18	4.1	75.8	5.6	80.1	11.9	15.1	17.4	28.3
(min)			(21.3)	(1.1)	(59.1)				(-4.2)
(max)			(232.9)	(7.6)	(94.5)				(67.1)
04/79	21	3.7	77.1	5.7	80.2	12.2	15.1	17.2	25.8
(min)			(34.7)	(1.6)	(61.7)				(-5.8)
(max)			(251.1)	(8.9)	(113.8)				(58.6)
05/79	23	3.7	72.6	6.9	76.2	12.7	13.9	16.0	37.4
(min)			(26.7)	(2.6)	(51.7)				(-10.2)
(max)			(220.8)	(11.6)	(112.5)				(73.3)
06/79	23	3.7	65.0	7.5	71.6	7.5	11.2	12.9	46.0
(min)			(11.8)	(2.4)	(44.4)				(-9.1)
(max)			(198.9)	(12.5)	(106.4)				(75.8)
07/79	24	3.8	68.2	7.5	68.9	7.4	11.1	12.9	46.8
(min)			(21.1)	(3.5)	(44.4)				(-10.5)
(max)			(216.1)	(11.4)	(101.5)				(84.9)
08/79	24	3.7	72.6	7.7	69.3	6.7	12.1	14.0	52.9
(min)			(30.0)	(3.1)	(35.7)				(15.3)
(max)			(220.8)	(11.1)	(100.6)				(94.6)
AVERAGE/DAY/YEAR	3.5		82.1	4.8	77.5	12.2	15.4	17.9	38.3

TABLE 2 - Interim Performance Results

For example, one of the monitoring sites in Pennsylvania could be classified as being oversized. This site was inspected before the monitoring equipment was installed and has the following specifications:

PENNSYLVANIA SITE

GENERIC SYSTEM TYPE - Closed loop, single tank
GROSS COLLECTOR AREA - 72 square feet
BACKUP TANK SIZE - 120 gallons
ESTIMATED LOSSES - 3.03 KWHR/day
SOLAR FLUID TYPE - Propylene glycol/water (50/50)
PUMP RATING - .083 Hp
INSTALLED SYSTEM COST - $3,000

The six month performance evaluation has revealed that the solar contribution (%SOL) is higher than predicted for that sized system (45 vs 43%). The data in Table 3 show that the hot water load for the site was only 22.3 gallons/day for three adults. Since the consumption was minimal, the solar system operated an average of only 3.7 hours/day as compared to 6.8 hours/day for surrounding monitoring sites. In the absence of the HUD 45% requirement, prior knowledge of the hot water demand may have resulted in the installation of a smaller sized, lower costing solar system with a similar performance.

TABLE 3 SIX MONTH PERFORMANCE EVALUATION FOR PENNSYLVANIA SITE (3/79-8/79)

GAL/DAY CONSUMED	OPERATION TIME/DAY	QPUR (KWHR)	QWAT (KWHR)	QTOT (KWHR)	%SOL
21.3	4.0	3.99	3.74	6.88	42.0
22.4	3.2	4.60	4.03	7.16	35.8
26.7	3.8	3.75	3.84	6.79	44.8
23.3	4.0	2.91	3.28	6.24	53.3
21.1	3.6	3.06	2.78	5.74	46.7
24.7	3.6	2.51	2.14	4.60	45.5
23.3	3.7	3.47	3.30	6.24	44.7

One example of a site that has improved the performance of a solar system by altering the load profile is located in Vermont. This site was inspected before the monitoring equipment was installed and has the following specifications:

VERMONT SITE

GENERIC SYSTEM TYPE - Closed loop, single tank
GROSS COLLECTOR AREA - 100 square feet
BACKUP TANK SIZE - 120 gallons
ESTIMATED LOSSES - 1.69 KWHR/day
SOLAR FLUID TYPE - Propylene glycol/water (50/50)
PUMP RATING - .083 Hp
INSTALLED SYSTEM COST - $2,430

The one year performance evaluation has revealed that the solar system contribution is higher than predicted (63 vs 49% annual contribution). The data in Table 4 show that this site used 105 gallons/day even though only three adults occupied the site.

TABLE 4 ONE YEAR PERFORMANCE EVALUATION FOR VERMONT SITE (9/78-8/79)

GAL/DAY CONSUMED	OPERATION TIME/DAY	QPUR (KWHR)	QWAT (KWHR)	QTOT (KWHR)	%SOL
105	5.1	8.9	22.5	24.2	63

Since the consumption was higher than predicted (105 gallons/day vs 70 gallons/day), it would be expected that the solar contribution would be smaller. But in this case, the homeowners controlled the backup system so the solar systems full potential could be realized. They achieved this better performance by installing a time clock on the auxilliary heating element and shut-off the backup at predetermined intervals, thereby reducing losses.

Variations in other parameters, specifically water temperature differences, solar insolation, system design or installation, could result in different performances from the systems than predicted. Detailed analysis of these variations is being undertaken, as described below, but as of this writing, no specific results are available.

TANK AND PIPE ENERGY LOSS EVALUATION

The energy supplied to conventional domestic hot water systems is either lost or used. The quantity of conventional energy supplied is relatively easily metered, but because the losses are distributed throughout the system it is difficult to measure system efficiency. Measurement of the actual energy delivered to the load is more easily accomplished since, in a hot water system, the load energy generally passes through a single pipe in the form of hot water. Stated as an energy balance, Load Energy = Source Energy - Losses.

Due to the limits of the LCS, only an approximation of the transient heat losses can be utilized. Several publications are available that have evaluated the energy losses for residential water heaters[3,4,5] and detailed computer models are developed to calculate energy flows.[6,7] The computer model developed by Robert Hoskins of the Oak Ridge National Laboratory[8] was used to simulate water heater size, type and operating conditions for several studies, as shown in Table 5, and shows that the largest heat loss in an electric water heater is conduction through jacket walls (14% of energy input). An additional 5% is lost through the distribution pipe for a 7.6 meter (25 ft) long pipe. The remaining 81% of energy input is used to heat water. This computer model was validated by comparing results

with three other studies on the heat flow from residential water systems. Table 5 compares results with those of Mutch, Spann and Lee.

TABLE 5 COMPARISON OF ENERGY MODEL WITH RELATED STUDIES FOR ELECTRIC WATER HEATERS

	Unit 1[a] (10^9 J/YR)		Unit 2[b] (10^9 J/YR)		Unit 3[c] (10^9 J/YR)	
	MODEL	MUTCH	MODEL	SPANN	MODEL	LEE
USEFUL ENERGY	12.5	12.5	22.9	22.9	19.5	19.6
JACKET LOSS	3.4	3.4	3.8	6.0	4.3	4.1
% TOTAL INPUT	15.9	15.9	26.7	28.9	23.8	23.7
EFFICIENCY	78.8%	78.8%	85.8%	79.1%	82.0%	83.0%

[a] 0.19 m^3, hot water consumption 0.19 m^3/day, hot water temperature 60°C, inlet water temperature 12.8°C, ambient temperature 21.1°C, jacket insulation thickness 0.051 m.

[b] 0.19 m^3, hot water consumption 0.28 m^3/day, hot water temperature 65.6°C, inlet water temp. 12.8°C, ambient temperature 21.1°C, jacket insulation thickness 0.051 m.

[c] 0.20 m^3, hot water consumption 0.27 m^3/day, hot water temperature 62.8°C, inlet water temp. 15.6°C, ambient temperature 21.1°C, jacket insulation thickness 0.045 m.

DATA VALIDATION

The technique utilized by the monitoring program provides data that, in most cases, are cumulative over time and can be accurately recorded once a day. Only the temperature measurements have a potential weakness in that one must assume that the variations in outlet temperature (T_o) and inlet (T_i) are small during the one day sampling interval. Because the thermometers only indicate the temperature at the time of the reading, they cannot note transients which may have occurred in T_o and T_i during loads. On the other hand, T_i and T_o should be relatively constant. T_i in most cases, is controlled by local ground temperature and should not vary rapidly. T_o is essentially fixed by the tank thermostat and maintained by the auxilliary heating system. The precision with which T_o is typically maintained in residential domestic hot water systems is not well documented. If the variation in T_i and T_o between load and no load is small, then the less expensive monitoring system will be satisfactory and not require validation with a BTU meter.

Because the BTU meter cost is at least 100 percent greater than the LCS (low cost system), there is considerable merit in studying the accuracy of the LCS. Because the performance regulation of T_i and T_o is highly dependent on local site conditions and use patterns, laboratory tests alone will not suffice to determine the applicability of the LCS. Distance of inlet pipes from the surface, length of run from the main, condition of the element, rate of draw and rate of solar input are all potential local variables which can influence individual accuracy.

Because the LCS is attractive in cost, other groups have been and are using this method. The LCS needs to be evaluated; if its accuracy can be related to ite data, such as family size, water usage, etc., hen curves which project variations in accuracy as a function of the easily acquired site variables can be used for correction of data.

Because of potential inaccuracies in the LCS approach, a provision has been made on each monitoring system for accomodating a BTU meter. As data from sites using the LCS are analyzed and show discrepancies, BTU meters will be installed to quantify the error. Generally, the data analyses will consist of:

1. Correlations of accuracy with site variables. These analyses will be used to determine new sites for the BTU meters.
2. Comparisons of BTU meter data and LCS data on a monthly and annual basis.

It is expected that over 80 of the 200 monitoring sites will be checked with BTU meters. Since selection of the sites is important, the following sequence (Figure 2) is being developed.

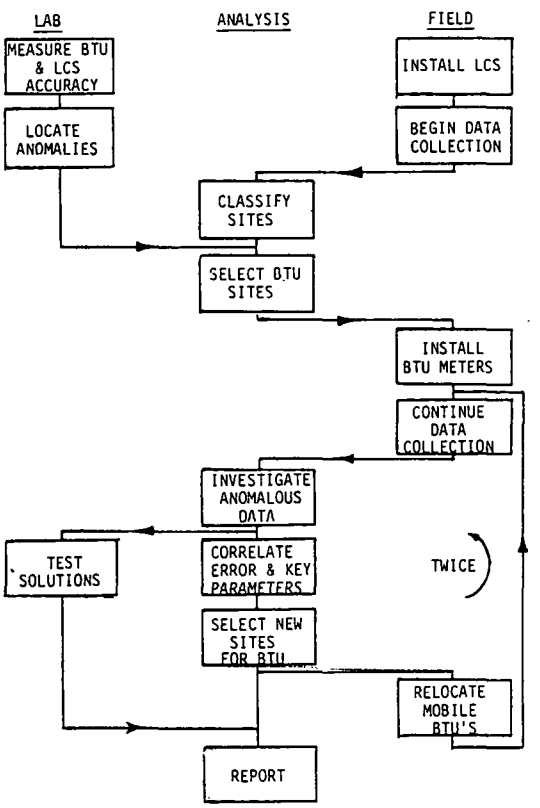


Fig. 2 Data Validation

SUMMARY

As discussed in this paper, a need exists for determining the performance of domestic solar water heaters. Interim results have revealed that systems are performing over a wide range of solar contributions. The solar industry, solar-oriented government agencies and the general public will need to know why these performances vary so much and, through continued monitoring, accurate answers to these questions will be obtained.

Before any final performance results can be determined, more information needs to be obtained about system heat losses, monitoring equipment accuracy and system operation. Data are being collected concerning these issues and, within a year, some hard facts will be available. In the interim, it can be said that the systems are performing as expected with variations occurring as a result of deviations in site and usage parameters.

NOMENCLATURE

MO/YR	Month and year of performance
GAL/DAY CONSUMED	# gallons of hot water used daily
OPERATION TIME/DAY	# hours the solar system operates daily
(T_o-T_i)	Temperature difference between inlet and outlet water (°F)
QWAT	Total KWHRs used daily to heat water $[(T_o-T_i) \times {}^{GAL/DAY}_{CONSUMED} \times .0024]$
QTOT	Total KWHRs used daily (QWAT +QLOSS)
QLOSS	KWHRs loss from tank and pipes
QPUR	KWHRs purchased daily
QSAV	KWHRs saved by solar water heating system daily (QTOT - QPUR)
%SOL	Average percent of hot water supplied by solar (QSAV/QTOT)

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PRELIMINARY RESULTS
SOLAR DOMESTIC HOT WATER MONITORING
IN PENNSYLVANIA

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ABSTRACT

The Pennsylvania State University, acting for the Northeast Solar Energy Center, is aiding in a monitoring program for fifty solar domestic hot water systems in Pennsylvania. Each of the monitored sites was the recipient of a $400 HUD grant as part of HUD's Solar Hot Water Initiative Project.

The monitoring program utilizes relatively low cost instrumentation, which provides information which is collected by each individual homeowner. This data is collected by PSU and the performance of each system is determined using computers at PSU. Feedback in the form of a summary of the results of the computer analysis is provided to each homeowner approximately every four months.

Performance results for about twenty of the sites have been obtained for time periods ranging from three to ten months. The average solar contribution toward the energy required for the domestic hot water is approximately 34% for all the sites evaluated to date. The best ten sites are averaging approximately 44%.

INTRODUCTION

In the early planning stages of HUD's Solar Domestic Hot Water Initiative Grant Program, it was the general opinion of the participating State Representatives that some type of performance evaluation should be performed on a sample of the homeowner sites awarded the $400 grant. Out of this concern the Northeast Solar Energy Center (NESEC) developed a monitoring program for 200 sites in the northeastern part of the country covered by HUD's program. Pennsylvania was to have 50 sites monitored as part of the total monitoring project. At the time this paper is being written (Mid September, 1979), 48 sites have been selected, with equipment installed at 37 sites, and three or more months of data evaluated for 20 sites. The Pennsylvania Governor's Energy Council (GEC) has approved the $400 grant for approximately 1800 homeowners. If 80% of these homeowners actually install a solar hot water system, then approximately 3 1/2% of the sites will be monitored.

Since the Pennsylvania State University (PSU) at apitol Campus (Middletown, Pa.) aided the GEC in dministering the HUD Grant Program, NESEC awarded PSU a contract to administer the Monitoring Project in Pennsylvania. The first monitoring sites in Pennsylvania were selected in October of 1978, and several sites collected data as early as November, 1978. Site selection, equipment installation, data collection and data evaluation have continued into the fall of 1979.

MONITORING EQUIPMENT

The equipment selected for the Monitoring Project by NESEC has been documented previously [1] [2], and will be touched on here only briefly. This relatively low-cost data acquisition instrumentation consists basically of:

1. Water Meter (digital read-out)
2. Two Thermometers (dial type)
3. Electric KW-HR Meter (digital read-out)
4. Elapsed Time Meter (digital read-out)

The Pennsylvania homeowner selected to participate in the Monitoring Project is provided a package of instrumentation (including necessary plumbing fittings, mounting hardware, and installation instructions), normally prior to the time of the

solar system's installation. When the solar hot water system is installed, the monitoring equipment is also installed. In some cases the monitoring instrumentation has been retrofitted to the previously installed solar system.

The instrumentation has been designed to monitor the amount of hot water consumed (water meter); the temperatures of the water as it enters and leaves the water heating system (two thermometers); the amount of electrical energy purchased by the homeowner to provide the hot water (electric meter); and the amount of time the solar system's pump or fan operates (elapsed time meter). A simplified schematic of the placement of the instrumentation on a typical system is shown in Fig. 1.

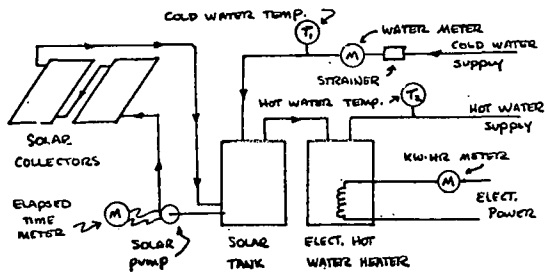


Fig. 1 Monitoring Equipment Placement

DATA COLLECTION

Each individual homeowner is requested to take readings from the monitoring instruments at least once each day. In addition to the meter and thermometer readings, the homeowner is requested to indicate the time the readings are recorded, the general weather conditions, and the number of people being serviced by the hot water system that day. The ambient temperature in the area of solar tanks is normally recorded two or three times each month. The data sheets are then either picked up by PSU students or mailed to PSU.

The problems associated with the data collection process are:

1. Inaccurate temperature readings (instrument calibration)
2. Mis-recorded readings (numbers reversed)
3. Missing daily readings (vacations/forgetting)
4. Illegible writing

In almost all cases these problems are not a major obstacle in evaluating the system's performance, since adjustments and corrections are easily made once PSU and the homeowner are aware of the problem.

DATA EVALUATION

One of the major objectives of the Monitoring Project is to determine system performance as a function of the percentage of the solar energy contribution toward the total hot water energy requirements. This percentage is determined using:

$$\% \text{ Solar} = \frac{\text{Total Energy Used - Total Purchased Energy}}{\text{Total Energy Used}} = \frac{Q_T - Q_P}{Q_T}$$

Where:

Q_T = Energy required to heat hot water + Estimated energy losses from heater tank = $Q_W + Q_L$

Q_P = Purchased energy = Elect. energy from KW-HR meter + Calculated solar pump energy = $\left(\frac{\text{KW-HR}}{\text{day}}\right)_{\text{elect. meter}} + \left(\text{Solar pump KW}\right)\left(t\right)$

t = Hours that solar pump or fan operated (elapsed time meter) each day

Q_W = Energy required to heat the domestic hot water supply = $\dot{m}\, c_p \Delta t$

$\dot{m}$ = Mass flow rate of water; (gallons/day) x (8.33 lbm/gal)

c_p = Specific heat for water (1.0 BTU/lbm-°R)

t = Hot water supply temp. - Cold water supply temp. = $(T_2 - T_1)$

Q_L = Estimated energy losses from hot water heating tank (not the solar storage tank)

The value of Q_L is estimated from a set of curves which plot energy loss vs. ambient temperature for each nominal tank size with various thicknesses of insulation on the tank. An example is shown in Fig. 2.

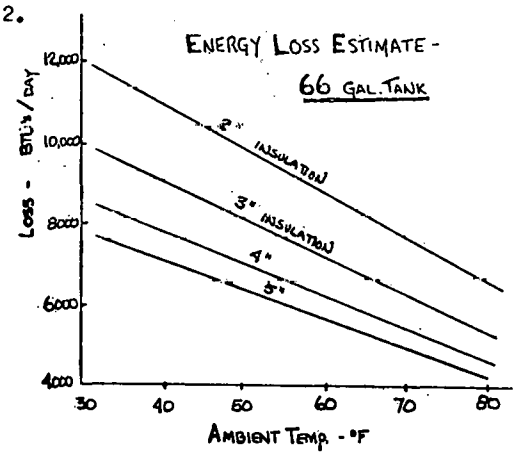


Fig. 2 Tank Loss Curves

A set of computer programs have been written which take the daily data and provide an analysis of the data on a daily basis, with monthly totals and averages. A summary of the monthly averages, with overall averages, is also available when desired. Three examples of computer print-outs showing the format for the computer data processing are provided. Figure 3 (Monthly Summaries) gives a summary of the results for a given site. Figures 4 and 5 (shown at the end of the manuscript) show the data and analysis for a single month.

Note that the monthly analysis (Fig. 4) also provides the solar energy utilized per area of collector surface (BTU/Ft^2), and the energy purchased per quantity of hot water consumed (KW-HR/gal).

NAME: ROBERT XXXXXXX
ADDRESS: RD#1 LAUREL DR,
MIDDLETOWN PA 17057

PHONE-(HOME): (717) 944-4169
(WORK): (717) 783-6097

NESEC NO.: PA 01 GEC NO.: H 22 023

SYSTEM MFG.: GRUMMAN
MODEL NO.: 60 FS

2 TANK SYSTEM: 80 GAL. SOLAR TANK
80 GAL. HEATER TANK

SYSTEM COST: $ XXXX FOR EQUIPMENT
$ XXX FOR INSTALLATION
$ XXX TOTAL COST

KILOWATT RATING: 4.5 KW.
COLLECTOR AREA: 48 SQ. FT.
PUMP/FAN: 0.1000 HP.
AVG. ESTIMATED TANK LOSS: 6350.0 BTUS PER DAY

NORMAL OCCUPANCY: 3 ADULTS
0 CHILDREN

GALLONS: GAL. OF HOT WATER USED DAILY
TIME: HOURS THE SOLAR PUMP RUNS DAILY
TEMP: TEMP. DIFFERENCE IN DEGREES F
QWAT: BTUS USED TO HEAT WATER DAILY

QTOT: TOTAL BTUS USED DAILY
QPUR: BTUS PURCHASED DAILY
QSAV: BTUS SAVED BY SOLAR DAILY
PSOL: AVERAGE PERCENT OF HOT WATER FROM SOLAR
PSUN: AVERAGE PERCENT SUNSHINE-FROM WEATHER BUREAU

MONTH	YEAR	GALLONS	TIME	TEMP	QWAT	QTOT	QPUR	QSAV	PSOL	PSUN
11	1978	73.7	4.4	86.4	53102.6	59352.5	46513.0	12839.6	21.6	46.4
12	1978	64.4	4.2	87.7	46646.3	53096.3	44438.3	8658.0	16.3	48.6
1	1979	59.9	2.3	91.5	45959.0	52758.9	47716.0	5043.1	9.6	36.4
2	1979	52.0	2.7	88.6	38216.7	45216.6	43960.2	1256.4	2.8	54.5
3	1979	67.5	6.5	71.0	40434.9	47134.8	41852.4	5282.5	11.2	65.3
4	1979	57.5	6.6	82.8	39522.4	45922.3	35816.3	10106.1	22.0	55.5
5	1979	74.7	8.5	67.9	42448.0	48448.0	40917.4	7530.7	15.5	57.8
6	1979	107.5	10.3	77.4	69282.4	75082.4	48922.7	26159.8	34.8	63.6
7	1979	56.0	8.6	73.1	34269.9	39969.8	24434.9	15535.0	38.9	55.7
8	1979	56.3	18.8	59.1	28377.7	34077.7	26699.2	7378.5	21.7	52.2
MONTHLY AVE=		67.0	7.3	78.4	43781.3	50055.4	40053.5	10002.0	20.0	55.6

NOTE: INDUSTRY TEST STANDARDS NOT USED FOR THE NORTHEAST STATES SOLAR WATER HEATER MONITORING PROGRAM.

Fig. 3 Monthly Summary

PRELIMINARY RESULTS

An attempt has been made to summarize the more significant results obtained to date for the twenty sites for which the data has been processed. Table 1 shows this summary. Note that the % solar (column 6) values range from a low of 4.4% (site No. 06) to a high of 64.3% (site No. 26), with the average being approximately 34%. Site No. 06 operated without pipe insulation on fairly long pipe runs, and actually had a negative % solar contribution for two months.

Column 7 gives the Annual % solar as predicted by a f-chart analysis for the system in question and a family of four using approximately 70 gal. of hot water per day. Note that in most cases the % solar to date is quite a bit less than the predicted value. Note also that this comparison is not quite fair at this time since a full year's data has not been analyzed for any of the sites, and also the percentage of sunshine for the time evaluated to date has been below normal. For the total ten months studied, the average percent sunshine in the Harrisburg, Pa. area was 53.9%, while the normal average for the same period is 56.5%. However, for the last two months (July and August), the actual percent sunshine was 13.5% below normal (54% vs. 67.5%).

Column 8 lists the total solar energy utilized by the homeowner's hot water system per unit area of collector surface. Estimates (theoretical) for central and southeastern Pennsylvania (where the majority of the 20 sites are located) indicates an annual contribution of from approximately 184,000 to 201,000 BTU's per square foot of collector for solar water heating [3]. The values listed in Column 8 are for three to ten months only, but it appears that they will not approach the theoretical

Site No.	Dates	No. Months of Results	Manufacturer of System	Collector Area (Ft^2)	Avg. % Solar to Date	Annual f-chart % Solar	BTU/Ft^2 Solar Energy Utilized	Avg. Gallons/day Consumed	No. Occupants Adults/Children
[1]	[2]	[3]	[4]	[5]	[6]	[7]	[8]	[9]	[10]
01	11/78 - 8/79	10	Grumman	48	20.0	47	63,346	67.0	3/0
02	3/79 - 8/79	6	Suntap	71	44.4	54	24,452	23.2	3/0
04	11/78 - 8/79	10	Solaron	77	17.3	54	97,850	211.1	2/3
05	12/78 - 7/79	8	Grumman	48	22.9	47	64,392	91.8	3/0
06	1/79 - 7/79	7	Grumman	72	4.4	55	7,140	72.0	2/0
07	12/78 - 8/79	9	W. L. Jackson	45	45.9	54	94,244	39.5	2/1
08	12/78 - 8/79	9	Grumman	48	20.4	47	80,484	94.0	2/3
09	12/78 - 8/79	9	Sunearth	66	40.8	52	82,337	62.6	3/0
11	1/79 - 8/79	8	Grumman	48	21.3	47	48,984	68.3	2/0
14	2/79 - 8/79	7	Revere	56	39.5	63	83,180	81.5	2/3
16	2/79 - 7/79	6	W. L. Jackson	45	34.4	54	46,625	39.2	2/2
17	1/79 - 7/79	7	Sunworks	82	48.6	62	59,720	59.3	2/3
18	1/79 - 7/79	7	Practical Solar	43	46.5	49	103,986	64.2	2/2
21	3/79 - 6/79	4	P P & G	57	27.9	64	37,030	79.8	2/2
25	5/79 - 7/79	3	State Ind.	79	39.8	51	21,063	68.5	4/0
26	2/79 - 8/79	7	Revere	56	64.3	63	147,410	85.3	2/3
27	2/79 - 8/79	7	Grumman	48	31.4	47	29,046	28.4	2/0
31	4/79 - 7/79	4	Revere	56	37.1	61	48,695	91.1	2/1
33	3/79 - 7/79	5	Sunworks	82	32.6	62	61,832	133.5	2/3
35	3/79 - 7/79	5	Practical Solar	64	35.6	62	31,097	34.9	2/3

Table 1. Preliminary Results Summary

values in most cases after a full year's data has been evaluated. Site No. 26, having the highest % solar and the highest BTU/Ft^2 listed is an exception.

Column 9 and Column 10 list the average water consumed per day (gallons/day) consumed and the average number of occupants being serviced by the hot water system respectively. The majority of the design information available indicates that a family of four should use approximately 70 gallons of hot water per day, or 17.5 gal/day per person [4] [5]. The results from Table 1 indicate an average hot water consumption of 20.7 gal/day per person; with the highest being Site No. 04 at 42.2 gal/day per person (two very small children in this family) and the lowest Site No. 02 at 7.7 gal/day per person (no explanation).

CONCLUSION

The Northeast Solar Energy Centers (NESEC) Monitoring Project currently being conducted in Pennsylvania is providing very valuable information about the performance of solar domestic hot water systems in Pennsylvania. The preliminary results reported here indicate that the systems are performing quite well, although not up to design expectations. The information collected, and the information to be collected for the remainder of 1979 and through 1980, will provide both homeowners and system designers valuable information on which to base their decisions concerning either installation of a solar domestic hot water system or the design and sizing of future solar hot water systems.

Information concerning the reliability of the various solar system components is being collected, and the response to problems which occur by the solar installers, dealers, and manufacturers will be evaluated. Also, depending on the total length of time the monitoring project is carried out, system performance as a function of time may be evaluated.

Comparisons of the various systems may also be made if a way can be found to normalize the many variables which affect system performance. Variables such as available solar insolation, collector orientation, collector tilt angles, water use patterns, the amounts of water consumed, and pipe and tank insulation, play an important part in the solar system's overall performance and should be taken into account in any comparisons.

The Pennsylvania State University is happy to aid NESEC in this worthwhile project and is looking forward to sharing the results with those intereste as the project continues into the 1980's.

PA 01 JUNE 1979

INITIAL VALUES: GALLONS= 14166.199
KW-HR= 3126.000
HOURS= 1094.100

INPUT DATA: PUMP/FAN= 0.1000 HORSEPOWER
TANK LOSS= 5000.0000 BTUS PER DAY
COLL AREA= 48.0000 SQ FT

DAY	TEMP IN	TEMP OUT	GALLONS	KW-HR	HOURS	PSUN	COMMENT
1	58.00	140.00	14309.20	3137.00	1105.80	55.00	
2	64.00	140.00	14362.60	3137.00	1116.70	68.00	
3	64.00	140.00	14525.20	3168.00	1126.70	42.00	C
4	59.00	140.00	14644.00	3181.00	1137.50	83.00	
5	59.00	140.00	14762.60	3196.00	1149.50	75.00	C
6	66.00	140.00	14868.00	3211.00	1159.50	7.00	
7	59.00	140.00	14974.70	3226.00	1170.00	71.00	
8	67.00	140.00	15078.90	3244.00	1180.50	79.00	
9	68.00	140.00	15253.40	3267.00	1191.60	79.00	
10	60.00	140.00	15354.10	3283.00	1202.50	52.00	
11	68.00	140.00	15427.60	3294.00	1212.50	67.00	
12	65.00	140.00	15531.00	3313.00	1219.70	100.00	C
13	60.00	140.00	15674.40	3333.00	1229.50	100.00	
14	60.00	140.00	15775.90	3343.00	1241.00	100.00	
15	60.00	140.00	15920.50	3355.00	1250.50	100.00	
16	60.00	140.00	16088.50	3376.00	1262.80	67.00	
17	62.00	140.00	16178.40	3385.00	1273.00	33.00	
18	67.00	140.00	16325.80	3404.00	1290.10	33.00	
19	60.00	140.00	16398.30	3414.00	1294.50	100.00	
20	68.00	140.00	16492.70	3424.00	1305.50	83.00	
21	63.00	140.00	16592.50	3435.00	1316.50	41.00	
22	63.00	140.00	16777.90	3458.00	1326.90	26.00	
23	60.00	140.00	16866.00	3472.00	1335.70	84.00	
24	65.00	140.00	16929.80	3477.00	1345.60	28.00	
25	61.00	140.00	17005.00	3491.00	1353.90	100.00	
26	66.00	140.00	17097.60	3501.00	1364.10	100.00	
27	58.00	140.00	17172.70	3507.00	1373.70	100.00	
28	65.00	140.00	17233.40	3514.00	1384.50	35.00	
29	66.00	140.00	17319.10	3522.00	1392.00	1.00	
30	60.00	140.00	17409.80	3533.00	1402.80	0.0	

Fig. 4. Data

PA-01 JUNE 1979

DAY	DGAL	DTIME	DTEMP	QW	QT	QP	QSAV	PSOL	PSUN	COMM
	GALLONS	HOURS	DEG F	BTUS	BTUS	BTUS	BTUS	PERCENT	PERCENT	
1	123.0	11.7	82.0	84016.4	89816.4	40521.9	49294.5	54.9	55.0	
2	53.4	10.9	76.0	33808.0	39608.0	2775.2	36832.7	93.0	68.0	
3	162.6	10.0	76.0	102937.3	108737.3	108449.0	288.3	0.4	42.0	C
4	118.8	10.8	81.0	80158.4	85958.4	47118.8	38839.6	45.2	83.0	
5	118.6	12.0	81.0	80024.0	85824.0	54250.3	31573.7	36.8	75.0	C
6	101.4	10.0	74.0	62504.0	68304.0	53741.1	14562.9	21.3	7.0	
7	110.7	10.5	81.0	74692.1	80492.1	53868.4	26623.7	33.1	71.0	
8	104.2	10.5	73.0	63362.5	69162.5	64107.4	5055.1	7.3	79.0	
9	174.5	11.1	72.0	104658.1	110458.1	81325.1	29132.9	26.4	79.0	
10	100.7	10.9	80.0	67108.5	72908.5	57383.2	15525.3	21.3	52.0	
11	73.5	10.0	72.0	44082.4	49802.4	40089.1	9793.3	19.6	67.0	
12	103.4	7.2	75.0	64598.2	70398.1	66680.1	3718.0	5.3	100.0	C
13	143.4	9.8	80.0	95560.7	101360.7	70755.1	30605.6	30.2	100.0	
14	101.5	11.5	80.0	67639.6	73439.6	37058.0	36381.6	49.5	100.0	
15	144.4	9.5	80.0	96229.7	102029.7	43374.8	58654.9	57.5	100.0	
16	168.2	12.3	80.0	112087.9	117887.9	74804.6	43083.3	36.5	67.0	
17	89.9	10.2	78.0	58810.6	64210.6	33314.0	30896.6	48.1	33.0	
18	147.4	17.1	73.0	89633.9	95433.9	69200.8	26233.1	27.5	33.0	
19	72.5	4.4	80.0	48314.0	54114.0	35250.3	18863.7	34.9	100.0	
20	94.4	11.0	72.0	56616.4	62416.4	36930.7	25485.7	40.8	83.0	
21	99.8	11.0	77.0	64013.2	69813.2	40343.7	29469.5	42.2	41.0	
22	165.4	10.4	77.0	110916.4	124716.4	81146.8	43569.6	34.9	26.0	
23	88.1	8.6	80.0	58710.9	64510.9	50022.6	14488.3	22.5	84.0	
24	63.8	9.9	75.0	39859.5	45659.5	19585.7	26073.9	57.1	28.0	
25	73.2	8.3	79.0	48170.2	53970.2	49895.2	4075.0	7.6	100.0	
26	94.8	10.2	76.0	60016.5	65816.4	36727.1	29089.4	44.2	100.0	
27	74.9	9.6	82.0	51160.1	56960.1	22922.2	34037.9	59.8	100.0	
28	60.7	10.8	75.0	37921.8	43721.8	26640.8	17081.0	39.1	35.0	
29	85.7	7.5	74.0	52829.1	58629.1	29213.6	29415.5	50.2	1.0	
30	90.7	10.8	80.0	60402.0	66241.9	40292.8	25949.1	39.2	0.0	
TOT=	3223.6	306.7	2321.0	2078473.0	2252472.0	1467682.0	784792.9	34.8	1909.0	
AVG=	107.5	10.3	77.4	69282.4	75082.4	48922.7	26159.8	34.8	63.6	

BTU SOLAR/SQ FT COLLECTION= 16349.9
GALLONS HOT WATER/KW HR= 7.50

Fig. 5. Monthly Analysis

REFERENCES:

[1] Northeast States' Solar Water Heater Performance Monitoring - Installation Guidelines (1968).
Prepared by: Northeast Solar Energy Center (NESEC)
70 Memorial Drive
Cambridge, Mass. 01242
(617) 661-3500

[2] Paul Erhartic (NESEC), Interim Performance Results From The Northeast States Water Heater Performance Monitoring Program (1979).

[3] INTRODUCTION TO SOLAR ENERGY IN PENNSYLVANIA; (page 31); (1969)
Prepared by: The Governor's Energy Council
1625 North Front St.
Harrisburg, Pa. 17102
(717) 783-8610

[3] HUD MINIMUM PROPERTY STANDARDS - 1973 Ed.
ONE AND TWO FAMILY DWELLINGS
(Tables 6-15.2 and 6-15.3; pages 6-15-17 and 6-15-18).

[4] ASHRAE HANDBOOK & PRODUCT DIRECTORY - 1976 SYSTEMS VOLUME, Ch. 16.

PERFORMANCE DATA ON SOLAR-ASSISTED HOT-WATER SYSTEMS IN FIVE TUCSON RESIDENCES

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The University of Arizona, Tucson

ABSTRACT

The performance of five residential solar hot-water systems has been monitored since early January 1979. Two conventional systems have also been studied for comparative purposes. All homes are in the same housing development and are occupied by private individuals. Hot-water and electric-power consumption are measured at weekly intervals. From an analysis of the data, the fraction of required energy supplied by the insolation is determined. Solar energy incident on the collectors is calculated using measurements made locally for a horizontal surface. The fraction of energy supplied by solar was found to vary from approximately 60% in January to 97% in July, although one house had values above 90% for the entire period. Representative system efficiencies were calculated wherever possible. This varied greatly over the weeks and months and was found to be strongly dependent on the family size and water-use patterns.

INTRODUCTION

The domestic hot-water systems of five single-family dwellings have been chosen for this evaluation. These are all located within the same housing development, and the solar units were put into operation at approximately the same time. This was during the fall of 1978. Monitoring of the systems began in early January 1979, and data has been collected through August of this year.

Of the five homes, two have solar space-heating systems as well as domestic hot-water. Both are of the unmixed type, and only results for the hot-water components are reported here. The three remaining houses each have basically the same mixed (open) system configurations but different manufacturers and solar installers.

The housing development, known as the Tucson Inner City Parade of Homes, is the result of a cooperative effort between the City of Tucson, the Southern Arizona Homebuilders' Association, and the University of Arizona. It was agreed at the outset that the study would not include performance ratings of individual collectors, from which a relative ranking could be determined. Rather, the purpose of the study is to determine the extent to which conventional energy in Southern Arizona can be replaced by solar energy for hot-water production. In all cases, electricity is the conventional energy source.

The test houses are all occupied by private individuals living under uncontrolled conditions. The family sizes range from one to four individuals.

DESCRIPTION OF SOLAR SYSTEMS

The solar systems are characterized in Table 1. All have flat-plate collectors with single glazing.

The systems with space heating are shown schematically in Figs. 1 and 2. System-K (Fig. 1) uses a mixture of water and propylene glycol circulated between a storage reservoir and the collectors. The basic space heating (not evaluated here) is provided by a solar-assisted heat pump in which outside ambient air is preheated by the glycol-water mixture before entering the evaporator unit. The domestic water is heated in a single-walled heat exchanger located inside the storage tank for the collector fluid. This tank has a capacity of 1018 liters (269 gals) and is located above ground in an enclosed storage closet. There is a blanket of fiber glass insulation (3.8 cm thick) surrounding the tank.

System-A is an air system and is shown in Fig. 2. There is a pebble-bed heat-storage bin located below ground. The bin is filled with 5 cm rocks and has a volume of 3.83 m^3. Domestic water from the 310 liter (82 gals) storage tank is preheated in a heat exchanger located in the air-handling unit. The main water-heater tank houses the auxiliary electric heating element.

The three remaining hot-water systems (Systems -H, -P, and -R) have the same basic configuration, as shown in Fig. 3. The main differences are in the size of the collectors and water tanks, the pump-

---

[1] Professor and Technical Director, Solar Energy Research Facility

[2] Graduate Assistant

TABLE 1

PHYSICAL CHARACTERISTICS OF SOLAR SYSTEMS

	Space Heating and Hot Water		Hot Water Only		
			System		
	K	A	H	P	R
Collector Area					
m^2	7.46	12.78	1.47	2.73	2.61
(ft^2)	(80.3)	(137.5)	(15.8)	(29.4)	(28.1)
Collector Tilt Angle					
degrees	49	45	49	30	35
Hot Water Tank Capacity					
liters	151	500	151	250	197
(gallons)	(40)	(132)	(40)	(66)	(52)

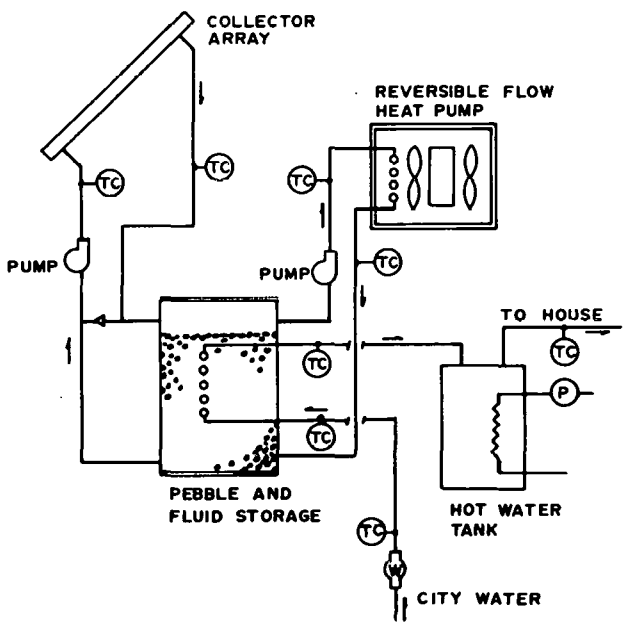


Fig. 1. Schematic of Propylene Glycol System-K

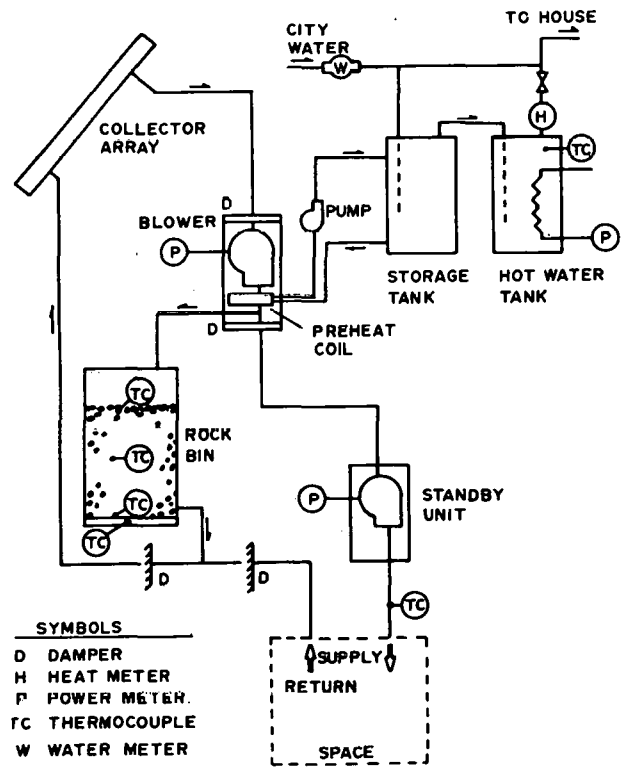


Fig. 2. Schematic of Air System-A

motor control units and the type of glazing on the collectors.

All five systems have similar kilowatt-hour meters to measure the power consumption of the auxiliary heating elements. The hot water consumption in gallons is measured by totalizing meters. System-A also has power meters connected to the blowers in the air handling units.

The Systems -K and -A have additional thermocouple instrumentation, as shown in Figs. 1 and 2. System-A also has a totalizing heat meter installed in the hot-water line leading to the house and located just upstream of the mixing valve.

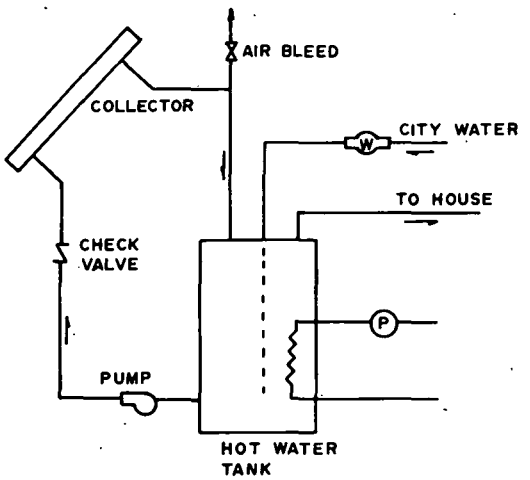


Fig. 3. Schematic of Mixed Water Systems -H, -P, and -R

INSTRUMENTATION ACCURACY AND MEASUREMENT PROCEDURES

The accuracy of the various instruments, as determined from manufacturers' specifications, is summarized in Table 2. All of the totalizing meters (electric, water, and heat) are recorded at weekly intervals. The temperature measurements, on the other hand, are made at hourly intervals and recorded on magnetic tape by a microcomputer-based data acquisition system. There have been only brief intervals during the past nine months in which temperature measurements could not be made because of equipment malfunctions.

TABLE 2 - NOMINAL METER ACCURACY

Meter Type	Unit Resolution	Accuracy
Electric	10 Kilowatt-hr	±3%
Flow	.1 Gallon	±1.5%
Heat	1000 Btu	±1.5%
Temperature	.9°F	±2°F

Hourly and daily total insolation on a horizontal surface is measured at the University of Arizona, Department of Atmospheric Sciences, using a PSP Eppley pyranometer. Normal incident direct-beam insolation is also measured. These measurements are made approximately 3 miles distance from the site of the housing development. Because of generally prevailing clear-sky conditions, this data is expected to be a good representation of that existing at the actual collectors.

EARLY OPERATIONAL PROBLEMS

Only relatively minor problems with the operating components have been uncovered. System-H had early problems with the circulating-water pump control. The pump would not activate automatically, even on warm sunny days. It was necessary to activate it first with the switch in the manual position.

The problem was found to be the poor location of the differential sensors. Although originally installed according to manufacturer's instructions, the collector sensor was later moved from an exterior pipe to the collector absorber plate inside the collector box. The tank sensor was also relocated from the water supply line to the collector to a position inside the jacket insulation of the tank. Thereafter, the system functioned properly in the automatic mode. The collector box originally had double glazing, the inside pane of which fractured almost immediately after installation. A replacement pane also broke. This was attributed to the thermal stresses in the non-tempered glass. The second pane of glass has since been removed, and no further breaks have occurred.

The air collectors of System-A have become partially clouded by a white deposit of unknown composition between the two panes of glass. The double glazing is supposed to be composed of single sealed units, but moisture condensate is clearly visible between the glass. There is a marked deterioration of the transmittance of the glass, which has not been replaced. Also, a motorized damper was found to be malfunctioning, but this was quickly corrected. This caused only a slightly adverse effect on the space heating system.

RESULTS AND DISCUSSION

The performance parameter of most interest to the present study is the fraction of energy required by the hot water systems which is met by the solar energy input. In this case, the energy required is the useful energy delivered to the hot water (called Q_{load}), plus losses in the system. Note that all the losses are included, not just those from the water storage tanks. It is reasoned that the added piping, plus the solar collectors themselves, represent an extension of the hot water system. Thus the losses associated with them should be included as part of the thermal energy requirements.

The total energy input includes the solar energy incident on the collectors plus any auxiliary energy needed for back-up heating and the operation of sizeable pumps and blowers. From an energy balance performed on the entire system, one has

$$Q_{solar} + Q_{aux} = Q_{load} + Q_{losses} \quad (1)$$

where

$$Q_{load} = [M C_p (T_{hot} - T_{cold})]_{water} \quad (2)$$

The time interval over which the energy balance is performed is one week. Thus, Q_{aux} and M are the electrical energy and total water flow, respectively, as determined by the totalizing meters. In applying Eqn. (2), it is assumed that over the interval of one week the temperatures of the hot and cold water streams are constant. The hot water temperature is taken to be that maintained in the tank by thermostatic control, and the cold water temperature is that of the city water supply, as determined by direct measurement or reasonable estimate. The validity of these assumptions will be discussed later.

The solar fraction used in this study is defined as follows:

$$\text{Solar Fraction} = \frac{Q_{solar}}{Q_{load} + Q_{losses}} . \qquad (3)$$

In view of Eqn. (1), one has

$$\text{Solar Fraction} = \frac{Q_{solar}}{Q_{solar} + Q_{aux}} . \qquad (4)$$

The auxiliary energy is obtained by direct measurement, and the solar energy input is deduced from local insolation measurements. This will be discussed subsequently. Before doing so, however, note that Eqn. (4) does not require knowledge of the hot water load and losses.

The solar energy incident on the plane of the collectors was inferred from measurements of total energy made on a horizontal surface. That is, the measured totals obtained at a remote site three miles distant from the homes were adjusted to account for the actual collector tilt angles. To accomplish this, it was necessary to separate the diffuse-sky and direct-normal components of insolation. For simplicity, the fraction of the total-horizontal insolation due to the sky was taken from the 1977 ASHRAE Handbook of Fundamentals [1]. This ratio is called the diffuse radiation factor and is tabulated for each month of the year.

For any given hour, the diffuse radiation actually striking the collector surfaces was estimated using the diffuse radiation factor multiplied by the angle factor between the collector and the unobstructed sky. Reflections from surrounding surfaces were ignored. The inferred direct component of radiation is for a surface aligned normal to the sun's rays. This was multiplied by the cosine of the angle between the normal to the collector and the earth-sun line. The two components were then added.

The diffuse radiation factor was assumed to be constant over the given month. The earth's declination angle, which was needed to find the sun's position relative to the collectors, was adjusted at five-day intervals. Insolation on the collectors was first determined in this manner for each hour of the day, from which daily and finally monthly totals were obtained.

In order to determine the importance of the diffuse radiation factor, sample calculations were performed using actual values obtained from direct-beam radiation measurements made at the same remote site used for the total insolation measurements. For the day of August 21 at solar noon, the measurements yielded a diffuse radiation factor of .028, whereas the value from [1] was .12. Holding these constant, the daily total insolation obtained using the measured factor was found to be 2% higher than that obtained using the handbook value. This difference was judged to be insignificant, and no adjustments to the calculated values were made.

The performance results are summarized in Table 3 for each of the solar systems. Averages for each month are given for the solar fraction and the thermal conversion efficiency of the systems. The determination of this latter quantity is discussed next.

TABLE 3 - SUMMARY OF RESULTS FOR SYSTEM PERFORMANCE PARAMETERS

	System-A		System-K		System-H		System-R		System-P	
Month	Solar Fraction	η_{TC}	Solar Fraction	η_{TC}	Solar Fraction	η_{TC}	Solar Fraction	η_{TC}	Solar Fraction	η_{TC}
Jan.	.79	.14	.93	.02	-	-	.65	.11	.57	.45
Feb.	.84	.11	.94	.02	-	-	.74	.11	.69	.38
Mar.	.87	.14	.94	.03	-	-	.77	.10	.71	.42
Apr.	.93	.12	.95	.04	.81	.09	.81	.16	.84	.37
May	.94	.12	.97	.04	.84	.08	.90	.08	.81	.53
June	.95	.10	.96	.07	.90	.07	.97	.03	.92	.42
July	.96	.10	.97	.08	.90	.08	1.00	.01	.95	.41
Aug.	.95	.11	.98	.02	-	-	-	-	.91	.45

The thermal conversion efficiency, η_{TC}, is definded in this study to be the fraction of the energy input which is delivered to the house by the hot water. Clearly, the volume of water used is important in determining this efficiency, and if no hot water is drawn by the house, the efficiency will be zero.

For this study, one has

$$\eta_{TC} = \frac{Q_{load}}{Q_{solar} + Q_{aux}} , \qquad (5)$$

where Q_{load} is given by Eqn. (2), Q_{solar} is the total energy incident on the collector, and Q_{aux} is the auxiliary energy supplied to the resistance heaters, blowers, and pumps.

The thermal conversion efficiency is tabulated in Table 3 for each of the five systems. It is evident that except for System-P, the values of η_{TC} are generally .10 or less. This is indicative of the fact that much of the energy is simply used to overcome losses, especially those from the hot water storage tanks. Excessively large solar energy input can also lead to low efficiencies, as in the case of System-A, which is an air system. The large collector area leads to a large amount of incident solar energy. Much of this collected energy is stored in the rock bin and eventually leaves the system as losses. Therefore, the thermal conversion efficiency can be expected to be low.

The thermal conversion efficiencies of two conventional electric hot-water systems were also determined for purposes of comparison. These are shown in Table 4. Identical measuring techniques were used in these two homes, and the results lie in the range .40 to .65, over the same time period. Only the results for solar System-P are reasonably close to these, lying in the range .37 to .53.

TADLE 4 - THERMAL CONVERSION EFFICIENCIES FOR TWO CONVENTIONAL SYSTEMS

Month	System-C η_{TC}	System-F η_{TC}
Jan.	.72	.57
Feb.	.67	.55
Mar.	.72	.34
Apr.	.68	.29
May	.72	.36
June	.62	.49
July	.67	.47
Aug.	.68	.48

In viewing these results, it is important to keep in mind that all of the solar energy incident on the collectors is counted as energy input, even when none is called for by the system. This is because the solar input cannot be switched on and off, nor can the collector area be varied. Nevertheless, it would be desirable to determine an efficiency which more nearly characterizes the performance of the solar collector systems alone, and which does not include losses from storage tanks. It was possible to determine such a solar system efficiency for only one of the systems, and this was for the period April through August.

For a two-week period in March, the circulating pump in the solar System-R was disconnected. The only energy input was through the electric auxiliary heater. Hot water use was minimal during this period because of only brief occupancy by the single owner. Thus, a good determination of the tank heat losses could be made. This amounted to 80 KW-hr per week. Using this figure, the system energy balance of Eqn. (1) was rewritten as follows:

$$Q_{solar} + Q_{aux} - (Q_{loss})_{system} = Q_{load} + (Q_{loss})_{tank} \qquad (6)$$

That is, the total losses were broken down into tank and system losses. A solar system efficiency can now be defined, which is given by

$$\eta_{SS} = \frac{Q_{solar} - (Q_{loss})_{system}}{Q_{solar}} . \qquad (7)$$

In the foregoing, the system losses include those from the collector. If there are no piping losses, η_{SS} is effectively the average efficiency of the solar collector.

The numerator of Eqn. (7) can now be determined from Eqn. (6) since all other quantities are known. The solar system efficiency, thus calculated for System-R, is given in Table 5. The 64% to 70% figure is indicative of a good collector system.

TABLE 5 - ESTIMATED SOLAR-SYSTEM EFFICIENCIES FOR SYSTEM-R

Month	η_{SS}
Apr.	.64
May	.70
June	.65
July	.66
Aug.	.68

SOURCES OF ERROR IN PERFORMANCE RESULTS

The solar fraction is judged to be the most reliable and also the most useful performance indicator for this study. It is calculated using the directly measured quantity, Q_{aux}, plus a quantity derived from direct measurement, Q_{solar}. In determining this latter quantity, errors introduced by the use of tabulated values of the diffuse radiation factor in the calculations could amount to 2% of the daily total insolation. This would be carried through for the monthly totals as well. Thus, it is judged that the solar fraction has a probable uncertainty of ±2% overall.

The thermal conversion and solar-systems efficiencies have considerable uncertainty, primarily due to the uncertainty in Q_{load}, as obtained from Eqn. (2). This assumes that the hot and cold temperatures are constant over the time interval used to measure M, in this case one week.

The hot temperature was checked at intervals by a mercury-in-glass thermometer submerged in the hot water from the tap. This was reasonably constant and found to be close to the thermostatic setting on the tanks. The variation was ±1.1°C. This is to be expected since some auxiliary energy was required each week to maintain the water at the desired temperatures.

It was found that the thermometer measurements did not correlate well with the thermocouple measurements made on Systems -A and -K. Unless water was actually being drawn when the thermocouples were read, a reliable reading could not be expected. This is because the thermocouples were mounted on the outside of the copper water lines (but under a layer of insulation) and not in the tanks. Late in the study, an immersion thermocouple was installed in the tank of System-A, and thus a more accurate measure of the hot water supply temperature is now being obtained, but only for this one system.

The cold water temperature of the city supply was even more difficult to obtain. The temperatures indicated by the one thermocouple mounted on the cold-water supply pipe for System-K were to be used for the entire sample of houses, and values were available at hourly intervals. However, unless water was flowing, a better measurement of the ambient temperature inside the storage closet was obtained than that of the city water. Thus, it was necessary to use fair estimates of this temperature in the calculations.

The cold water temperature was varied from 23.9°C (Jan., Feb., Mar.) to 26.7°C (April, May) to 29.4°C (June, July, Aug.). These are judged to be reasonably good averages, which could be slightly high. This would cause low estimates to be made of the η_{TC} and η_{SS}.

The temperature uncertainties can be removed if the heat delivered to load is measured directly with a heat meter, and one was installed in System-A in August. This meter is a totalizing device which integrates the product of the instantaneous flow rate and the instantaneous temperature differences between the hot-delivery and cold-supply water temperatures. In this case, the meter registers in 1000's of Btu's.

The load, as calculated from Eqn. (2) using measured hot and cold water temperatures, was compared to that measured by the heat meter. For a one-week period in September, the calculated load was 206 megajoules, and the measured load was 80 megajoules. Although at first alarming, the reason for the large discrepancy was soon discovered. The uninsulated "cold" water line in the heater closet was abnormally hot. Closet temperatures typically reached 49°C (120°F) during this week, and the "cold" water supply temperature under static conditions was measured to be 45°C. Thus the heat meter consistently sensed the incorrect supply temperature, which was measured inside the house to be 28°C.

From these tests, it can be concluded that the simple load calculation is often times more accurate than the direct measurement. However, proper installation of the cold water sensor should overcome this problem. Until further tests can be made, it is estimated that the load determination can have an uncertainty as high as 15%.

ACKNOWLEDGEMENTS

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DESIGN AND OPERATION OF SOLAR DHW SYSTEMS

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ABSTRACT

This paper discusses the design, installation and performance of three different types of solar domestic hot water systems. The first system is one which uses silicone oil as the heat transfer fluid. This was an early system which did not work well. Problem areas such as entrapped air, leaks, poor soldering techniques, stagnation, and the low specific heat of the silicone oil are all covered as well as the poor overall performance of the tube on shell type heat exchanger.

A readily available prepackaged dual wall tank system using a 50% Prestone II and 50% water solution is shown. Installation of additional insulation and the value of a water heater timer is discussed with emphasis on the savings of fossil fuel. Actual performance fulfilling the hot water needs of a family of three during a 30-day period is shown.

A unique, simplified drain back system where the drain back tank is used as an expansion tank as well as reservoir is shown; its performance fulfilling the needs of a family of four over a 30-day period is also shown.

INTRODUCTION

This paper will be presented in three parts. Part I will cover an 82 gallon system using silicone oil as the heat transfer fluid. Part II will be a 50% water/50% Ethelyene glycol system using a dual wall dual tank heat exchanger and in Part III a simplified drain back system using distilled water as the heat transfer fluid.

At the very beginning you should be warned that absolutely no government funds have been used in any of these installations and while we shall speak only of three systems, several similar installations have been made in the Montgomery, Alabama area. Perhaps the documentation of results may not be what you are used to seeing and that is directly related to no Federal subsidy for instrumentation. Nonetheless, we will provide what we consider to be the bottom line. "How did the system perform and what did the owners think about solar water heaters?"

Since the presentation will cover three different heat transfer fluids it might be well to identify some fairly important characteristics about these fluids.

From Table I below (1) it becomes clear that water is by far the best heat transfer fluid with the exception of freezing. It has far better heat capacity and a much

TABLE I TRANSFER FLUIDS

Medium	Specific Gravity	Viscosity[1/] Centipoise	Heat Capacity (BTU/lb.°F)	Freezing Point, °F
Water	1,000	0.5 to 0.9	1.00	+32
50 wt. % Water-ethylene glycol	1.05	1.2 to 4.4	.83	-33
50 wt. % Water-propylene glycol	1.02	1.4 to 7.0	.85	-28
Paraffinic Oils	.82	12 to 30	.51	+15
Aromatic Oils	.85	0.6 to 0.8	.45	-100
Silicone Oils	.94	10 to 20	.38	-120

1/ Because viscosity is sensitive to temperature, values are given for a temperature range of approximately 80°F to 140°F.

lower viscosity and thus lower pump energy requirements. You can see that Glycol/ water solutions provide good heat capacity and fairly low pumping requirements while insuring good freeze protection. On the other hand silicone oil probably provides the best anti-corrosion properties but from the table we see we have to pay a severe penalty in its low heat capacity of .38 and the high energy pumping requirements when compared to Glycol or water. However, in extremely high temperature systems such as concentrating collectors where high flash points are a necessity, it may very well be an excellent choice. Silicone oils are reported to be practically incombustible (2), certainly with a flash point above 600°F.

THE SILICONE OIL SYSTEM

The first system to be discussed is depicted in Fig. 1. It was designed and installed in 1973 at a time when prepackaged systems were simply not available. The collectors used were copper waterways with aluminum fins, housed in extruded aluminum enclosures with a high quality, water-white stipple finish tempered glass. We had a great deal of difficulty in purging air from the dual wall heat exchanger which helped deteriorate its already poor performance. This problem was largely overcome by completely draining and refilling the system, this time from the top. A device was fabricated which allowed us to fill through the down line, thus forcing the air through the heat exchanger and the up line, through the collectors until the oil reached the top of the collector manifold. Then a 50 PSI pressure relief valve was inserted in the opening thus sealing the closed loop system. About one month after the system

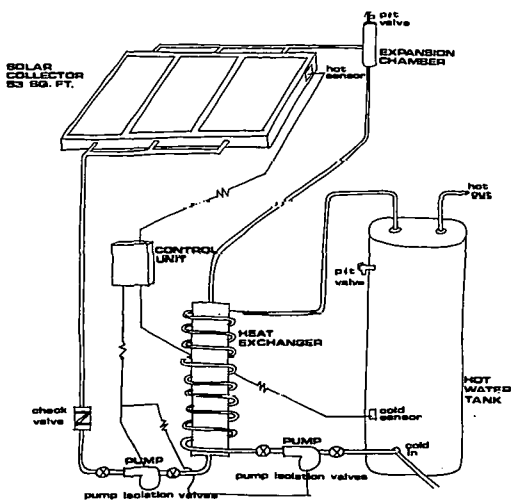


FIGURE 1: SILICONE OIL SYSTEM

had been in operation we experienced a 2-hour power outage. Temperatures went high enough so they were off scale on our basic electronic measuring equipment. In making the sweat connections for the manifold we had used 60/40 solder, a basic mistake. Needless to say, we lost several joints and had spillage of silicone oil on the roof. Since then all joints have been made using 95/5 solder and no further difficulty has been encountered. During installation we experienced leaks through threaded joints, even though a water pressure check of 40 PSI did not indicate any problems. Needless to say, if the joint is not absolutely perfect, silicone oil will find it. For that reason, sweat connections are recommended with threaded joints only where it cannot be done otherwise. We were finally able to correct our leaks by using a combination of Crane JE-30 thread sealer with teflon and teflon tape. Incidentally, anytime one is working with problems such as these, it is a good idea to keep collectors covered. We have made several such covers from painters drop cloths and they work fine. We find that in new construction, often we have to install collectors and then get held up with the remainder of the installation because others have not finished their work, i.e., plumbers, electricians and even sheet rock people. While the collectors have passed their stagnation test, it just doesn't make good sense to put them through this kind of torture unnecessarily.

Once we solved our problems the system was started again. The collector performance was outstanding. The closed loop consistently achieved silicone oil temperatures on the order of 170° to 190°F. However, the overall system was a dismal failure. Water temperature in excess of 120°F was never achieved, with the norm being about 100°F. This was attributed to the penalties of a dual wall, wrap around type heat exchanger and the low heat capacity of the silicone oil. The heat exchanger never exceeded 30% efficiency. The system has been dismantled and replaced with a water drain back type which will be discussed later in this paper.

THE DUAL WALL TANK SYSTEM

The next system to be looked at is a closed loop system using a 50% solution of water and Prestone II and a tank with an integral dual wall heat exchanger, Fig. 2. Essentially this is a tank within a tank. It is a prepackaged unit that has the control package mounted atop the tank. The manufacturer feels all the hard work is done but we submit to you that working on

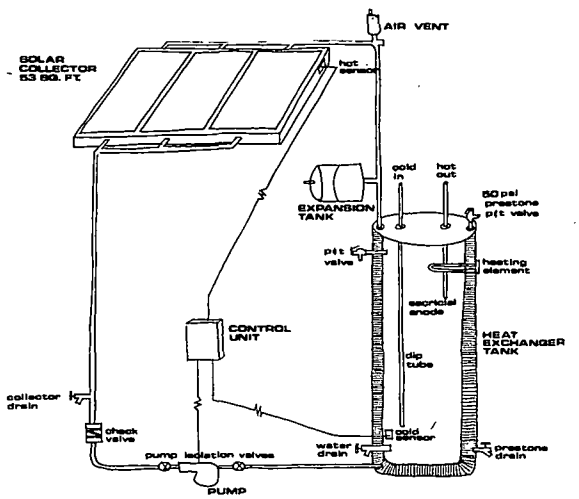


FIGURE 2: DUAL WALL TANK SYSTEM

steep roofs and crawling through oven-like insulated attics is also work. But we must admit that working with the pre-packaged units is really great with respect to time saving. Unless we run into something unforseen, the entire system can be installed in one day with the check-out the following day. The connections for the collector piping both in and out are flare fittings. Only the tempering valve and the hot and cold water lines require sweat fittings. Charging the system requires some care and procedures in addition to those specified by the manufacturer. We again found we had air trapped in the system. Some of the air would circulate, causing the pump to cavitate but even though we had a float type vent atop the collector array, it took several days before the system came clear. In each case water had to be added to bring the closed loop pressure up to 18 PSI as recommended by the manufacturer. Needless to say, that took time and money. Finally it was learned that we were actually putting the air into the system. This happened because after pressurizing and testing the system with water, we drained the water through the input water hose which filled with air as the tank emptied. Then after filling the heat exchanger tank with prestone from the top as specified we filled the system with water from the bottom—only the hose was full of air which went into the heat exchanger before the head of water reached the tank—thus, trapped air. We have since fashioned a device which allows us to bleed off the air in the hose and only then open the valve to the heat exchanger tank. A pressure gauge was also added so that when the system pressure reaches 18 PSI the flow of water can be cut off and the system is properly charged.

Very little instrumentation was used but it was sufficient to prove out the operation and tell us just how well or poorly the system was performing. The Central Alabama Electric Cooperative was kind enough to furnish us with a standard meter and we added a time elapsed meter in parallel with the pump. A pyrometer it isn't but it does provide us with an indication of actual hours of sunshine. An electronic thermometer was used to measure the temperature of the transfer fluid in and out of the heat exchanger as well as temperature at the top and bottom of the tank before startup and after shut-down. We also asked the lady of the house to log the hot water usage for a thirty-day period.

We do a number of other things to each system we install. Some are by choice in an effort to save more fossil fuel while others are done to meet local code requirements. Each tank is installed on a 4-inch high wooden platform and a 4-inch piece of rigid foam insulation in an effort to reduce the conduction losses from the uninsulated bottom of the tank to the concrete slab. In this area most homes are built on a slab. Then in order to meet the Montgomery code, the tank is set in a galvanized pan that is 4 inches deep and has a one-inch drain to the outside. The code also requires that lead flashings be used for all roof penetrations. They have found that some of the newer flexible types break down after 3 or 4 years, causing leaks and unhappy solar customers, which none of us needs.

A water heater timer has also been installed in the heating element circuit on each installation. We felt this made good sense in several ways. First of all it seemed ridiculous to heat the water electrically in the morning (following the usual heavy usage) knowing full well that the sun would soon be up allowing us to save that precious fossil fuel. Also, it gave a degree of control we believe we needed. Now the trippers are adjusted to allow the heating element to heat if need be, but only between 4 A.M. and 5 A.M., and 5 P.M. to 6 P.M., both of which are off peak periods for the power company. That tends to help them and at the same time help the customer by only allowing two hours per day of heating time during extended periods of inclement weather. Needless to say, it certainly helps eliminate the cycling of the heating element due to stand-by losses.

Only with these considerations in mind can the data we gathered be at all meaningful. This data is for the month of April which is a pretty average month. It is certainly better than December and January which are characterized by prolonged periods of

overcast skies and rainy weather. Yet it isn't as good as the summer months through about November where clear, bright skies are usually the case.

From the data in Table 2, one can see that on days where the system operated in excess of 5 hours, a 50°F temperature increase could be expected. It's also easy to spot days of inclement weather simply by looking at the kilowatts consumed. You can also see a pattern of living and adjustment. For instance, this lady did no laundry on rainy days. You will also note that after the 3rd day of April the heating element thermostat was cut back from 140° to 110°. This reduced the kilowatts used on non-sunny days from 10 on the 3rd to 7.5 on the 8th.

The conclusion reached from the data indicated a 30-day consumption of 71 KW at a cost of about $3.50 produced 38 loads of laundry, 39 loads of dishwashing, 71 showers and 21 baths. The bottom line of course is a satisfied customer. There were of course times when the hot water was completely depelted. We offered to increase the time window of the timer or to provide extra trippers but our offer was politely declincd.

THE DRAIN BACK SYSTEM

The drain back system consists of three collectors of the copper pipe aluminum fin type, an on-off controller, a 1/20 H.P. pump and a 65 gallon, stone lined tank with a 3/4" finned copper pipe heat exchanger in the bottom. The system is depicted in Fig. 3. As is always the case, we do our best to design the simplest possible system that will achieve the desired result.

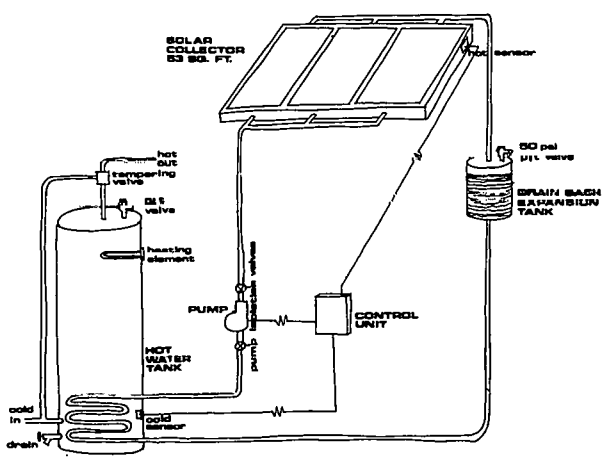


FIGURE 3: DRAIN BACK SYSTEM

This drain back system is simplicity in the greatest degree. You see no float type vents or vacuum breakers. No worries about air in the system either as it can flow directly to the drain back tank.

The heat transfer fluid is distilled water and in the event of power failure, it drains into the tank leaving the collectors empty. There is no worry about break-down during stagnation as is the case with a glycol system. The system is essentially closed loop in that it is sealed and pressurized to about 10 PSI. The drain down tank is not full and as a result it is also the expansion tank. The drain down tank is a 6-gallon glass-lined tank. It is usually located in the attic to reduce head pumping requirements at start up. It must, however, be located in an area protected from freezing or be given freeze protection such as a pipe wrap heater with a thermostat that cycles on at about 40°F. In most of the southern states, attics do just fine.

A sample of the data which was gathered from the drain back system is at Table 3.

ECONOMIC ANALYSIS

Each of the previously discussed systems was installed at an average cost of not more than $2000.00. Inflation pressures have since increased that cost to $2250.00. Readily available information indicates that nearly all two element water heaters serving a family of four people consume in excess of 600 kilowatts monthly.

Using these figures and being generously fair to the anti solar forces, one arrives at a payback period as indicated in Table 4. It goes without saying that as power prices escalate, the payoff period becomes shorter and shorter.

ACKNOWLEDGEMENTS

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TABLE 2:

April 1979

Date	Start Top	Temp Bottom	Stop Top	Temp Bottom	Hours Run	Electric Meter	Load			
1	105	96	151	140	7.2	.5	2L	3SH	1DW	1B
2	95	90	142	128	6.5	.5	1L	2SH	1DW	
3	98	82	140	80	0	11		2SH	1DW	(rainy)
7	113	105	150	142	7.1	13.5	2L	2SH	1DW	1B
8	96	80	110	80	0	21		3SH	1DW	(rainy)
16	96	90	147	127	7.5	41	2L	3SH	1DW	1B
17	96	85	150	131	7.8	41.5	2L	2SH	1DW	1B
28	98	76	115	110	5.2	61	2L	3SH	1DW	(partly cloudy)
29	99	76	110	76	0	70		3SH	1DW	(rainy)
30	96	76	156	135	8.1	71	4L	3SH	2DW	1B

30-Day Summary

Total Hours Run = 133

KW Used = 71

Load = 38 Loads Laundry, 71 Showers, 39 Loads Dishwasher, 21 Baths

Cost at 5¢/KW = $3.55

TABLE 3:

May 1979

Date	Start Top	Temp Bottom	Stop Top	Temp Bottom	Hours Run	Electric Meter	Load			
5	100	76	145	140	5	.4	2L	4SH	2DW	(partly cloudy)
6	110	78	173	165	8.1	1.0	1L	3SH	2DW	
7	120	75	168	159	8.1	1.6	3L	2SH	1DW	1B
11	110	75	120	115	1.3	4.6		2SH	1DW	(mostly cloudy)
12	110	75	110	75	0	10.8		3SH	2DW	(rainy)
13	96	75	175	165	8.2	11.4	4L	4SH	2DW	
14	122	100	175	165	8.1	11.8	2L	3SH	1DW	1B
29	95	75	110	75	0	27	1L	3SH	1DW	(rain)
30	100	75	178	170	8.4	27.5	3L	4SH	2DW	
31	110	78	178	170	8.8	28.1	2L	3SH	2DW	1B

30-Day Summary

Total Hours Run = 182
KW Used = 28
Load = 40 Loads Laundry, 92 Showers, 54 Loads Dishwasher, 16 Baths
Cost at 5¢/KW = $1.40

TABLE 4:

Cost Comparison/Family of Four

Average Power Consumption	Monthly	Annually
A. 2 Element Electric Water Heater	600 KW	7200 KW
B. 1 Element Solar Water Heater*	75	—900
Power Saved =		6300 KW
Dollars saved at present rates (.05) -		X .05
		$315.00

Cost of Solar Water Heater $2000.00
Federal Tax Credit 600.00

Actual Investment $1400.00 ÷ $315.00 = 4.4 year payoff

* Total actual kilowatts used during the 6-months period—April through September—was 202 including pump and controller power.

SYSTEM DESCRIPTION AND EARLY PERFORMANCE OF POOL AND DOMESTIC WATER HEATING INSTALLATION WILLMAR STATE HOSPITAL WILLMAR, MINNESOTA

PAUL F. CUMMINGS
STATE ARCHITECTURAL ENGINEER

ABSTRACT

The Willmar State Hospital, located in Willmar, Minnesota, August 1979 placed into service a new building, designed to provide physical and therapeutic treatment for mentally retarded, mentally ill, and chemically dependent patients. A solar collector system provides heating of swimming pool water, heating of domestic water, and tempering of ventilation air to the swimming pool area. The installation consists of approximately 3,000 square feet of flat-plate solar energy collectors, roof-mounted, which are designed to collect 54,000 MBTU during June and 22,000 MBTU during December. The primary heat exchange fluid is ethylene glycol and water which is pumped through heat exchangers for use or storage. This paper describes the instrumentation installed to measure performance of the solar system and the information collected in the initial operations.

The cost of the building, on which bids were received in July, 1977 was $2,000,000, including fees; and for the solar system, on which bids were received in June, 1978, was $205,000, including fees. The primary purpose of the solar installation is to develop operational and maintenance data applicable to other potential solar projects of the Minnesota state government.

INTRODUCTION

The Willmar State Hospital which is typical of nine operated by the Minnesota State Department of Public Welfare has a patient population of approximately 700. The activities building will be used predominantly by the mentally retarded patients which number about 300. The building provides space for training in arts, crafts, music and athletics. There is a gymnasium and 28,000 gallon swimming pool. The design domestic hot water load was 100 GPM with a 5° Farenheit temperature rise. The design pool water heating load was 150,000 BTUH. The building construction quality and decor is similar to that used in student activity centers for the Minnesota Community College System. To focus attention on the solar installation a recessed wall panel which displays a schematic of the system with dial thermometer indicators and pump running lights is mounted in a corridor near the main building entrance.

WEATHER

Willmar is located in Central Minnesota very slightly north of 45 degrees north latitude and experiences predominantly northwest winds. The average yearly total of degree days is approximately 8,000 with about 2,000 occurring during the month of January.

The average annual snowfall is approximately six feet which together with the wind from the prevailing northwest direction may create difficulties for a system operation through the snowfall season. The design temperature for the heating system of the building is -20° Farenheit although -25° Farenheit occurs a few times each winter.

SYSTEM DESCRIPTION

The solar collection panels are manufactured by Libby Owens Ford, model #121. These are single pane glass cover, extruded aluminum frame, coppertube fluid passageways units. There are 15 units per bank and 9 banks, roof mounted approximately 20 feet above grade. The lower collector header is 30 inches above the roofline. The total collector area is 3,000 square feet.

A schematic of the system is shown in Figure 1. A solution of 50 percent glycol and 50 percent water is circulated through the primary circuit by a 5 horsepower pump. Three way valves regulate the flow of glycol through the swimming pool heater, the domestic hot water heater, or the preheat coil for the swimming pool makeup air supply.

The first demand is for pool heating. As that demand is satisfied the domestic hot water heater places water in two 150 gallon tanks one for hot storage and the other for warm storage. If both pool heating and hot water storage are satisfied, glycol is circulated through the pre-heat coil for tempering ventialation air to the swimming pool area.

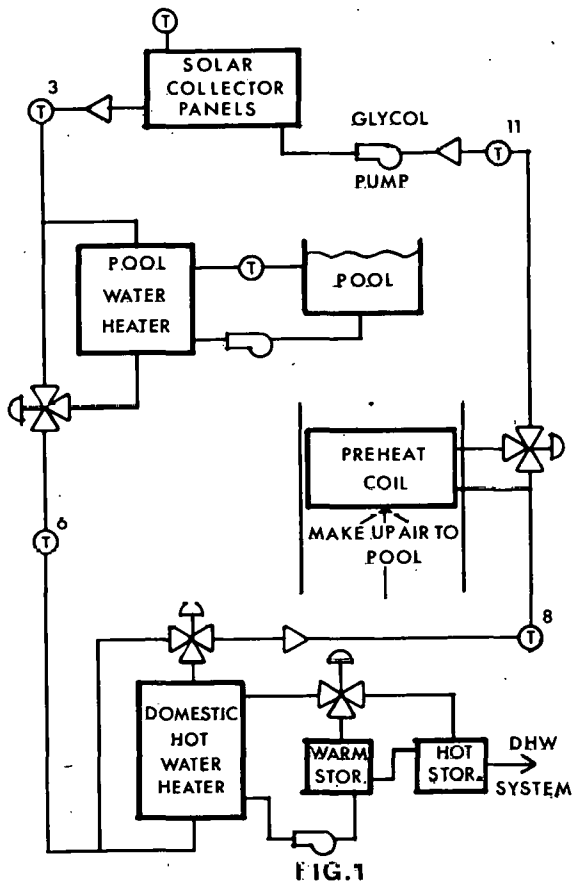


FIG.1

INITIAL OPERATIONS

A tabulation was kept each day beginning September 11 of the thermometers which describe temperatures of various points of the system. The primary interest was in number 3, temperature of glycol leaving the solar collection panels, number 6 glycol temperature leaving the swimming pool water heater, number 8 glycol temperature leaving the domestic hot water heater, and number 11 glycol temperature returning to the solar collector panels. The first day of the complete clear weather conditions was September 14 and the 1:00 p.m. readings are as shown on Table 1. Another day, of same weather conditions was September 21 and the 1:00 p.m. readings are as shown on Table 1.

TABLE 1

Thermometer Number	September 14 1:00 p.m.	September 21 1:00 p.m.
3	144° F	184° F
6	136° F	150° F
8	136° F	172° F
11	140° F	184° F

The September 14 reading indicates a temperature drop through the pool heater of eight degrees Farenheit about twice the theoretical (maximum) no drop or bypass through the domestic hot water heater and an increase of four degrees farenheit either through or bypass to make up air coil. The September 21 readings show a roughly similar situation of a drop through the pool heater but a gain in glycol temperature downstream of the domestic hot water heater and a gain in glycol temperature downstream of the make up air unit. The readings were not indicating actual performance.

Several possibilities were examined. Hospital staff maintenance men were instructed to take the readings each hour during the working day and while it is possible that some error occurred in the taking of the readings, it is more likely that the termometers were correctly read as there was and is high interest on the part of the maintenance staff in assisting in establishing the performance of this installation. The thermometers could be reading erratically but they had been calibrated and certified correct by the temperature control contractor. This left the probability of short circuiting or other malfunctions in the system operation.

ecause ethylene glycol is used as the primary luid a double wall separation is maintained between the glycol and the swimming pool water, or the domestic hot water. On the pool water heating circuit two heat exchanges are used. In the first heat is transferred from the glycol to circulating water which is circulated through a second pool heat exchanger which in turn heats the water transmitted to the pool. The water which acts as a transfer agent is circulated by pump between the two heat exchangers. In trying to establish a pattern of consistent temperature readings it was discovered that the intermediate water circulating pump had burned out so no effective heat could be transferred from the glycol to the pool water. This was typical of the early operational difficulties encountered with this installation.

DESIGN

For design purposes in sizing the primary heat exchangers the pool water heating load was assumed to be 150,000 BTU per hour. The domestic hot water load was 250,000 BTUH, and the make up air tempering demand at 200,000 BTUH. The solar collector system was estimated to be capable of delivering on an average a day 75,000 BTUH and possibly double this value under peak conditions. Of course, neither the pool demand, the hot water demand, or the make up air demand is likely to reach a maximum value at the same time. Since they are non concurrent in their requirements it was presumed that any one of these three loads could occur on such time as to make use of the available solar energy. Beyond immediate demands, the 3,000 gallon of storage connected to the domestic hot water system provides for utilization of any solar energy available which cannot be utilized directly by any of the three possibilities at any given moment.

OPERATIONAL RESULTS

(Due to the instrumentation difficulties mentioned above, valid operating data was not available prior to the dealine for the advance proceedings. The operational results will be published as an addendum to the proceedings.)

SYSTEM DESCRIPTION AND EARLY PERFORMANCE
OF
POOL AND DOMESTIC WATER HEATING INSTALLATION
WILLMAR STATE HOSPITAL
WILLMAR, MINNESOTA

ADDENDUM

PAUL F. CUMMINGS
STATE ARCHITECTURAL ENGINEER

OPERATIONAL RESULTS

On October 2, 1979 the designing engineers and the temperature control contractor calibrated the thermometers with the result that Number 11 was reading 11° F. high. During the balance of October until mid November the output of the collectors as measured by the flow rate of the glycol solution multiplied by the temperature difference of glycol to the collectors and from the collectors resulted in a collection of 3.5 million BTU on clear days with ambient temperatures of 45° F. to zero BTU on overcast days at outdoor temperatures of 30° F.

The steam supply to the pool heater has been shut off since early August until mid November except for a period of time for repair of the heat exchanger pump. Consequently practically all of the pool heating load has been carried by the solar system. However, pool heating is assisted by good architectural design which provides passive solar heating through south facing fenestration in the pool area.

The steam supply to the domestic hot water heating system was shut off in early October and had to be opened three days in mid November, so the entire domestic hot water requirements of the building have been satisfied by the solar installation.

DEFICIENCIES

More instrumentation is required to determine the actual heat recovery of this system and for that purpose meters will be installed to read the quantity of domestic hot water used as well as the flow of the pool water through the pool water heater. It appears that the hot water load is less than originally designed. Also the requirements of the building during the summer months for pool heating and make-up air heating will be quite small. It is our intention to make a connection with the hospital laundry and run any excess heat collected through the laundry system during the summer months.

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CONCLUSIONS

The installation will provide the operating experience desired. It appears to be over designed for the loads encountered, but perhaps with increased scheduling of activities within the building a better balance will be achieved.

ACKNOWLEDGEMENTS

The consulting enginneer for the mechanical and solar systems was Mr. Robert Parupsky of Hammel, Green, and Abrahamson, St. Paul, Minnesota and the solar schematic was developed by Dr. Grenn Yuhill of Unies, Winnepeg, Canada. The data was collected by Mr. Rolf Arneson, Physical Plant Director of the Willmar State Hospital.

COLLECTOR ARRAY PERFORMANCE OF INSTRUMENTED SITES IN THE NATIONAL SOLAR DATA NETWORK

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ABSTRACT

Data acquired by the National Solar Data Network (NSDN), as part of the National Solar Heating and Cooling Demonstration Program, has been converted into objective empirical collector array performance information. This information was presented to the solar community at the 1979 Department of Energy Regional Solar Update Conferences, in the form of computer-generated tabular and graphical data. That report provides the only source of operational data with sufficient breadth and depth to be used as an empirical reference. The report contains sufficient data to support a comparison of actual collector array performance to design goals; comparison of actual collector array performance for several months in a specific application; comparison between types of collectors (flat plate, evacuated tube, concentrators, air, liquid); and comparison among products within types. This paper illustrates a number of extractions which may be obtained from the Collector Array Performance Report. Examples include: causes of deviation of array performance from single panel test performance; variations of collector array performance between sites; variations of single site array performance from month to month; "typical" performance expectations for the generic collector types; and a guide to collector component selection for generic solar energy system designs.

INTRODUCTION

Six months after IBM entered into a contract with the Department of Energy to analyze the performance of Instrumented Solar Energy Installations of the National Solar Heating and Cooling Demonstration Program, it was evident that the data being acquired could also be useful in the analysis of subsystems and components of those installations. Toward that objective, IBM developed an interactive terminal-based software set with which to analyze the field performance of collector arrays (Reference [1]).

Examination of a small sample of results from the interactive software set led to the observation that collector arrays, in a field environment, do not behave as one would expect from single-panel laboratory test data. It became evident that solar energy system designers need more information than is provided by the performance characteristic curve of a single-panel in order to properly size the collector array to the design load. To fulfill that apparent need, IBM expanded the analysis of Reference [1] to include a histogram of collector array operating points and compare actual energy gain to predicted energy gain. The energy gain comparison contrasted the actual measured solar energy acquired by an array to the gain which would have been predicted (for identical operating conditions) from the single-panel curve. Operating points are known precisely and projecting to the instantaneous efficiency curve (Figure 1) computes the energy gain predicted by the curve. These results were reported at the 1978 Solar Heating and Cooling Operational Results Conference (Reference [2]), and indicated deviations on the order of 30 percent (low) between actual energy gain and predicted energy gain.

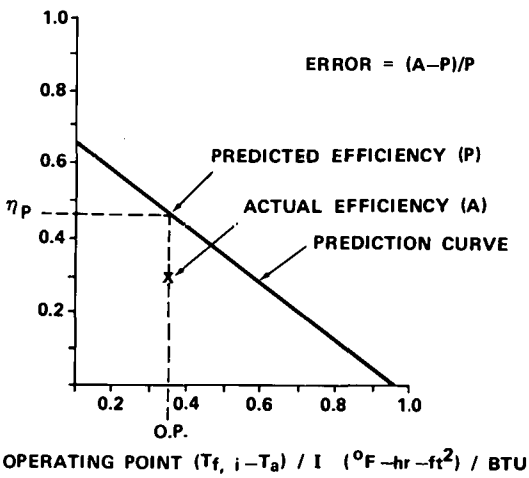


Figure 1. Energy Gain Comparison Methodolody

The next efforts concentrated on quantifying the dynamic effects of field operating conditions on the energy gain of a collector array. This problem was approached by deriving an instantaneous efficiency curve from field data by the techniques described in Reference [1] and using the field derived curve (instead of the single-panel laboratory derived curve) in the energy gain comparison technique described in Reference [2]. This led to

the interesting conclusions that, in general, dynamic effects result in errors on the order of five percent (Figure 2) and that the field-derived collector array instantaneous efficiency curve is the proper model to be used in design (Reference [3]). However, until July 1979, there was no source of empirical array performance data available for reference by designers.

Two avenues of investigation are indicated. First, since array effects dominate single-panel effects, a study was suggested which would quantify array effects and make the information available to the solar design community. Second, since the National Solar Data Network (NSDN) has access to numerous operational solar energy sites, a report was published which presents the field operational characteristics of over 50 sites, Reference [4].

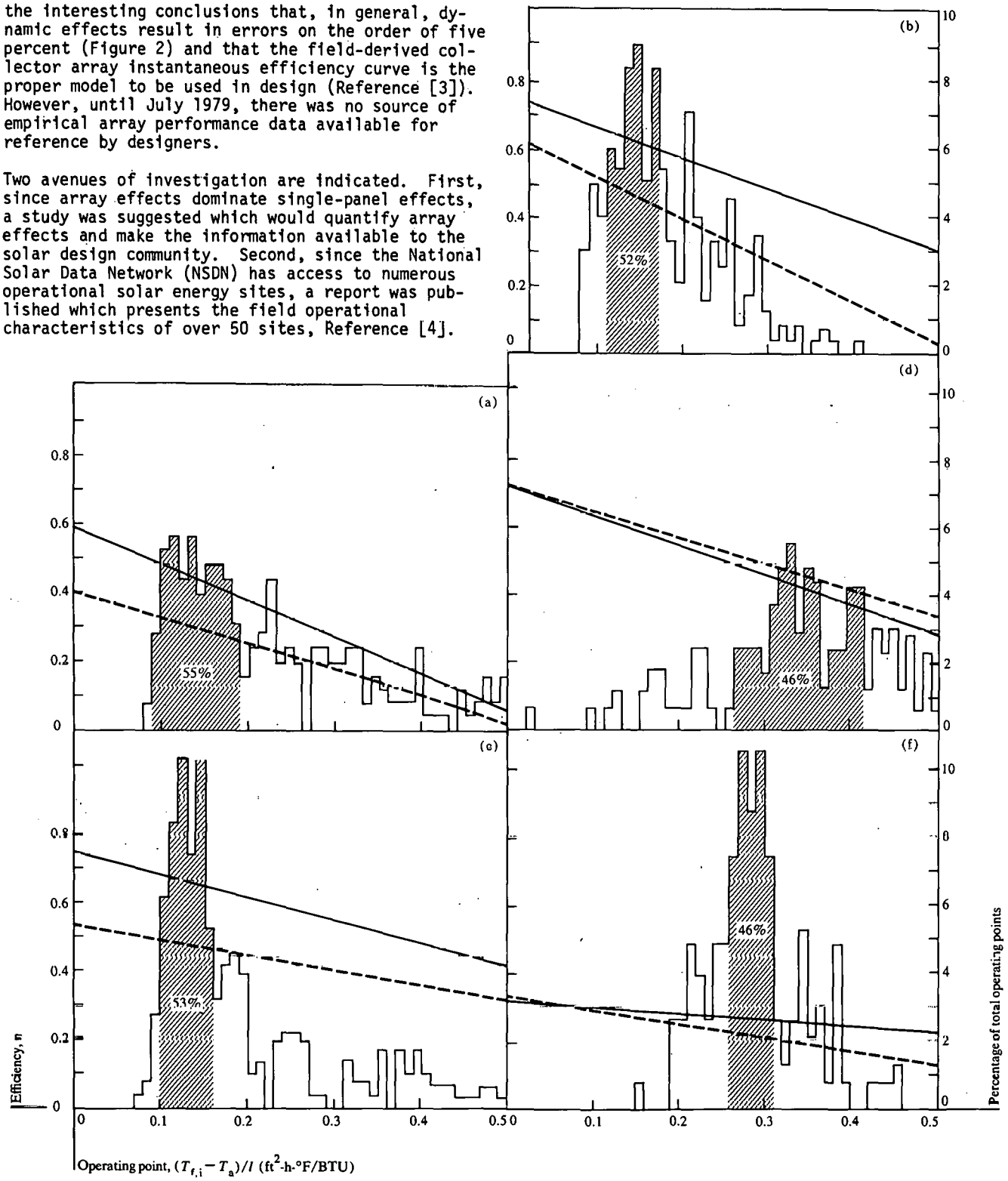


Figure 2. Collector Array Performance Comparison and Operating Point Histograms (Lab Curve —; Field Curve - -)

he latter report provides an immediately available guide (by similarity) for collector array performance predictions, which may be used in the interim period before publication of the results of an array effect quantification study.

INTERPRETATION OF THE COLLECTOR ARRAY PERFORMANCE REPORT

The Collector Array Performance Report presents the following information for each collector type, each site, each month, grouped by basic design concept (e.g., flat-plate, evacuated tube, etc.):

- Single-panel instantaneous efficiency curve coefficients
- Current month instantaneous efficiency curve coefficients
- Cumulative (through current month) instantaneous efficiency curve coefficients
- Cumulative (through current month) second order curve coefficients
- Single-panel description
- Array description
- Daily energy gain comparisons for the current month
- Properties of the collector transport fluid
- Array flow rate (average for the month)
- Single-panel flow rate
- Array temperature gain (average for the month)
- Monthly energy gain comparisons

The report also contains sections which define the mathematical approach used to derive the results for each collector design concept; and appendices which define the statistical techniques employed to develop the instantaneous efficiency curves, the transport fluid properties, and the accuracy of field derived results.

To produce data for such a large number of months required the development of automatic software to replace the interactive software system. The automatic analysis is based upon the philosophy and guidelines of ASHRAE Standard 93-77 (Reference [5]), and the implementation techniques defined in References [1], [2] and [3]. Since the Collector Array Performance Report was assembled from computer-printed pages, it does not contain interpretations of the data presented. This paper provides the user of the report with a guide to its interpretation.

MATHEMATICAL PRELIMINARIES

Time-coherent measurements of collector array parameters are taken once each 320 seconds by the site data acquisition system of the NSDN. These measurements are combined and integrated into an instantaneous efficiency value and an operating point value for each 320 second period. These two quantities are then paired and plotted on a cartesian coordinate system in a "scatter diagram."

When a linear least-squares regression line is plotted through the scatter diagram, we have derived the field-equivalent of the Hottel-Whillier-Bliss (HWB) instantaneous efficiency curve, which may be written:

$$\eta = F_R (\tau\alpha) - F_R U_L (T_{f,i} - T_a) / I \qquad (1)$$

where:

η	is the instantaneous efficiency	(Dimensionless)
F_R	is the heat removal factor	(Dimensionless)
U_L	is the collector heat loss coefficient	(Btu/ft^2-hr-°F)
$(\tau\alpha)$	is the effective cover transmittance - plate absorptance product	(Dimensionless)
$T_{f,i}$	is the collector array fluid inlet temperature	(°F)
T_a	is the ambient (sink) temperature	(°F)
I	is the insolation rate	(Btu/ft^2-hr)

Since the equation is derived from points which, frequently, are widely scattered, we also calculate a coefficient of determination for the curve. The coefficient of determination is the product of the slope of the regression of efficiency on operating point and the slope of the regression of operating point on efficiency; and is designated R**2. A value of 1.0 denotes a perfect fit. Values less than 1.0, down to about 0.3, indicate decreasing goodness of fit. Values less than 0.3 indicate relatively poor fit of the regression line to the scatter diagram.

Second order regression lines are also fit to the scatter diagram. There being no goodness of fit parameter corresponding to R**2, second order curve fits should be used with caution.

CAUSES OF DEVIATIONS FROM THE PANEL CURVE

Efficiency Axis Intercept Low

When the efficiency axis intercept is below expectations, the following points should be checked.

F_R, the heat removal factor, may be low. This is a direct function of flow rate and thermal conductivity of the bond between the absorber plate and the fluid transport tubes, and an inverse function of the heat loss coefficient.

τ, cover transmissivity, may be low due to opaque substances on the outer surface of the glazing, or condensation or outgassing residue on the inner surface of the glazing.

α, plate absorptivity, may be low due to deterioration of the absorber coating.

The operating point intercept is found by setting Eq. 1 to zero and solving for the operating point, X,

$$X_o = \tau\alpha/U_L \quad (2)$$

where:

$$X = (T_{f,i} - T_a)/I. \quad (3)$$

From Eq. 2, it is seen that variations in the heat removal factor, F_R, do not effect the X intercept. Decreases in transmissivity or absorptivity or increases in U_L will decrease the X intercept, while a decrease in U_L will increase the X intercept. The collector heat loss coefficient, U_L, is strongly affected by wind and ambient temperature, and is also subject to increase or decrease as a function of installation and perimeter insulation. Figure 3 illustrates an application of these principles.

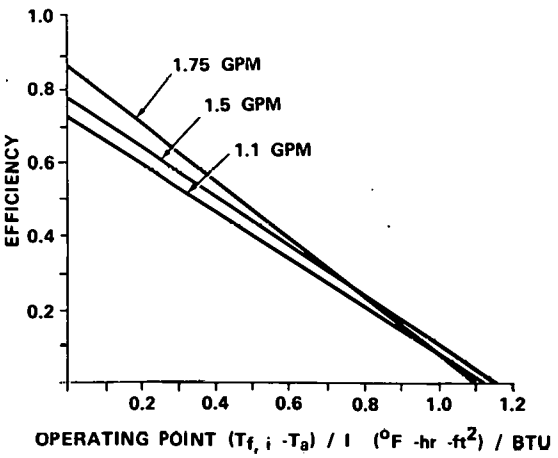


Figure 3. Identical Collector Arrays in Three Different Systems

TABLE STRUCTURE

The Collector Array Performance Report for sites of the National Solar Heating and Cooling Demonstration Program contains four tables and a histogram for each month of collector operation. These are illustrated in Tables 1 through 4 and Figure 4.

Table 1. Energy Gain Comparison Table (Computer Printout)

DAY 1 ACTUAL 5.137×10^5 BTU MONTH -0.011 LONG TERM 0.024 2ND ORDER 0.020 PANEL -0.215

SITE: FACILITIES DEVELOPMENT SAN DIEGO, CA.

	ACTUAL	ERROR: FIELD DERIVED MONTH	FIELD DERIVED LONG TERM	FIELD DERIVED 2ND ORDER	LAB PANEL
DAY					
1	5.137E+05	-0.011	0.024	0.020	-0.215
2	4.730E+05	-0.002	0.021	0.014	-0.211
		-0.014	0.005	0.001	-0.222
			0.011	0.003	-0.218
				0.019	-0.265
25	4.162E+05	-0.004			-0.197
26	4.485E+05	-0.037			
27	3.273E+05	0.048	0.024		
28	2.865E+05	0.001	0.102	0.086	
29	4.468E+05	0.035	0.069	0.040	-0.201
30	2.611E+05		0.071	0.067	-0.178
	1.234E+07	-0.016	0.021	0.013	-0.218

TOTAL ENERGY GAIN 1.234×10^7

MONTH WEIGHTED AVERAGE -0.016

LONG TERM WEIGHTED AVERAGE ERROR 0.021

2ND ORDER WEIGHTED AVERAGE ERROR 0.013

PANEL WEIGHTED AVERAGE ERROR -0.218

Table 2. Coefficients Table (Computer Printout)

CURVE	COEFFICIENTS A0 (FRTA)	A1 (FRUL)	A2 (*)	R**2
PANEL	0.730	-0.640	N.A.	N.A.
MONTH	0.965	-0.380	N.A.	0.131
LT1ST	0.578	-0.654	N.A.	0.395
LT2ND	0.588	-0.841	0.726	N.A.

Table 3. Fluid Properties Table (Computer Printout)

FLUID PROPERTIES - APRIL

	WATER			
	COEFFICIENTS			
PROPERTY	A0	A1	A2	A3
SPECIFIC HEAT	1.011E+00	-2.348E-04	1.037E-06	
DENSITY	6.346E+00	4.129E-04	-9.961E-06	

Table 4. Energy Gain Comparison Annual (Computer Printout)

SITE: FACILITIES DEVELOPMENT SAN DIEGO, CA.

			ERROR: FIELD DERIVED MONTH	FIELD DERIVED LONG TERM	FIELD DERIVED 2ND ORDER	LAB PANEL
MONTH	YEAR	ACTUAL				
JANUARY	79	7.696E+06	-0.024	0.001	-0.013	-0.242
FEBRUARY	79	1.134E+07	-0.018	0.002	-0.014	-0.241
MARCH	79	1.210E+07	-0.004	0.016	0.005	-0.219
APRIL	79	1.234E+07	-0.016	0.021	0.013	-0.218
MAY		0.000E+00	0.000	0.000	0.000	0.000
JUNE	78	5.266E+06	-0.051	-0.051	-0.021	-0.280
JULY	78	1.002E+07	-0.037	-0.038	-0.027	-0.271
AUGUST	78	1.068E+07	-0.021	-0.063	-0.055	-0.309
SEPTEMBER	78	1.018E+07	-0.019	-0.006	-0.012	-0.265
OCTOBER	78	1.012E+07	-0.006	0.017	0.010	-0.244
NOVEMBER	78	5.822E+06	-0.017	0.009	-0.007	-0.246
DECEMBER	78	5.667E+06	-0.024	0.010	-0.006	-0.238
AVERAGE		1.032E+07	-0.021	0.021	0.013	-0.251

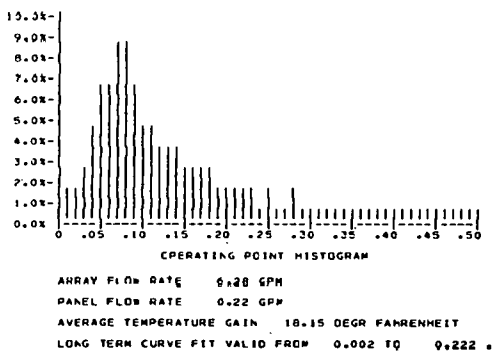


Figure 4. Histogram Example

COMPARISON OF ARRAY PERFORMANCE TO SINGLE-PANEL PREDICTION

The collector array analysis report investigates 50 collector installations. Of these 50 installations, 4 were in close agreement (5%) with the single-panel projection, 12 were substantially better (actually collected more) than the single-panel prediction and 34 were substantially worse (actually collected less) than the single-panel prediction. Table 5 summarizes these findings.

ble 5. Statistical Comparison of Array Performance to Single-Panel Prediction

			WITHIN 5%	BETTER	WORSE
LIQUID	SINGLE-GLAZED	NON-SELECTIVE	0	6	2
		SELECTIVE	1	0	3
	DOUBLE-GLAZED	NON-SELECTIVE	1	3	7
		SELECTIVE	0	0	7
	EVACUATED TUBE		1	0	1
	CONCENTRATOR		0	0	2
AIR	SINGLE-GLAZED	NON-SELECTIVE	0	0	6
		SELECTIVE	0	0	1
	DOUBLE-GLAZED	NON-SELECTIVE	1	3	5
		SELECTIVE	0	0	0

VARIATION OF INSTANTANEOUS EFFICIENCY CURVE WITH TIME

Figure 5 presents the monthly instantaneous efficiency curves for a single site. Notice that the efficiency axis intercept varies little, but the operating point axis intercept varies from 0.6 to 0.72. Variations of the operating point axis intercept appear to be due to changes in U_L brought about by changes in the external ambient (sink) temperature.

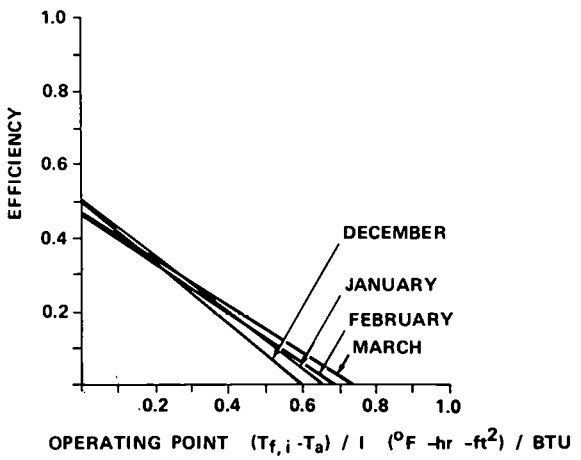


Figure 5. Variations of Efficiency Curves by Month

RELATIVE PERFORMANCE OF COLLECTOR TYPES

Figure 6 presents the relative performance of representative collector types. Figure 6a is the HWB curve for a single-glazed, non-selective absorber collector array plotted over a histogram of the collector operating points for an example month. Similarly, Figure 6b-6j depict other generic collector configurations. The note on the graph is the energy acquired by the collector array, normalized to the number of square feet of collector area. The significant aspects of any collector array are (1) energy acquired, (2) the cost of its acquisition and (3) the delivery temperature. The histogram (Figure 4) indirectly indicates the delivery temperature. Since the operating points are defined by Eq. 3, higher delivery temperatures are implied by a concentration of operating points (longer histogram bars) to the right of the graph. Relative comparisons among types must consider the design operating point. Thus a single-glazed, non-selective absorber collector, which has a high efficiency intercept and low operating point intercept will not operate efficiently at the inlet temperatures (as reflected in the operating points) required for absorption chiller operation (> 160°F).

COLLECTOR SELECTION FOR TYPICAL APPLICATIONS

Table 6 presents an empirical guide to the selection of collector types for specific applications.

The empirical guide in Table 6 lists the indications of trends extracted from a survey of the collector array performance report. Array effects are included in the derived efficiency numbers, and each new design should refer to the array from which the efficiency numbers are obtained.

Table 6. Collector Selection

APPLICATION	DELIVERY TEMPERATURE	OPERATING POINT RANGE	MEDIUM	TYPE	EFFICIENCY
HOT WATER	140°	0.08 - 0.34	LIQUID	SG,NS	0.478
HEATING	110°	0.20 - 0.44	LIQUID	SG,NS,CS	0.562
	110°	0.20 - 0.44	LIQUID	SG,S	0.415
	90°	0.12 - 0.33	AIR	SG,NS	0.346
COOLING	160°	0.26 - 0.55	LIQUID	ET	0.397

SG = SINGLE GLAZED S = SELECTIVE ABSORBER
NS = NON-SELECTIVE ABSORBER CS = INTERNAL CONVECTION SUPPRESSION
ET = EVACUATED TUBE

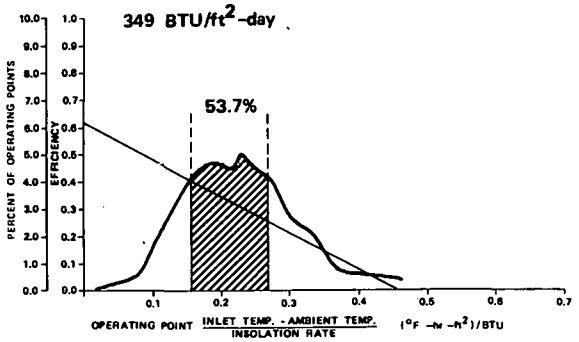
SUMMARY

The Collector Array Performance Report for sites of the National Solar Heating and Cooling Demonstration Program (Reference [4]) provides a basis for empirical selection of collector array design concepts. This paper illustrates a few of the observations which may be extracted from Reference [4]. Designers have been provided with the information necessary to make sound collector component selection decisions; and with data relating the actual performance of operational collector arrays. This information, when intelligently integrated into cost trade analyses and system configuration selection, will contribute to the minimization of the cost of solar energy systems; and, ultimately, to the expansion of the solar energy industry.

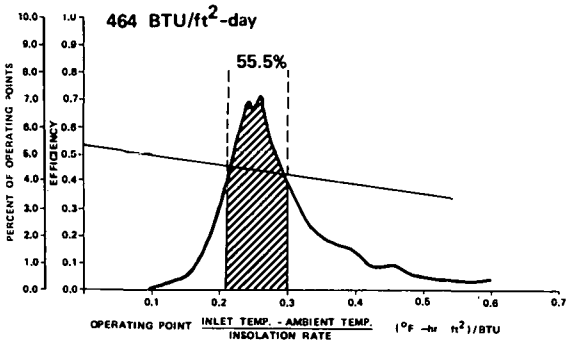
ACKNOWLEDGEMENT

This work was sponsored by the Department of Energy, Office of Conservation and Solar Applications under Contract EG-77-C-01-4049.

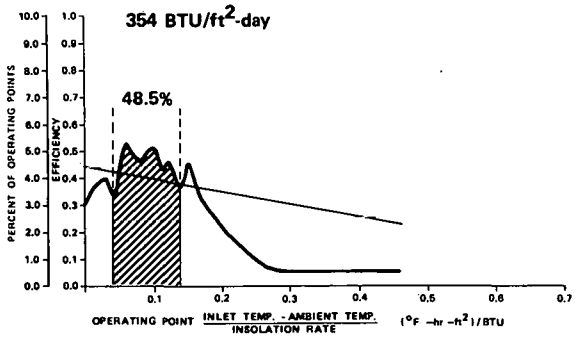
(a) Flat Plate Liquid Collector, Single Glazed, Non-Selective



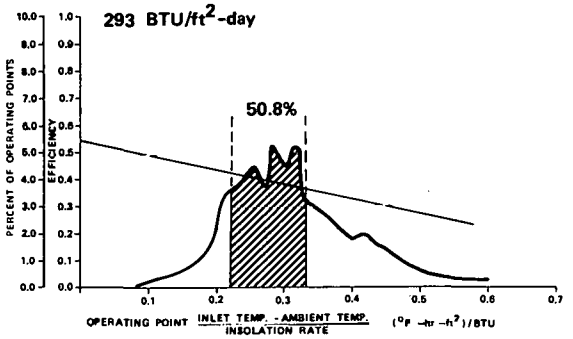
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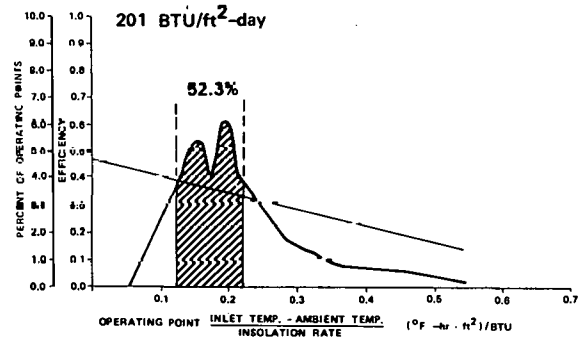
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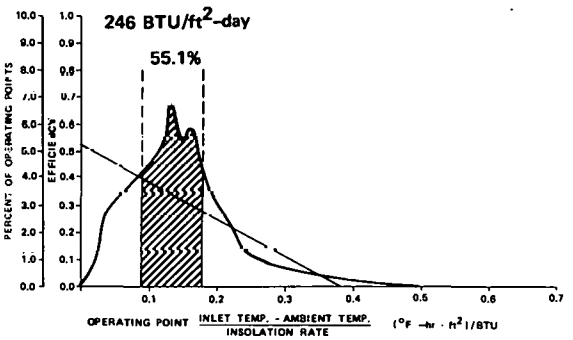
(d) Flat Plate Liquid Collector, Double Glazed, Selective



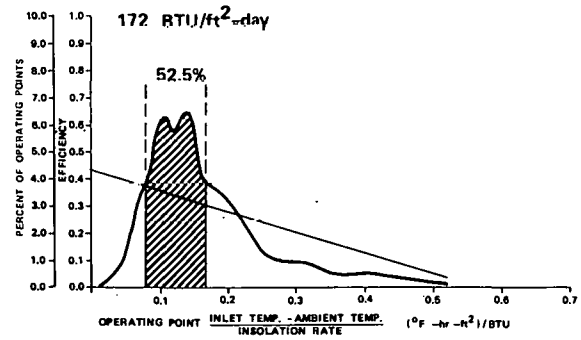
(e) Flat Plate Air Collector, Single Glazed, Non-Selective



(f) Flat Plate Air Collector, Single Glazed, Selective



(g) Flat Plate Air Collector, Double Glazed, Non-Selective

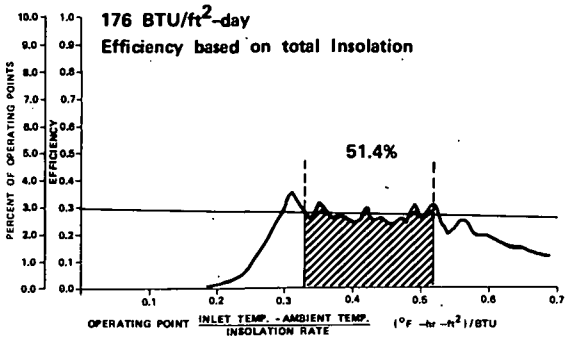


(h) Flat Plate Air Collector, Double Glazed, Selective

[NO EXAMPLE]

Figure 6. Comparative Performance at Representative Collector Designs

Concentrating Collector



(j) Evacuated Tube Collector

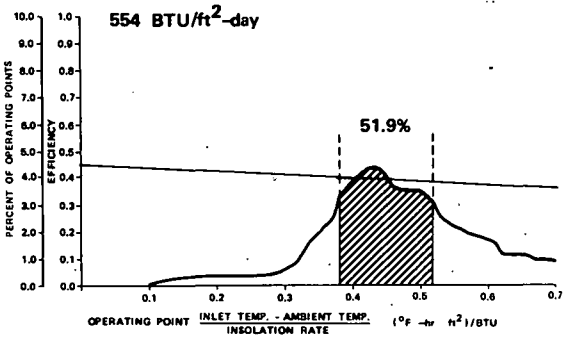


Figure 6. Comparative Performance at Representative Collector Designs (Cont.)

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NOTES

PERFORMANCE OF A SOLAR HOT WATER SYSTEM INSTALLED IN AN INTERSTATE HIGHWAY VISITOR CENTER

RESULTS OF ONE YEAR OF MONITORING

David R. Jackson Director, Solar Energy Division
University of Connecticut Energy Center

ABSTRACT

The Connecticut Department of Transportation maintains Rest Area Visitor Centers at a number of locations on the interstate highways passing through Connecticut. At one of these centers, located in Willington, Connecticut on Interstate 86 Eastbound, a solar hot water system has been installed and has been operating since early July, 1978. The system is heavily instrumented and is being monitored by a 24 channel magnetic tape data logger with a one minute sampling rate.

The objective of the project for the funding agency, the Connecticut Department of Transportation, is to determine the feasibility of long term energy conservation through the use of solar augmentation for the entire rest area system. The unusual hot water use profile of such a facility, however, prevents reliance on a paper study using the "F-Chart" method for this feasibility determination. The rapid sampling rate of the data acquisition system provides an extremely detailed picture of the operation of the system under this widely fluctuating hot water demand, and ultimately insight into the effectiveness and economics of solar energy in this application.

Since becoming operational in July 1978 very little trouble has been encountered with solar system operation (excluding the data system which has experienced some down time). Performance predictions using the F-Chart method have produced sizeable deviations from actual performance in individual months, all of which fall below actual measured monthly values.

Operational results for the first three months of monitored performance were reported at the First Annual Operational Results Conference. This paper includes performance data for the full one year period from September 1, 1978 to August 31, 1979.

Monthly summaries of all important energy flows are presented. Incident solar radiation will be traced through the system and the various losses accounted for. Insolation data and hourly and daily water use characteristics for the facility are also presented.

INTRODUCTION

State and municipal agencies throughout the country are in the face of rising utility costs becoming increasingly aware of the need to reduce consumption of conventional energy. The Connecticut Department of Transportation through the Joint Highway Research Council decided in the fall of 1977 to fund the subject pilot project. The roadside rest area/visitor center is an interesting application for a solar hot water system because of the unusual and somewhat unpredictable hot water demand characteristics. It was felt at the outset that strictly a paper economic study based on the usual "F-Chart" [1] system sizing procedure could not be depended on to determine the efficacy of solar energy for this application because of both the lack of knowledge of the total amount of water used in such a facility, and an almost certain deviation of the hourly water use characteristics from those used in determining the "F-Chart" correlation. Consequently it was felt that an actual installation, heavily instrumented, would yield superior feasibility information for the Department of Transportation and also provide valuable engineering data as well.

The facility chosen for the experiment is located on Interstate 86 Eastbound in Willington, Connecticut. This location is conveniently close to the University of Connecticut and is located on a major traffic corridor linking New York, New Haven, and Hartford to Boston and other points serviced by the Massachusetts Turnpike which it intersects 20 miles farther East. In addition to restroom facilities, the visitor center offers travelers information and picnic table facilities, both of which tend to increase hot water consumption in the warmer months.

SYSTEM DESCRIPTION

The system component breakdown and plumbing schematic are given in Table 1 and Fig. 1, respectively.

The collectors are mounted at a 45 degree angle with respect to the horizontal and are supported at this altitude by a framework of galvanized iron pipe and aluminum extrusions structurally tied to the trusses of the 30 degree pitched, south-facing roof. The storage tank, pumps and other mechanical

equipment are located in the basement. The round trip pipe run to the collectors is approximately 200 feet (60 M).

The system is of the closed loop type, employing a finned tubing heat exchanger in the 120 gallon storage tank. Preheated water from the storage tank is delivered to the inlet of the existing 120 gallon hot water heater where, if necessary, it is heated further to delivery temperature. Scalding protection is accomplished by an adjustable tempering valve at the hot water heater outlet. Bypass and isolation valves are employed to allow a return to the previous mode of operation should the solar part of the system need to be shut down and drained.

Silicone heat transfer fluid was selected over various water/glycol mixtures to give as maintenance-free a system as possible over a long time period and an opportunity to study system performance with a non-aquous fluid. It was necessary to use two circulating pumps in series to obtain a sufficiently high capacitive flow rate in the collectors approximately 5 gallons/minute (19 L/Min) which corresponds to a water flow rate of 1.75 gallons/minute (6.6 L/Min).

TABLE I

SYSTEM COMPONENT BREAKDOWN

ITEM	SYSTEM COMPONENT BREAKDOWN
Solar Collectors	- (5) L.O.F. Model 221 type 3277 Double-glazed with low iron glass - Black chrome on copper tube and sheet absorber - Extruded aluminum frame - 3' X 7' gross dimensions - Fiberglass insulation - $F_R(\tau\alpha)$ = .648, F_RU_L= 4.42 w/m^2/°C
Storage Tank	- Vaughn-sepco 120 gallon, stone-lined - Integral finned copper heat exchanger of 40 ft^2 claimed "effective" area
Solar Loop Circulator(s)	- (2) Grundfus UP-26-64 canned circulators in series
Controller	- (1) Hawthorne "Variflo" variable speed
Auxiliary Heater	- Hubbell 120 gallon stone lined electric 3-phase upper and lower elements each 12 kilowatts. This is the original hot water heater.
Insulation (pipe)	- 3 inch O.D. X 3/4 inch I.D. rigid Urethate foam with polyvinyl chloride outer cover.
Insulation (tanks) (supplementary)	- 1.5 inch vinyl-clad fiberglass outer blanket wrap
Heat Transfer Fluid	Dow Corning Q2-1132 Silicone

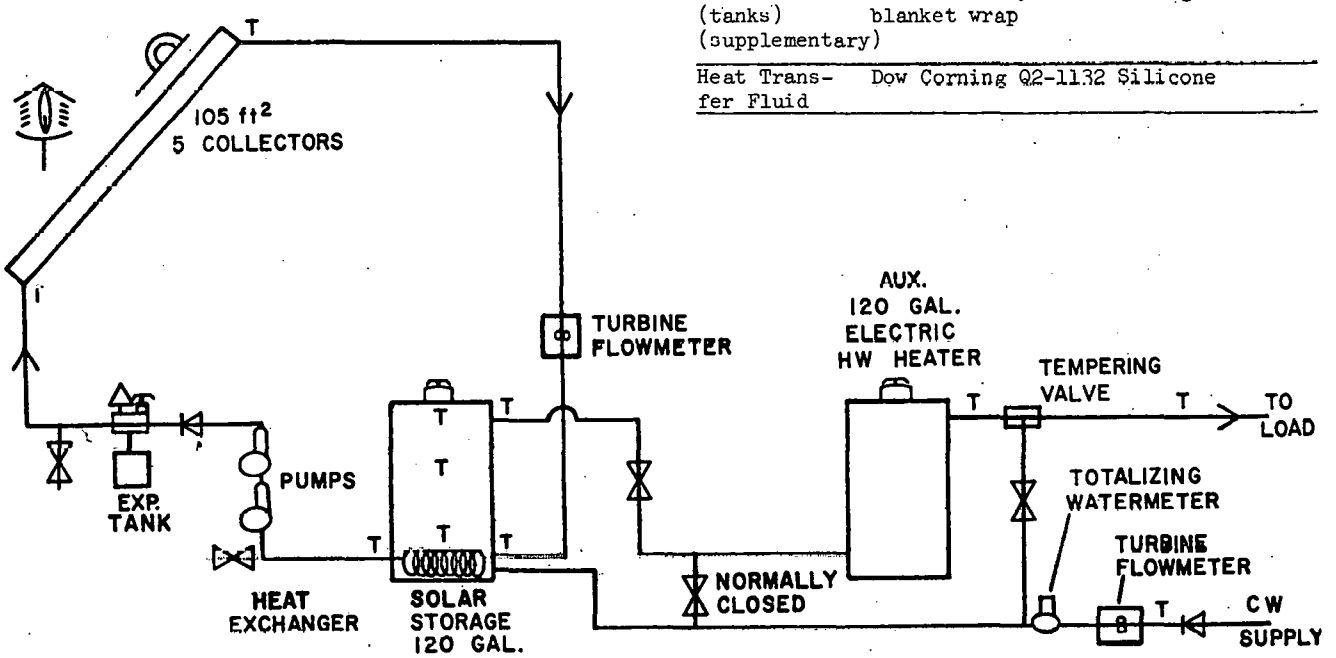


Fig. 1 Solar Hot Water System Schematic

DATA ACQUISITION

The data logger was custom built for this application by the investigators. The heart of the machine is a Datel Systems LPS-16 Data Logger module which performs a 12 bit analog to digital conversion on 16 channels and records the resulting words on a tape cassette. This module is interfaced with "homebrew" control circuitry and a crystal clock to control sampling sequence and rate. The module is also capable of recording digital serial words from the clock and flow totalizers. Sixteen channels of adjustable gain instrumentation amplifiers serve to boost transducer signals to the 0 to 5 volt range required by the analog to digital converter. The tapes produced by the data logger are processed by a Datel Systems LPR-16 tape reader which has been incorporated into the PDP-1155 computer located in the Mechanical Engineering Department of the University.

The disk files from each tape are then reduced to hourly summaries and are archived in other disk files for further processing to daily and monthly summaries.

The data acquisition scheme employed to monitor the system involves running the data logger at a rather high sampling rate (one complete scan every minute) and recording every sample rather than recording averages computed in software on board the logger. The advantages of recording all samples is that the data may be archived in the raw state thus allowing various analyses to be run at a later date which inherently need the transient data and are either too complicated to be incorporated in data logger software or simply have not been thought of yet. The disadvantage of recording 20 16-bit words every minute is that one cassette which lasts only 3 days contains over 80 kilobytes of digital information - enough to overwhelm many minicomputer systems. In this project, each data tape is read into a PDP 11-55 minicomputer which converts each sample to engineering units and stores the resulting data file on disk. The data system has not been totally free from problems over the year of data acquisition. In addition to the normal occurrence of occasional individual transducer or channel problems, there have been a few clock stoppages and incidents of failure to change the cassette tape. The data presented in this paper for each month, however, contains no less than 20 days and in most cases is either complete or missing one or two days. Overall, the data yield is 85%.

INSTRUMENTATION

An attempt was made in designing the instrumentation for the project to come as close as practically possible to laboratory measurement accuracy in all parameters needed to define the principal energy flows in the system. Incident solar energy is measured by an Eppley Laboratories Precision Pyranometer mounted in the collector plane. Since the output of this instrument is recorded every minute, an extremely detailed picture of an hourly and daily insolation profile is available through software averaging while preserving the instantaneous values for transient calculations. Temperatures in the collector loop are measured at the collector bank inlet and outlet headers and at either side of the storage heat exchanger. Omega Engineering Precision Thermistors with a ± 0.36°F (±0.2°C) nominal accuracy are used to measure all fluid temperatures, with the exception of the internal tank temperatures which are measured with type T thermocouples. The thermistors are purchased encapsulated in stainless steel probes which are immersed directly in the medium to be measured at a depth of at least 3 inches. In instances where temperature readings will be subtracted in subsequent calculations, (e.g., the temperature difference across the collectors) matched pairs were selected in the laboratory to further reduce errors in the results. Thermistor resistances are converted to voltages in wheatstone bridge circuits built into the data logger. The bridges utilize military grade precision low drift resistors, and the bridge exitation voltage is recorded on a separate data channel to eliminate inaccuracy resulting from power supply drift. Of critical importance in monitoring energy flow in the collector loop is the accurate determination of collector heat transfer fluid flow rate. A Fisher and Porter 3/4 inch turbine flow meter with associated electronics produces both a digital pulse train and analog current output so that total flow over the one minute period and instantaneous flow may be recorded. The pulse train is totalized by a set of decade counters within the data logger which are reset after the total flow for the minute is recorded. The turbine flow meter has a rated accuracy of ±1/4% with water but the water calibration obviously could not be considered valid for other fluids. A gravimetric calibration was carried out using silicone fluid with the transducer installed in the system.

Other temperature measurements include the incoming water, storage tank outlet, three points within the storage tank, the water leaving the auxiliary hot water heater, and the delivery temperature downstream of the tempering valve. These temperatures are measured in the same manner as those in the collector loop. Outdoor ambient temperature data is acquired by a similar thermistor located in a radiation shield on the facility roof near the collectors.

Water flow through the system is measured by both the conventional water meter and a 3/8 inch Fisher and Porter turbine with totalizing electronics similar to those described in the collector loop flow measuring system. The totalizer is especially important here because of the "on-off" nature of handwashing water flow.

The instantaneous power consumption of the two electric elements in the auxiliary hot water heater is monitored by a status signal which indicates whether one or both auxiliary elements are on by the level of the status voltage.

The status signals are later correlated with the reading of the auxiliary heater watt-hour meter and the heater power for each minute of operation is determined. A similar approach is used for the pumping power data. A transducer breakdown is provided in Table II and the measurement locations are described schematically in fig. 1.

OPERATIONAL RESULTS

A year of data corresponding to the period September 1, 1978 through August 31, 1979 has been analyzed and is reported in this section. The period of September 1 1978 through December 31, 1979 is reported in somewhat less detail as is subsequent data because of limitations in the software used to reduce the earlier tapes and hardware problems which existed over that earlier period.

An energy flow summary is presented in Table III, giving daily averages for all sources and losses of solar and electrical energy in the system for the 12 month reporting period. Energy flows which were measured in the collector loop are estimated to be of ±10% accuracy whereas energy flows derived from measurements made in water lines are at least ±5% accuracy. This difference in accuracy stems from the difficulty experienced in metering the solar collector loop flow using a turbine flow meter with a silicone oil collector fluid. Electrical energy consumption and insolation measurements are estimated to be better than ±2% accuracy. The absence of entries in the table for the September 1978 through December 1978 period was the result of the shortcomings of early data reduction software in the handling of collector loop energy flows.

A number of performance indicators and figures of merit derived from the energy flows of Table III are presented in Table IV. The manner in which these parameters were calculated is revealed by row number references to the energy flows of Table III which went into the calculation of each parameter.

DISCUSSION OF RESULTS

One of the more interesting characteristics of this system is the degree to which pump electrical energy contributes to the thermal energy delivered to the load - 7.5% as a yearly average. The conversion COP from pump electricity to collector loop thermal energy (and also collector loop piping losses) were determined in detail for the month of July and August and the results applied to the rest of the year. The data from these two months were used because the four temperature data channels needed to determine pump contribution and piping losses were totally free from problems over that period. The piping heat loss values for this period agree well with heat transfer estimates based on average temperature and insulation thickness and properties. The seemingly excessive pumping energy found in this system is the indirect result of the use of the silicone oil transfer fluid and the particular flow meter (an existing piece of equipment) used to meter the collector loop flow. A second pump was added in August of 1978 to raise the flow rate into an easier metering range (from 2.5 gal/min to 5.0 gal/min). While the higher flow rate is certainly better from a performance standpoint, it is dubious that the small gains made thermally in better collector and heat exchanger performance would ever offset the extra electrical consumption of the second pump. The amount of energy allocated to pumping in this system is also felt in the SCOP values, which would be nearly doubled if one of the pumps were eliminated.

Also presented in Table IV is a comparison of the solar load satisfaction fraction compared with "F-Chart" prediction for each month. There are two load fractions presented in Table IV. The first, labeled "Solar energy load fraction" is the ratio of solar energy to total energy delivered to the hot water faucets. This parameter is the lower of the two and is constrained to be less than one because some electrical energy must be expended in the collection of solar energy. The second F value, labeled "Solar system fraction" is derived from an energy balance made on the auxiliary heater (through which all delivered energy flows in this system). If this "F" is equal to one, no auxiliary heater power is consumed in the production of hot water. The system is not penalized for pumping energy in the calculation of this parameter but is required to supply enough energy to overcome auxiliary heater losses inherent in satisfying the load. As can be seen from Table IV, the F-Chart predictions are bracketed on the yearly basis by these two reported F values and, on the whole, the agreement is quite reasonable. This is particularly encouraging when one compares the "F-Chart" assumed hot water consumption profile and the one measured in this experiment (fig. 2). The more even nature of the time distribution of water consumption in the visitor center probably accounts to some degree for the good performance of the system.

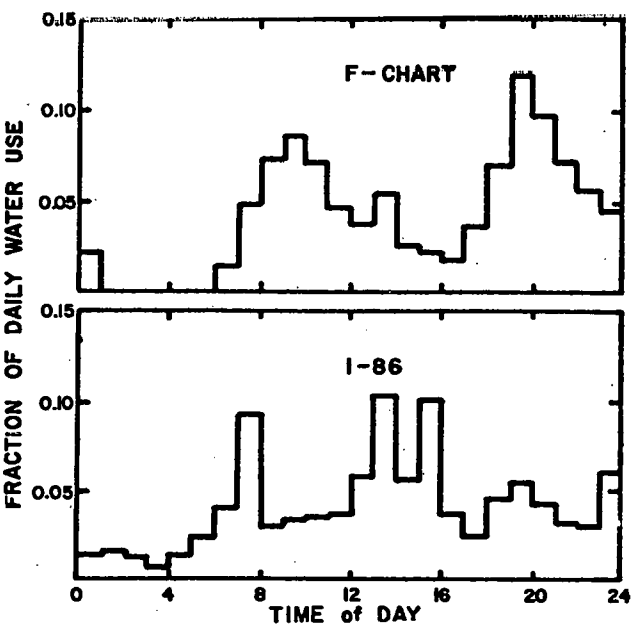


Fig. 2 Hot Water Use Characteristics

TABLE II

INSTRUMENTATION LIST

PARAMETER	TRANSDUCER	TYPE OF SIGNAL	PRECONDITIONING	OVERALL ACCURACY
(1) Insolation	Eppley PSP Pyranometer in Plane of Absorber	D.C. Millivoltage	Amplifier	± 2%
(2) Collector Loop Silicone Flowrate	3/4 inch Turbine Flow Meter Fisher and Porter	Chopped Radio-frequency Beam	1. Preamplifier creates Digital Pulsetrain which is Totalized by Decade Counters then Recorded as Total Flow for One Minute 2. Analog Signal is Created by Frequency to Current Converters and Hence to Voltage Across Precision Resistor	± 2%
(3) Water Flowrate	3/8 inch Turbine and Totalizing Water Meter	Same as (2)	Same as (2)	± 1%
(4) Temperatures:				
a) Outdoor Ambient	Omega "Unicurve" Thermistor in Ventilated Radiation Shield on Roof Near Collectors	Variable Resistance	Precision Wheatstone Bridge with Low Temperature Drift Resistors and Amplifier	±0.9°F (0.5°C)
b) Collector Fluid Inlet Header	Omega Unicurve Thermistor Stainless Steel Sheathed Immersed Directly in Fluid Medium	"	"	"
c) Collector Fluid Outlet Header	"	"	"	"
d) Storage Heat Exchanger Inlet	"	"	"	"
e) Storage Heat Exchanger Outlet	"	"	"	"
f) Cold Water Inlet	"	"	"	"
g) Storage Tank Water Outlet	"	"	"	"
h) Water Leaving Storage Tank	"	"	"	"
i) Water Leaving Hot Water Heater	"	"	"	"
j) Hot Water Leaving Tempering Valve	"	"	"	"
k) Storage Tank top l) middle m) Internal bottom	Sheathed-type T thermocouples with electronic ice point reference	millivoltage	High Gain Amplifier	± 1°C
(5) Time	Crystal Oscillator	Pulsetrain	Digital Totalizer	± 1 min per week
(6) Bridge Voltage	Power Supply	D.C. Voltage	None	± 2 M.V.
(7) Auxiliary Heater Power	Transformer and Watt-hour meter	"	"	--
(8) Pump Power	"	"	"	--

The energy "bottom line" or displaced electrical energy reported in Table II amounts to 3270. KWH for the monitoring year. This amount of electricity generated largely by nuclear power plants located in Connecticut has a current dollar value of about $163.00, representing roughly 5% of the current purchase and installation cost of the solar hot water system.

Considerable data has been accumulated in this project regarding storage tank thermal stratification and heat exchanger performance. Figures 3, 4 and 5 show average daily temperatures for the storage tank as well as the entering and exiting collector fluid and water temperatures for the months of February, June and an 8-month average (January - August). A glance at these figures indicates a rather mediocre heat exchanger performance (based on average tank temperature) and excellent tank thermal stratification which pretty much compensates for the former. The stability and magnitude of this stratification is one of the more pleasant surprises found in the data from the project. The heat exchanger performance is difficult to interpret under the conditions of stratification present here. The infinite capacity flow-mixed tank model of effectiveness reported is not really appropriate to the situation. If the storage tank system is considered a large heat exchanger, then very high effectiveness values would be calculated from this data. It is felt by the author that improvements in heat exchanger performance would probably be manifested in reduced stratification with little net gain in system performance.

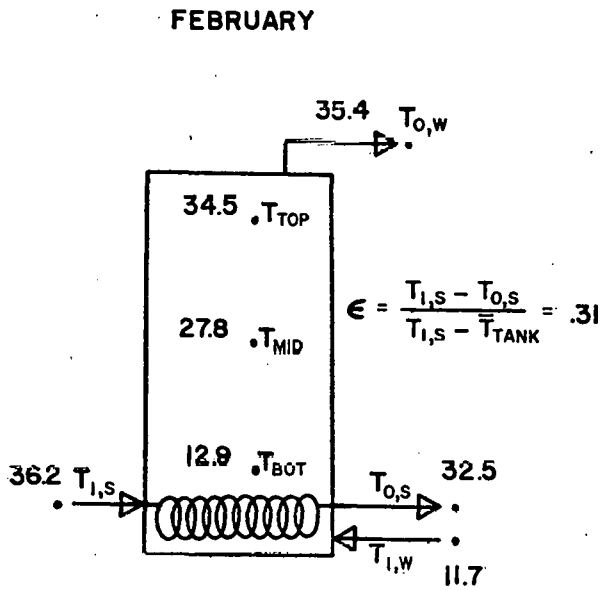


Fig. 3 Stratification and Heat Exchanger Performance (°C) for February 1979

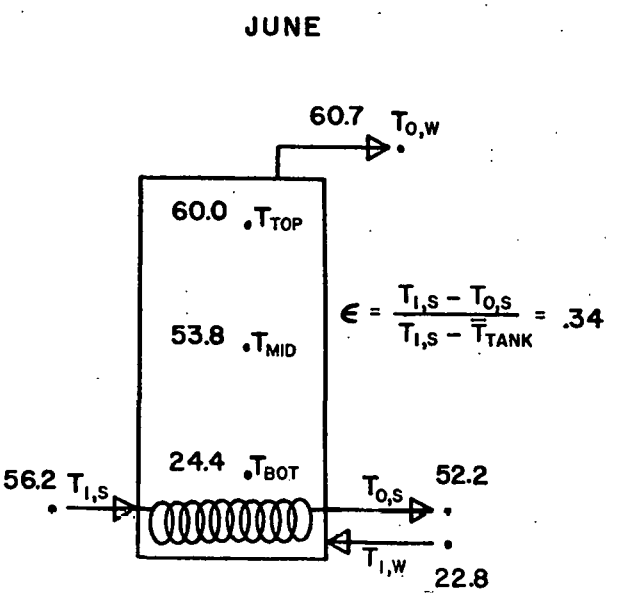


Fig. 4 Stratification and Heat Exchanger Performance (°C) for June 1979

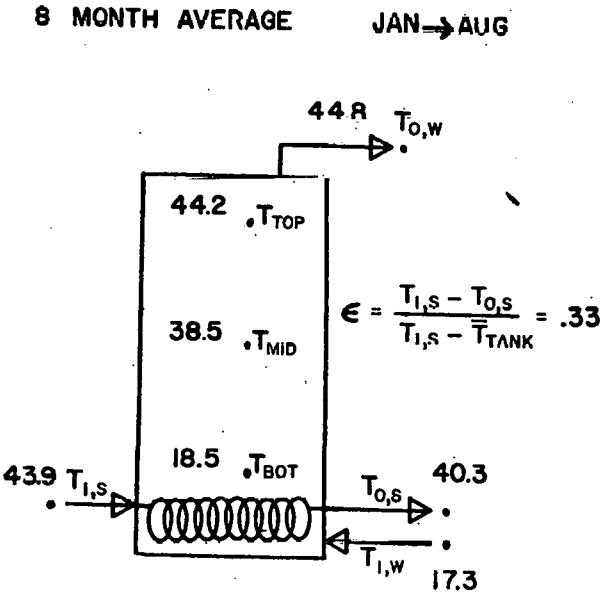


Fig. 5 Average Stratification and Heat Exchanger Performance (°C) for January to August

TABLE IV

DAILY AVERAGE OPERATING PARAMETERS
I-86 VISITOR CENTER

	SEPT	OCT	NOV	←1978 DEC	1979→ JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	YEARLY AVERAGE
(19) Component C.O.P.													
a) Collector Loop Piping	.85*	.85*	.85*	.85*	.852*	.854*	.851*	.854*	.861*	.842*	.849	.843	.851
b) Solar Storage Tank	.9*	.92*	.94*	.95*	1.0	.965	.941	.944	.903	.873	.886	.905	.927
c) Auxiliary Heater	.961	.901	.916	.854	.889	.919	.872	.868	.850	.825	.813	.896	.880
d) Pump Conversion of Electricity to Thermal Energy	.74*	.74*	.74*	.74*	.74*	.74*	.74*	.74*	.74*	.74*	.74	.74	.74
(20) System Solar Efficiency $\eta_s = \frac{(16)}{(1)}$**	.321	.285	.271	.232	.244	.233	.269	.244	.253	.213	.242	.288	.260
(21) System Thermal Efficiency $\eta_{ST} = \frac{(14)}{(11)}$	.357	.315	.304	.250	.278	.266	.312	.288	.297	.246	.279	.334	.297
(22) Collector Daily Efficiency $\eta_c = \frac{\bar{Q}_u}{A_c \bar{I}_T}$					.321	.307	.385	.349	.384	.352	.397	.422	
(23) SCOP= SCOP' $\frac{(16)}{(5)}$	4.82	5.02	4.45	3.63	3.92	4.02	3.23	2.95	2.89	2.98	3.02	3.28	3.61
(24) Solar Energy Load Fraction (16)/(11)	.686	.559	.405	.359	.266	.398	.507	.458	.677	.835	.763	.716	.534
(25) Solar System Fraction $\frac{(8)}{(11+10)}$	.763	.618	.455	.410	.304	.456	.588	.539	.790	.963	.879	.828	.613
(26) "F-Chart" Prediction	.643	.582	.406	.362	.187	.44	.488	.497	.674	.837	.776	.626	.562

(* Estimate)
(** Numbers refer to energy flows entries of Table III)

CONCLUSIONS

(1) The performance of the I-86 visitor center hot water system is satisfactory and very close to "F-Chart" predictions.

(2) Apparently low values reported for solar system COP are the result of extra pumping capacity added to the system for flow metering purposes. The reported values could be nearly doubled by the elimination of one pump.

(3) Heat exchanger performance appears mediocre but is offset by a large and stable degree of storage tank thermal stratification.

(4) In its first year of monitored performance, the system saved 3270 KWH of electrical energy which corresponds to roughly 5% of its purchase price.

ACKNOWLEDGEMENTS

The author expresses his gratitude to the Connecticut Department of Transportation Research Division for providing the funding for the experiment, and the support of the North East Solar Energy Center for its support of our ongoing detailed data analysis.

REFERENCES

[1] Beckman, W. A., Klein, S. A., Duffie, J. A., Solar Heating Design by The F-Chart Method, Wiley Interscience, New York, 1977.

TABLE III

MONTHLY ENERGY FLOW SUMMARY
Megajoules/Day ~10^6J/Day (10^6J = 1055 Btu)
I-86 VISITOR CENTER, WEST WILLINGTON, CT.

	SEPT	OCT	←1978 NOV	DEC	1979→ JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	YEARLY AVERAGE
(1) Incident Radiation Total in Plane $A_c\bar{I}_T$	175.6	156.6	116.6	96.43	76.95	134.6	116.7	124.5	138.7	175.5	163.3	131.9	133.3
(2) Collector Output $\bar{Q}_u^A$					24.73	41.44	44.91	43.44	53.24	61.76	64.86	55.72	
(3) Collector Loop Piping Losses					4.18	6.87	7.73	7.43	8.65	11.2	11.26	10.04	
(4) Thermal Energy Added to Collector Loop by Pumps	8.7	6.6	5.3	4.3	3.5	5.8	7.2	7.6	9.0	9.3	9.68	8.57	7.13
(5) Electrical Power to Pumps ~$\bar{E}$	11.7	8.9	7.1	5.8	4.8	7.8	9.7	10.3	12.2	12.6	13.1	11.6	9.6
(6) Total Thermal Energy Delivered to Storage (2) + (4) - (3)					24.07	40.36	44.37	43.65	53.59	59.9	63.28	54.25	
(7) Storage Losses					0	1.38	2.58	2.46	5.19	7.6	7.18	5.56	
(8) System Thermal Energy Delivered to Aux. Heater	65.2	54.8	38.7	28.2	24.1	39.0	41.8	41.2	48.4	52.4	56.1	48.7	44.9
(9) Aux. Heater Electrical Energy	20.2	33.9	46.3	40.6	55.3	46.7	29.2	35.3	12.7	2.10	7.74	10.1	28.3
(10) Losses from Aux. Heater	3.3	8.8	7.1	10.0	8.8	6.9	9.1	10.1	9.2	9.5	11.9	5.6	8.3
(11) Total Thermal Energy Delivered to Load	82.2	79.9	77.9	58.7	70.5	78.8	61.9	66.4	52.0	44.9	51.9	53.2	64.9
(12) Solar Energy Delivered to Storage					21.1	35.4	38.2	37.1	45.8	52.0	54.91	47.0	
(13) Solar Energy Delivered to Aux. Heater	58.6	49.6	34.5	24.7	21.1	34.2	36.0	35.0	41.4	45.4	48.1	42.1	39.3
(14) Solar System Thermal Energy Delivered to Load $\bar{Q}_H + \xi\bar{E}$(COP)	62.7	49.4	35.5	24.1	21.4	35.8	36.5	35.8	41.2	43.2	45.6	44.1	39.6
(15) Pump Thermal Energy Delivered To Load $\xi\bar{E}$(COP)	6.3	4.7	3.9	3.0	2.6	4.4	5.1	5.4	6.0	5.7	6.0	6.0	4.9
(16) Solar Energy Delivered to Load $\bar{Q}_H$	56.4	44.7	31.6	21.1	18.8	31.4	31.4	30.4	35.2	37.5	39.6	38.1	34.68
(17) Electrical Energy Displaced by Solar-"Bottom Line" (8) - (5)	53.5	46.0	31.6	22.4	19.3	31.2	32.1	30.9	36.2	39.7	43.0	37.1	32.25 3270 KWH/yr.
(18) Current $ Value of (17)(per day) @ $.05/KWH	0.74	0.64	0.44	0.31	0.27	0.43	0.45	0.43	0.50	0.55	0.60	0.52	0.448 $163/yr

SOLAR WATER HEATER DEMONSTRATION

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ABSTRACT

A demonstration of six commercially available solar water heaters has been erected by the Alabama Solar Energy Center for the purpose of developing public awareness of commercialized technology and acquiring factual performance and operational data. The six systems are installed side-by-side on a typical roof structure and provide two examples each of silicone oil, antifreeze and drain-back freeze protection. Each system is instrumented with BTU and KWH meters to assess performance under an imposed F-CHART load profile. This paper describes the systems under evaluation, the instrumentation and results acquired over a recent three week period.

INTRODUCTION

The public has shown a general unawareness of commercialized solar technology and where that awareness exists, it is accompanied by an apprehension regarding technical readiness and ability to provide savings sufficient to justify the high initial cost. These impressions are fostered by the wide publicity being given to Federal research and development programs directed at advancing the more exotic solar technologies. These concerns of the consumer can be removed by an education process and systematic evaluation of commercialized solar equipment performance.

A display of six commercially available solar water heaters has been erected in the Alabama Solar Energy Center's Solar Test Facility for the purpose of developing public awareness of commercialized technology and acquiring factual performance and operational data. The display incorporates into a typical residential roof structure, two examples of each of the three freeze protection types of solar water heaters being marketed in Northern Alabama, e.g., oil, antifreeze, and drain-back. Each supplier was allowed to configure and install his system exactly as he might for a typical customer's application.

The systems are sized to meet a residential hot water requirement for a family consisting of three members, e.g., 228 lpd (60 gpd). A timer and cam device is used to operate a dump valve for various time intervals, thereby, imposing on each system the hot water demand profile used to generate F-CHART.

Each system is instrumented with a BTU meter and two KWH meters to evaluate system performance. A 114 l (30 gal) electric water heater is used for comparison relative to a conventional system. System efficiencies are determined using radiation data from a pyranometer mounted in the collector plane. Performance and operational data will be analyzed monthly and reported through various public information channels.

In this paper the details of the facility, systems and instrumentation will be described in detail. Operational problems experienced with the systems and instrumentation will be discussed, and performance data will be presented for the short period of time the facility has been operational.

DESCRIPTION OF FACILITY

Overall Description An external view of the Solar Water Heater Display is presented in Fig. 1. A framed roof structure was constructed alongside a 3.7m X 17.1m (12ft X 56ft) mobile home which houses the tanks, controls, etc. The 3.7m X 12m (12ft X 40 ft) roof is oriented toward magnetic South (which varies from true South by less than 1^{o}). The roof is sloped 35^{o} which is approximately the latitude of the Huntsville site (34^{o} 4').

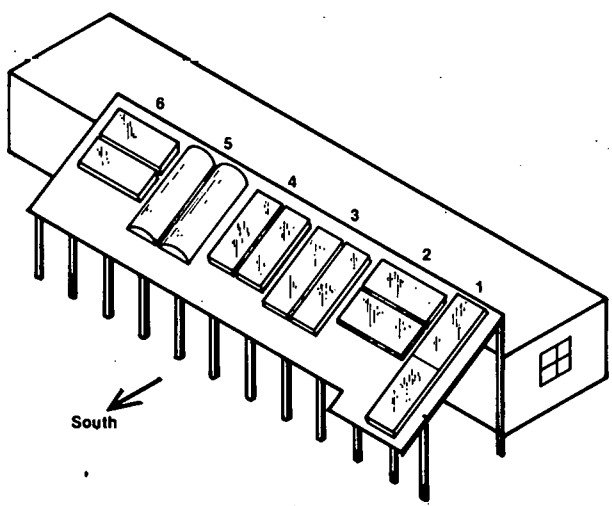


Figure 1 External View of Solar Water Heater Display

Figure 2 is a plan view of the facility illustrating the arrangement of the system elements. All piping has been color coded to assist in describing solar system function to the visiting laymen. A schematic is displayed alongside each system and the subsystems are labeled for easy identification.

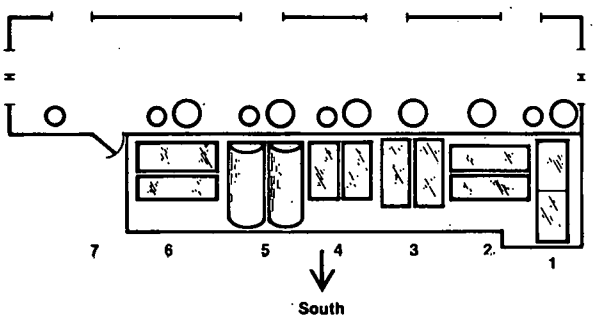


Figure 2 Plan View of Solar Water Heater Display

The cold water supply is provided from the utility service and the hot water is presently being dumped to waste. A cam device was fabricated to impose an hourly DHW water flow according to the RAND profile used to develop F-CHART. After installation each system was carefully balanced to achieve a constant flow rate of 1.9 lpm (0.5 gpm). The desired profile is achieved by varying the draw interval during each hour. The total daily flow is about 228 lpd (60 gpd) for each system. To impose the same thermal load on each system, the supplemental heater thermostats were adjusted to provide a 60°C (140°F) outlet temperature.

Description of DHW Systems The only systems being widely marketed in the North Alabama area are pumped closed types having various tank (single or double tank) and heat exchanger arrangements. Because the freeze protection mechanism probably has the single greatest influence on cost, performance, reliability and durability, the decision was made to select systems according to freeze protection, with the tank and heat exchanger configuration as a second consideration. Table 1 presents the various systems configurations and a lettering code which will be used throughout this paper as an identifier. With the collectors mounted side-by-side on the same roof, the tanks and other equipment exposed to the same environmental conditions, and the same DHW load imposed on each system, the variations in performance should be solely due to the variations in equipment design and control functions. In the following paragraphs, the details of each system will be described more fully.

TABLE 1 DHW SYSTEM CONFIGURATIONS

System	Code*	Freeze Protection	H-X Types	Tanks
1	SCD	Silicone Fluid	Counterflow	Double
2	SIS	Silicone Fluid	Internal Coil	Single
3	DWS	Drain-back	Wrap-around	Single
4	DID	Drain-back	Internal Coil	Double
5	GWD	Glycol	Wrap-around	Double
6	GCD	Glycol	Counterflow	Double
7	-	-	-	Single

*The first letter refers to the freeze protection type (S-Silicone, D-Drain-back, G-Glycol), the second letter to the H-X type (C-Counterflow, I-Internal Coil, W-Wrap around) and the third letter to the tank configuration (D-Double, S-Single).

The supplier of each system was given the option of installing his own system or to have it installed by the Alabama Solar Energy Center. Systems #1, #3 and #6 were installed by the Center according to manufacturer's instructions. Before they were put into operation a final checkout was performed by the supplier.

System #1, the Silicone, Counterflow H-X, Double Tank (SCD) system is manufactured by Solar Unlimited, Inc. of Huntsville, Alabama. The collector utilizes a vertical aluminum finned absorber with copper tubes painted flat black. A single glazing of tempered low iron diffusing glass is used to cover the box which is riveted aluminum and insulated with 5.1 cm (2 in) of fiberglass. The gross area of the collector is 3.62 m^2 (38.94 ft^2). The collector is mounted flat on the roof and flashed at the top. Collector piping is enclosed in the upper flashing so that it is not exposed to the weather.

The pre-packaged insulated heat exchanger assembly contains the heat exchanger, two circulating pumps and controls. The all copper heat exchanger is a finned tube coiled inside a shell to permit counterflow between the water in the tube and the silicone oil in the shell. The tube is double wall and externally vented. The solar storage is a 304 l (80

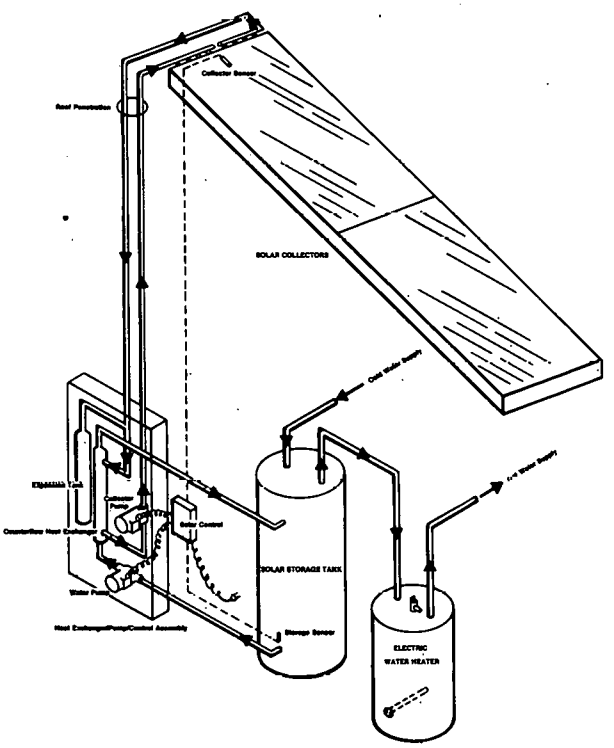


Figure 3 System #1, Silicone, Counterflow H-X, Double Tank (SCD)

) glass lined A. O. Smith tank with R 10 figlass insulation. The two circulation pumps are made by Grundfos and the differential temperature controller by Honeywell. A complete schematic of the system is shown in Fig. 3. The conventional tank supplying supplemental energy is a 114 l (30 gal) Jackson Water Heater which will be described later as System #7.

System #2, the Silicone, Internal Coil, Single Tank (SIS) system is distributed by the Alabama Solar Energy Center, Inc. of Birmingham, Alabama. The collector is manufactured by General Energy Devices and has a gross area of 5.33 m^2 (57.4 ft^2) for two panels.

Figure 4 illustrates the system which employs an aluminum roll bond absorber coated with a flat back polyurethane paint. The box is cast fiberglass with epoxy binder. The double galzing is re-inforced fiberglass plastic by Kalwall. Polyurethane foam 2.54 cm (1 in) thick is used to insulate the back and sides. The collectors are mounted to a square tube which elevates it 2.54 cm (1 in) above the roof.

The solar storage is a stone lined 304 l (80 gal) Sepco tank containing a finned copper coil heat exchanger. Fiberglass insulation provides a resistance value of 16. The tank contains two 4.5 kW elements for supplemental energy. The differential thermostat is manufactured by Hawthorne and the pump by Grundfos. The heat transfer fluid is silicone oil.

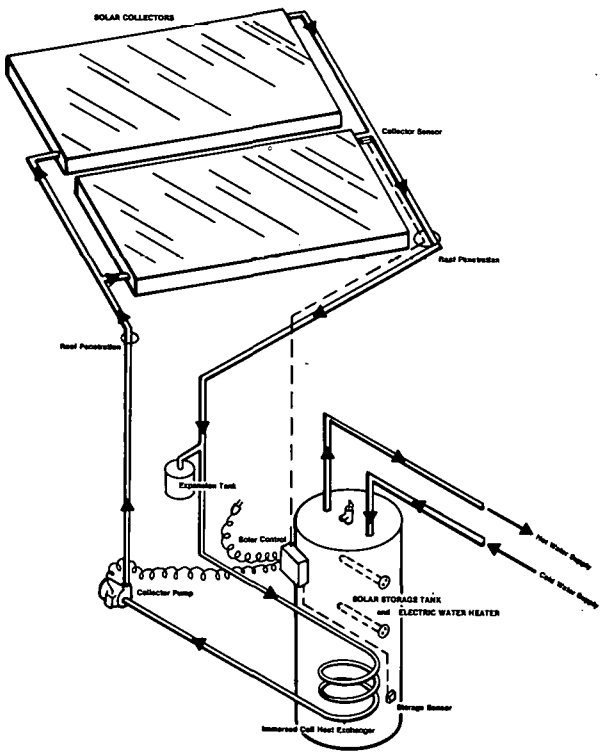


Figure 4 System #2, Silicone, Internal Coil, Single Tank (SIS)

System #3, the Drain-back, Wrap-around H-X, Single Tank (DWS) system is manufactured by Energy Converters of Chattanooga, Tenn. It is also sold by Jackson Water Heater Company. It uses a black painted aluminum roll-bond absorber in a painted steel box. The insulation is 7.62 cm (3 in) of fiberglass and a single covering of tempered low iron diffusing glass. The gross area of the two collectors is 4.59 m^2 (49.4 ft^2). The collector is mounted on pads to elevate it slightly off the roof.

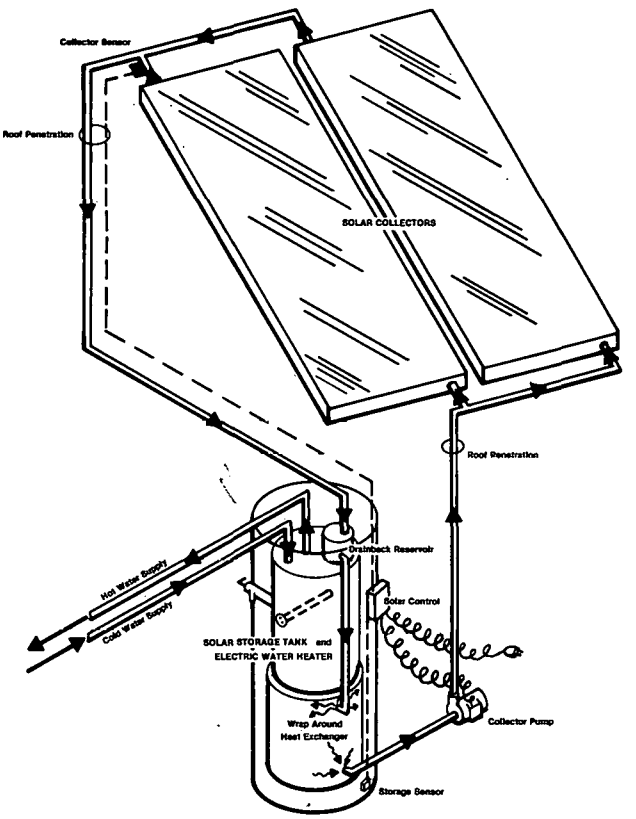


Figure 5 System #3, Drain-back, Wrap-around H-X, Single Tank (DWS)

The system schematic is shown in Fig. 5. It features a 304 l (80 gal) glasslined tank with an integral 6.0kW supplemental heater element and an aluminum wrap-around heat exchanger. This system is unusual in that it is the only one that uses CPVC piping in the collector loop. The heat transfer fluid is distilled water with corrosion inhibitor added. The March pump employs a plastic impeller thereby minimizing the corrosion potential of metal. On cessation of pump operations, the heat transfer fluid drains back into a sump mounted on top of the storage tank. The differential thermostat is made by Energy Converters.

System #4, the Drain-back, Internal Coil H-X, Double Tank (DID) system is manufactured by Aircraftsman of Millbrook, Alabama. The absorber is fabricated from copper tubes which penetrate vertical aluminum fins painted flat black. The collector box is extruded aluminum and the single glazing is tempered low iron diffusing glass. The collector insulation is 5.08 cm (2 in) of polyurethane foam and the gross collector area is 3.21 m^2 (34.5 ft^2).

The system, portrayed in Fig. 6 features a 304 l (80 gal) stone lined Ford tank containing an immersed coil heat exchanger. The drain-back sump is a standard 19 l (5 gal) water heater and distilled water is used as the heat transfer fluid. The collector is mounted flat on the roof and the pipes are covered with flashing at the top and bottom of the collector. The pump is made by Grundfos and the differential thermostat by Heliotrope General.

System #5, the Glycol, Wrap-around H-X, Double Tank (GWD) system is a Grumman system supplied by National Air Conditioning and Refrigeration Supply Company of Huntsville, Alabama. The collector uses an absorber formed from copper tubes clamped into flat aluminum fins and coated with flat black paint. The single glazing is acrylic curved to provide rigidity and reduce reflection losses. The aluminum box contains 3.81 cm (1½ in) of fiberglass and is mounted 5.1 cm (2 in) above the roof. Gross area of the two collectors is 5.81 m^2 (62.5 ft^2).

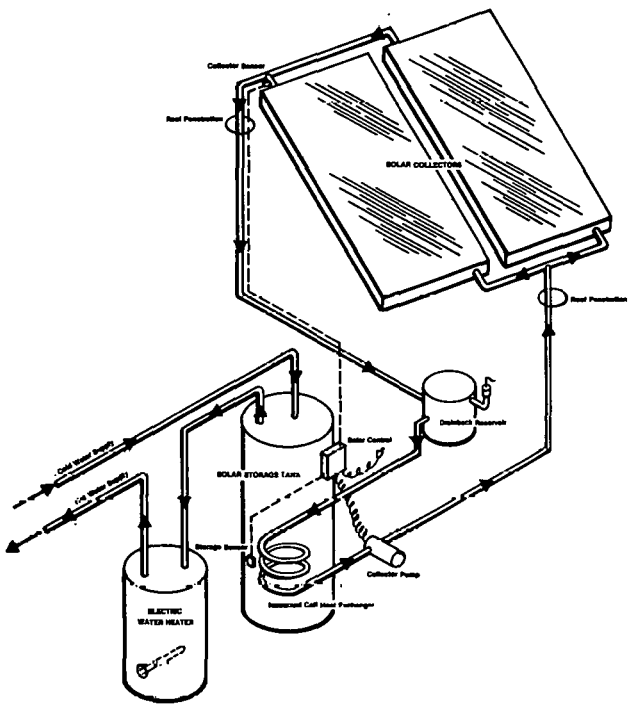


Figure 6 System #4, Drain-back, Internal Coil H-X, Double Tank (DID)

The system, illustrated in Fig. 7, features a glass-lined 304 l (80 gal) tank with the Wrap-around heat exchanger. The collector fluid is a proprietary formulation based upon ethylene glycol. The proportional differential thermostat is made by Rho Sigma and the pump by Grundfos.

System #6, the Glycol, Counterflow H-X, Double Tank (GCD) system is distributed by Jared, Walsh and Associates of Birmingham, Alabama. The collector is manufactured by Solara in Israel and distributed by National Energy Corporation. The absorber is fabricated from copper tubes soldered to a steel plate and coated with a flat black paint. The collector back and sides are insulated with 5.08 cm (2 in) of fiberglass and the box is galvanized steel. Gross area of the two collectors is 3.93 m^2 (42.3 ft^2). The collectors are mounted 5.08 cm (2 in) above the roof.

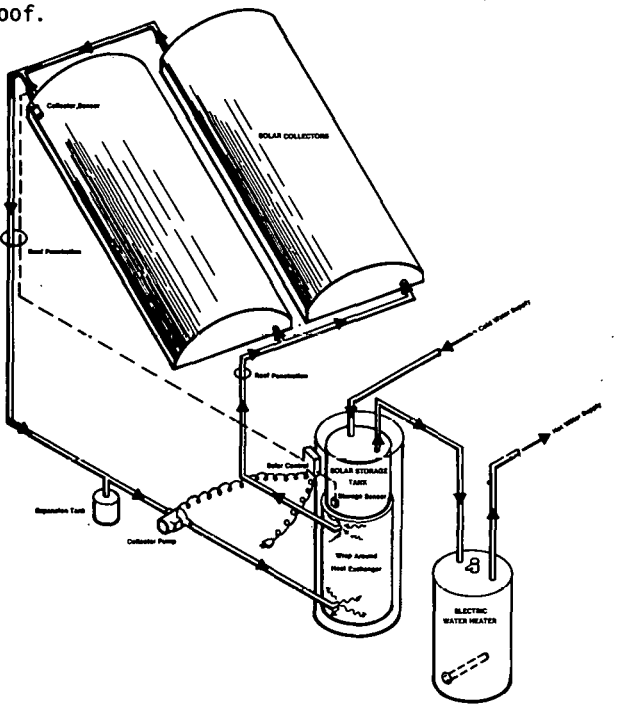


Figure 7 System #5, Glycol, Wrap-around H-X, Double Tank (GWD)

The system, portrayed in Fig. 8 features a Taco heat exchanger module. This pre-packaged unit contains a counterflow tube-in-tube heat exchanger. two circulating pumps, controls, etc. The heat transfer fluid is propolyne glycol and a 304 l (80 gal) A.O. Smith glasslined tank is used for solar storage. The differential thermostat is made by Independent Energy.

System #7, the Conventional Water Heater is a conventional 114 l (30 gal) water heater manufactured by Jackson Water Heater Co. It contains about 3.81 cm (1½ in) of insulation and one 2 kW element located at the bottom of the tank to provide supplemental energy.

Description of Instrumentation The primary objective of the instrumentation is to evaluate solar system performance over extended time intervals for known environmental conditions. A secondary objective was to evaluate thermodynamic behavior through periodic observation of critical temperatures within each system.

The specific measurements used to meet these objectives are:

- Energy Delivered to Load
- Supplemental Power Consumed
- Parasitic Power Consumed
- Cold Water Supply Temperature
- Hot Water Supply Temperature
- Solar Storage Tank Temperature
- Hot Water Consumption
- Incident Solar Radiation

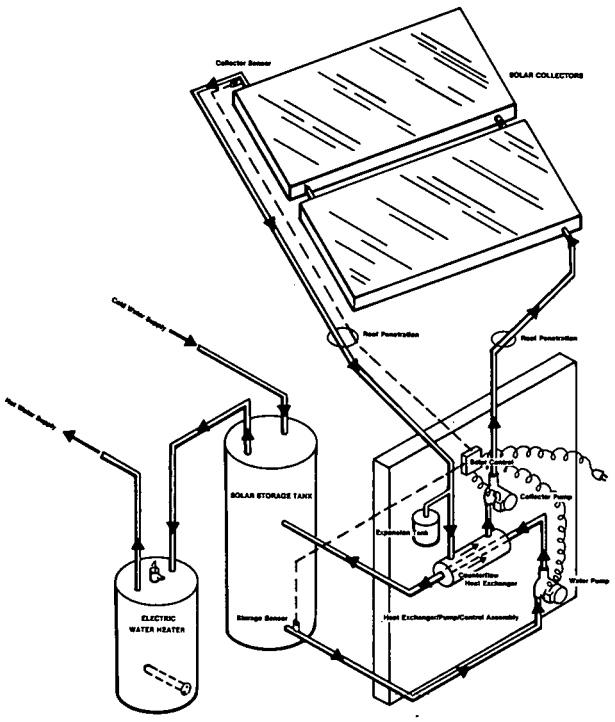


Figure 8 System #6, Glycol, Counterflow H-X, Double Tank (GCD)

A Wilgood Btu meter is used to simultaneously measure hot water consumption and the cumulative energy delivered to load over an extended period of time. The meter consists of a water flow sensor located on the cold water supply, two temperature sensors, electronic computer and registers for indicating totalized values. To assure maximum accuracy each meter was calibrated by Wilgood prior to shipment and the flow rate accuracy was found to be within ½% at 1.9 lpm (0.5 gpm). The hot temperature sensor is installed in the hot water supply as close to the tank as possible. The cold sensor is located within the flowmeter body.

Standard residential utility billing meters are used to measure the supplemental energy supplied through the auxiliary heater element and the parasitic energy consumed by the controls and pumps. The meters, manufactured by Duncan, were calibrated to within 1½ % by the Huntsville Utilities. A three wire meter was used to measure the consumption of the 240V heater element and a two wire meter was used for the 120V pumps and controls.

Temperatures are being measured by dial type thermometers mounted in thermowells. Prior to installation each was calibrated between the boiling and freezing points of water. The scale divisions permit no better reading accuracy than about 1.1°C (2°F).

Environmental parameters are being measured by the State Climatic Center which is located in the same test facility as the solar water heater display. Eppley pyranometers are used to measure total radiation in the horizontal plane. Other climatic parameters being continously monitored include ambient y bulb temperature, ambient wet bulb temperature, nd speed, wind direction and barametric pressure.

EXPERIMENTAL RESULTS

Operational Problems During initial facility checkout, several operational problems arose with both the operation of the solar systems and the instrumentation, affecting both solar system performance and valid collection of data. After the solar water heating systems were installed, six control malfunctions were experienced with four of the six systems. These control malfunctions required the replacement of four differential controls under warranty to eleviate the problems. The malfunctions, the control's associated time in service and repair actions taken for the control problems are listed in Table 2.

TABLE 2 SUMMARY OF CONTROL MALFUNCTIONS

Control Malfunction	Problem Identification	Time In Service	Repair Action
ON at night high current	relay chatter	3 months	controller replaced
ON at night	switch improperly factory wired on replacement control	2 days	switch rewired
OFF in bright sun	cold solder junction on power terminal	3 months	terminal resoldered
OFF in bright sun	control circuit failure	3 months	control board replaced
ON at night	control circuit failure	3 months	controller replaced
Proportional controller doesn't turn completely OFF	Controller malfunction inconsistant sometimes, OFF/ON	3 months	NONE

All repairs and replacement were made under warranty and since the repairs have been made the systems have been operating properly without problems. The proportional controller on system #5, (GWD) slowed the pump down to a stop but there was still sufficient current flow to heat the motor windings and create fluid thermosyphon up to the collector sensor. These losses were insufficient in magnitude to be indicated by the power meters. Cold nighttime temperatures in the fall have temporarily eliminated this phenomenon.

It is important to note, however, that none of the control malfunctions entirely stopped the system operation or generated obvious noises or other indication that the systems were malfunctioning. A final inspection immediately after installation had indicated that controls had been properly installed and were functioning as required. Because of the subtle nature of the malfunction they would have been unnoticed by a homeowner indefinitely, and probably even by an experienced technician because all malfunction indications were apparent only at night.

For simplicity and appearance the Btu meters were

installed in the cold water supply line to each system near the ceiling and above the tanks. After the systems had warmed up, it was discovered that the cold water supply line and the Btu meter were warm on some of the systems. This was due to both thermosyphoning and thermal expansion of the water up the cold water supply line as the tank heated up. All of the cold water supply lines were then retrofitted with an expansion tank and a U tube thermostatic trap to minimize these effects while still keeping the expanded water within the system and its associated instrumentation. The common cold water supply was also retrofitted with a check valve to prevent back flow of the water due to expansion. The preferrable arrangement for future monitoring projects would locate the Btu meter near the floor in the cold water line. A check valve and properly sized expansion tank would be installed between the Btu meter and the solar storage tank.

Measured Parameters The performance evaluation of a solar energy system requires the measurement of key energy flows and the inference of others through application of the first law of thermodynamics. At regular intervals (initially at sunset each day) readings are taken and recorded from the cumulating meters display as described in Description of Instrumentation. The following parameters are recorded along with the date and time of each reading.

I_t = total daily incident solar radiation per unit area on the plane of the collector, kJ/M^2 (Btu/h/ft^2)

E = cumulative total parasitic electrical power required to operate the solar system pumps and controls, kJ (Btu)

Q_{SHE_S} = cumulative total supplemental electrical power required to operate the supplemental heating element, kJ (Btu)

Q_{LOAD} = cumulative total heat delivered in hot water to the load, kJ (Btu)

W = cumulative total hot water delivered to the load, liters (gallons)

In order to fully evaluate the thermal operating characteristics of each solar system and the conventional water heater it was necessary to measure the characteristic heat loss from the conventional water heater portion of each system.

Q_{Loss} = characteristic heat loss from the water heater portion of solar system

This was accomplished by allowing all the water heaters to heat up to their thermostatic set point, closing the hot water draw valve, and recording the reading of the supplemental power used by each system for two days without the solar system operating.

Although the four supplemental water heaters and the conventional water heater were the same make, model, and age, contained the same wattage element and were wired identically, their characteristic heat loss varied by up to 50%. The water heaters were carefully inspected for insulation gaps or water leaks and the elements were replaced and the heat loss test repeated with the same results. After finding these efforts had no effect on the characteristic heat loss of the water heaters the local electic distributor was consulted. The local electric distributors, who had been monitoring electrical use in several water heaters for two years, reported that the variation in thermostat operation had resulted in electrical consumption variation of ± 50%. To minimize the effect on this variation in water heater heat loss on the validity of this comparative test, the energy savings resulting from each solar system will be calculated two ways. One way using the characteristic heat loss of the conventional water heater in the control system and one way using the heat loss from the water heater in the system under study. This will be discussed further under derived performance devices.

Derived Performance Indices A heat balance on the water heater portion of a solar water heating system gives the following relationships:

$$\text{ENERGY IN} = \text{ENERGY OUT}$$

$$Q_u + Q_{SHE} = Q_{Loss} + Q_{Load} \quad (1)$$

Therefore, the useful heat added by the solar system can be calculated by subtraction:

$$Q_u = Q_{Loss} + Q_{Load} - Q_{SHE} \quad (2)$$

This relationship is valid if the characteristic heat loss of the system (Q_{Loss}) remains constant over time.

From the measured values of parasitic power, (E) supplemental power (Q_{SHE}) and heat load (Q_{Load}) and the calculated value of useful heat delivered by the solar system (Q_u) the following solar performance parameter can be derived:

$$\eta_s = \frac{Q_u - E}{A_c I_t} \quad \text{System Overall Efficiency} \quad (3)$$

$$\eta_{st} = \frac{Q_u}{A_c I_t} \quad \text{Systems Thermal Efficiency} \quad (4)$$

$$SCOP = \frac{Q_u}{E} \quad \text{Solar Coefficient of Performance} \quad (5)$$

Energy savings are the difference in energy used by the solar system and a conventional water heater to provide the same hot water load. The average savings in non-renewable energy resources over a specified period by the solar water heating system, (Q_H) is given below:

$$Q_H = Q_{SHE_C} - Q_{SHE_S} + E \quad (6)$$

Where $_C$ and $_S$ refer to the conventional and solar systems respectively, the supplemental energy for the conventional system, Q_{SHE_C}, is taken to be either the consumption of the conventional water heater (System #7), or the energy consumption of the solor system water heater delivering the same load with the solar portion of the system inactive.

Continuing with the definition of other parameters:

$$f = \frac{Q_u - E}{Q_{Load}} \quad \text{Solar Load Fraction} \qquad (7)$$

$$FES = \frac{Q_H}{Q_{SHE_c}} = 1 - \frac{Q_{SHE_s} + E}{Q_{SHE_c}} \quad \text{Fractional Energy Savings} \qquad (8)$$

Again because of the variation in the heat loss of the water heaters the fractional energy savings must be calculated two ways, using the conventional water heater and the solar system water heater (with the solar collection system OFF).

The total purchased energy, E + Q_{SHE}, is presented in Figure 9 for the six solar systems and the conventional water heater. The relatively high consumption of systems #2 and #6 is probably due to flow restrictors placed in the collector loop to achieve the flow rate "recommended" by the collector manufacturer. Interestingly, one of the silicone oil systems, #1, consumes the least amount of total energy, but the greatest amount of parasitic energy.

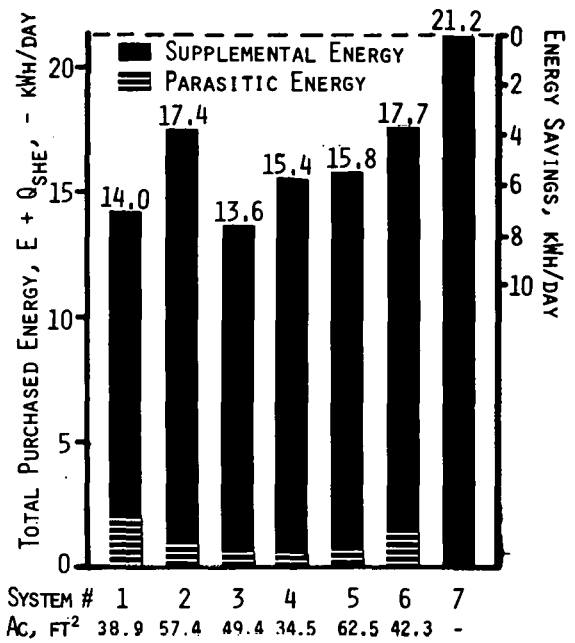


Figure 9 Average Total Daily Purchased Energy for Each System

Figure 10, presents the fractional energy savings as computed from Equation 8 using the supplemental energy requirements, Q_{SHEc}, of different "conventional" systems as a basis for comparison. Because the heat loss, Q_{Loss}, from the conventional components of the solar systems were generally less than System #7 (see Table 3), solar performance measured on the former basis is less (note the negative term in Equation 8). Again the two systems with the flow restrictors show the poorest performance (systems #2 and #6). The system with the largest collector area is one of the poorest performers in terms of energy savings (system #5) and the two with the

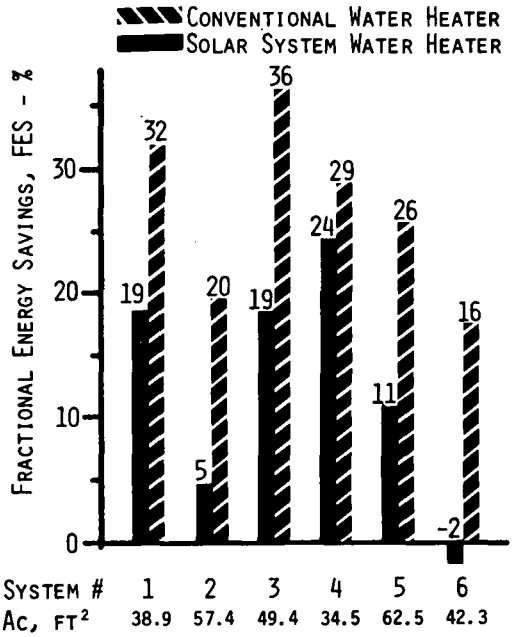


Figure 10 Fractional Energy Savings of Each System Compared to Conventional and Solar System Water Heater

smallest collector area (system #1 and #4) are the best performers. This same pattern is evident when the systems are compared on a solar load fraction basis, Figure 11. Finally, the system efficiencies, as computed from Equations 3 and 4 are presented in Figure 12. The high efficiency attained by system #4 may be somewhat misleading because the system heat loss, Q_{Loss}, used to compute the useful heat Q_u, is considerably higher than for the other system (see again Table 3).

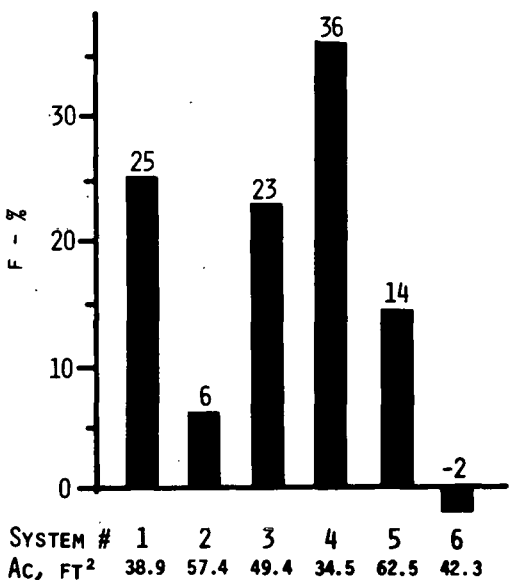


Figure 11 Average Solar Load Fraction

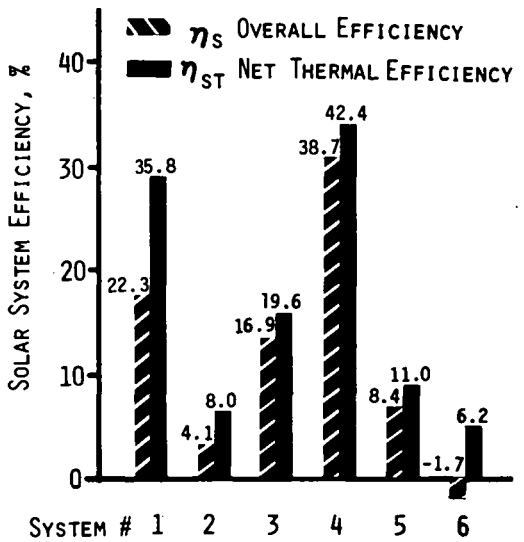


Figure 12 Average Solar System Efficiency

TABLE 3 APPARENT HEAT LOSS

System #	1	2	3	4	5	6	7
Q_{Loss}, kWh/da	4.46	4.46	3.21	6.51	4.34	4.08	6.32 (7.82)[1]

[1] measured over 20 day interval

Session III:
Solar Space Heating Systems

RESULTS OF PERFORMANCE ANALYSIS OF PASSIVE SYSTEMS IN THE NATIONAL SOLAR DATA NETWORK*

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ABSTRACT

The National Solar Data Network, as a part of the National Solar Heating and Cooling Demonstration Program, provides a means for the collection and analysis of data representing the thermal performance of selected instrumented solar energy systems. Among the systems for which data is being collected and analyzed are those systems utilizing passive techniques for space heating. This paper presents an overview of the thermal performance results for selected instrumented passive systems in the National Solar Data Program. System thermal performance is illustrated both in terms of energy utilization and savings and in terms of thermal comfort. Solar energy system characteristics which contribute to the success of the systems are discussed.

INTRODUCTION

This paper presents results obtained through the National Solar Data Network, as part of the government's solar demonstration activity, as they apply to passive space heating systems. Analysis of system thermal performance was accomplished using an energy balance technique adapted for use in the National Solar Data Program** [1], [2].

SYSTEMS EVALUATED

The following paragraphs present a brief description of each system evaluated. Additional system description data is contained in the references.

Hullco Construction: The Hullco Construction system [3], [4], [5] is a combination sunspace and mass wall system used for heating a single-family residence in Prescott, Arizona. The 1,056 square foot, south-facing building is a combination greenhouse and direct gain system. Solar energy enters the building through double-glazed pre-fabricated Kalwall panels. Sliding glass doors between the greenhouse and the living area admit solar energy directly into the master bedroom and living room. Energy collected is stored directly in the massive walls and floors of the building or indirectly by fans in the under-floor rock beds in the north half of the building. Auxiliary energy is supplied by either radiant electric heat panels or the wood stove located in the living room.

Greenmoss Builders, Inc.: The Greenmoss Builders, Inc. [6] solar energy system located in Waitsfield, Vermont is a mass wall heating system in a single-family residence with approximately 1,000 square feet of living space. The passive space heating system consists of a large vertical window on the south side of the two-story building which allows direct solar irradiation of a concrete block storage wall. Auxiliary space heat is provided by a standard gas-fired furnace. Summer overheating protection is provided by venting the space between the block wall and the window.

Living Systems: The Living Systems passive solar energy system [7] provides direct gain heating to an approximately 1,700 square foot single-family dwelling in Davis, California. Incident solar energy is admitted to the building through both the large south-facing window wall and the overheat skylight. Solar energy storage is provided by blue painted water filled tubes containing nearly 2,500 gallons placed near the south window wall and under the skylight. Additional storage is provided by the concrete slab floor of the building. All glass surfaces are double-glazed with minimum window area in non-south facing walls. Manually operated insulating curtains which cover the south window wall, along with manually operated insulating shutters for the skylight glazing, are used to reduce nighttime heat losses. Auxiliary space heating energy is provided by a gas-fired furnace and a wood burning stove.

Spence-Urban: The Spence-Urban and Associates site [8] is a single-family, 2,418 square foot residence located in West Des Moines, Iowa. The active and passive systems are designed to satisfy approximately 50 percent of the domestic hot water and space heating demands. The passive direct gain solar space heating system utilizes two glazed areas on the south side of the building for energy collection. The air in both the living room and the greenhouse is heated directly by solar radiation incident on the glazed areas. Energy storage for the passive system is provided

*Work performed under contract to the U.S. Department of Energy.

**Numbers in brackets [] denote reference number.

by the floor of the two rooms and by a brick wall inside the house just north of the living room. Distribution of collected solar energy is accomplished by natural convection and through the active system return air ducts.

Kalwall Corporation: The Kalwall Corporation solar energy system [9], [10], [11], [12] is a direct gain passive space heating system. The heated space is a portion of a commercial warehouse in Manchester, New Hampshire. Solar energy is admitted through vertical double-glazed Kalwall panels covering the entire south and east walls. Storage is provided by the concrete floor of the building and the contents of the warehouse. Summer overheat protection is provided by both an overhang on the south roof edge and by natural ventilation. Auxiliary energy is provided by two liquid-to-air heat exchangers heated by the building boiler. Five thermostatically controlled circulating fans located along the south wall assist in distribution of the collected solar energy by operating when the temperature near the fan falls below 60°F or rises above approximately 90°F. This system is designed to maintain a lower temperature than in residential applications.

Colorado Sunworks: The Colorado Sunworks system [13], [14], [15] is a direct gain passive system used for space heating of a single-family dwelling in Longmont, Colorado. The building is a three bedroom single-story house with approximately 1,800 square feet of living space. The passive space heating system is a combination drum wall and direct gain system. Sunlight enters the double-glazed windows on the south side of the building where the majority of the energy is absorbed by the black painted water filled drums. The remainder of the energy is either absorbed in the concrete slab floor or used to satisfy the daytime space heating demand. The exterior insulated reinforced concrete building walls also serve as a secondary solar storage mass. At night, or during periods of low incident solar energy, heat losses through the glazing are reduced by movable insulation in the form of a Beadwall*. Auxiliary space heating is provided by either natural gas-fired hydronic baseboard units or by a wood burning fireplace.

RESULTS

Measurement data is available for some of the systems evaluated for more than two years. For other systems, data has been available for only a few months. However, in each case, data reflecting system operation during a substantial portion of a heating season is available and has been used as a basis for both monthly and seasonal thermal performance evaluations and for special papers [16]. The results presented in this paper represent a summary of the information gathered over several heating seasons. The reader is referred to the list of references for more detailed information about these systems - both for system descriptions and for details of daily and monthly thermal performance. Primary performance indices used for thermal performance evaluations of the systems include energy savings and a measure of the comfort conditions obtained.

Energy savings refer to the difference in the energy requirements of the building with its glazed south areas replaced by a conventional insulated wall area, and the non-solar energy actually used. The performance index used to evaluate thermal comfort conditions is the operative temperature, which is defined as the average of the space dry bulb temperature and the space mean radiant temperature. The moisture level inside the building also effects comfort, but is presented as a separate parameter. No attempt is made in this paper to quantity the economic benefits of the systems. This data is presented in other papers.

System thermal performance is best illustrated by a comparison of both the energy use and the energy savings. Presented in Figure 1 for four of the systems, is the solar utilization efficiency. The solar utilization efficiency is the ratio of the solar energy used for space heating in the building to the total solar energy incident on the glazed south-facing surfaces. The energy used includes the thermal losses through the glazing. During the coldest winter months, this efficiency for all of the systems is in the range of 50 to 55 percent indicating that if the glazing losses are included in a design estimate of the load, that a somewhat conservative estimate of solar utilization against that load of 50 percent would be appropriate. The solar utilization efficiency is less during other months due primarily to increased transmission losses at the glazing. The rapid drop in efficiency shown for the Kalwall system in the spring is due to increased energy availability and collection on the glazed east wall.

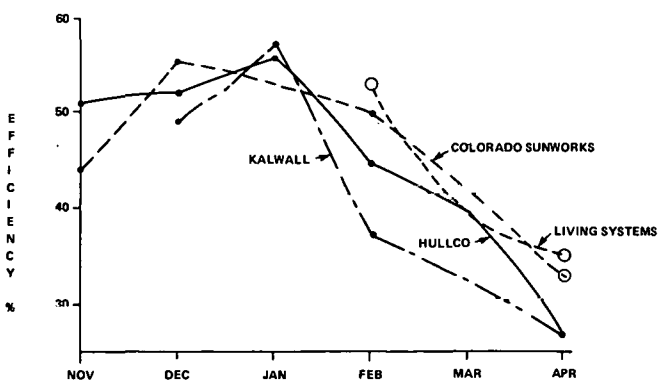


Figure 1. Monthly Solar Use Efficiency

In terms of energy savings the results do not correlate as well as those presented in Figure 1. Shown in Figure 2 is a savings efficiency for the same sites used in Figure 1. The savings efficiency is the ratio of the energy savings to the total incident solar energy. The data shown in Figure 2 reflects the differences in system perfor

*Beadwall® is a registered trademark of the Zomeworks Corporation, Albuquerque, NM.

nance due to different loss characteristics through the glazed areas. Systems which use some form of movable insulation to reduce glazing losses such as the Colorado Sunworks system are ten to fifteen percent more efficient in terms of the savings efficiency, thus illustrating the benefits of movable insulation assemblies. Even with the glazing losses removed, it can be seen from the data presented on Figure 2 that well designed passive space heating systems are able to save energy equivalent to 25 percent or more of the total incident solar energy.

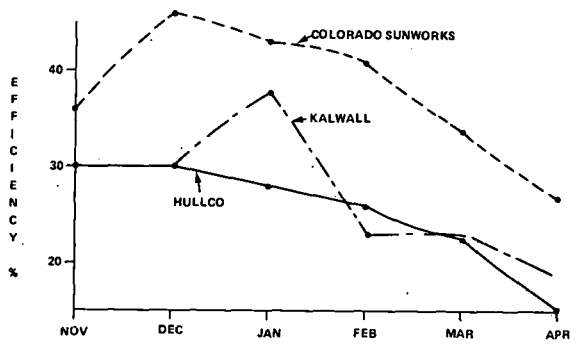


Figure 2. Monthly Savings Efficiency

A logical next step in this energy use comparison would be to compare the savings in energy to a building model where building temperatures are controlled to within a narrow range of temperature, thus examining the change in energy savings resulting from daily temperature variations which occur in passive solar structures. This was accomplished on several systems with the result that very little change in energy savings has been observed since daily temperature variations typically averaged no more than five to seven degrees F. Thus it was concluded that daily building temperature variations had little effect on the magnitudes of energy savings.

Another representation of energy use and savings commonly used for system comparisons is the use or savings normalized to the total collector area. Table 1 shows this data for four of the systems for six months. The energy use was consistently about 400 Btu per square foot per day while the energy savings were about 300 Btu per square foot per day in each case. The high value of energy use on the Hullco system was due to the high amount of available solar energy during the winter and the consequent warm temperatures in the greenhouse. The warm greenhouse temperatures caused an increase in solar energy use principally to satisfy the thermal losses through the greenhouse glazing. These higher losses are also reflected in the highest load as normalized to both the weather and the size of the building. The load data for the Kalwall system is low because of the fact that the Kalwall system is bordered on two sides by other conditioned spaces. The energy savings data is the highest for the Colorado Sunworks system because of

TABLE 1. ENERGY USE AND SAVINGS PER UNIT COLLECTION AREA

SITE	LOAD (BTU/FT^2-DAY)	ENERGY USE (BTU/FT^2-DAY)	ENERGY SAVINGS (BTU/FT^2-DAY)	AVAILABLE SOLAR (BTU/FT^2-DAY)
KALWALL	4.42	424	300	1023
HULLCO	11.1	716	407	1346
LIVING SYSTEMS	7.2	480	445	1140
COLORADO SUNWORKS	3.81	629	517	1127

the effectiveness of the building insulation and the Beadwall movable insulation assembly. Note that both the weather conditions and the effectiveness of movable insulation assemblies (if used) all affect the normalized values of use and savings when presented in this manner.

As a system to system comparative performance index, the measure of comfort conditions is somewhat difficult to use. A comfort measure on any scale merely refers to the average of comfort condition preferences of a large number of people. Since in any sample of building occupants, a relatively wide variation in comfort preferences may be encountered, then it is more appropriate to use a "yes/no" evaluation of comfort conditions when making comparative thermal performance evaluations. This evaluation should consider not only the preferences of the occupants, but also the design use of the building in a system such as the Kalwall system. Against these rating criteria, all of the systems evaluated performed satisfactorily by providing environments acceptable to the occupants of the buildings. The only exception is the summer performance of the Kalwall system where temperatures occasionally reached highs near 90°F due to energy collected through the glazed east wall.

Within a passively heated structure, comfort conditions are evaluated several ways. First the daily average of the building conditions is examined. Second, the change in comfort conditions between the south and north side of the building is analyzed. Finally, the day to night variations in comfort conditions are evaluated. When the passive solar contribution to the building heating demand is small, these factors are of little importance since the auxiliary system maintains acceptable comfort conditions inside the building. However, when conditions are such that the passive contribution is significant, then comfort criteria become more important.

Since the performance of the passive systems evaluated was generally acceptable with respect to the comfort criteria, data illustrating representative system comfort condition performance will be presented for the Colorado Sunworks System.

In the Colorado Sunworks system differences in the comfort index value between the two building zones were generally quite small, (Figure 3) averaging 2 to 3 degrees F over each month. This low difference is due to the capability of the system to effectively transfer collected solar energy from the south side of the building to the rooms on the north side. As illustrated in Figure 3 comfort levels in both zones of the building were near 70°F in all months except January, when very severe weather conditions coupled with the desires of the occupants to not use auxiliary energy caused the substantial reductions in the comfort index values. The larger difference in comfort index values observed in February is due to the near depletion of usable storage energy which occurred in January.

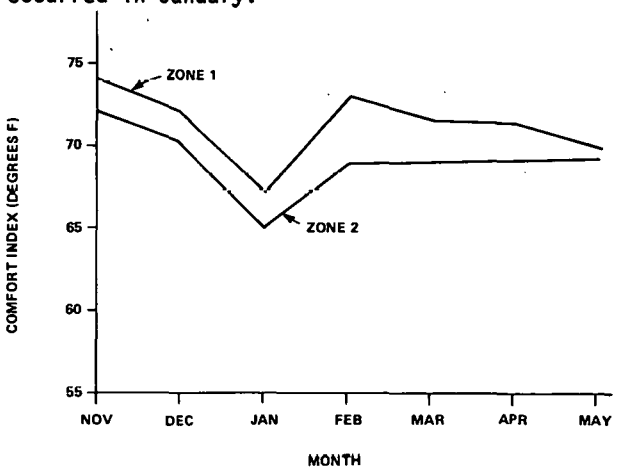


Figure 3. Colorado Sunworks Average Comfort Index

Daily variations in comfort conditions presented for zone 1 (Figure 4) and zone 2 (Figure 5) were within acceptable limits. The average change of 5°F shown for zone 1 is slightly higher than the average variation seen in zone 2. The larger variation in zone 1 is due to the proximity of the zone 1 areas to both the solar gains and higher losses of the glazing system.

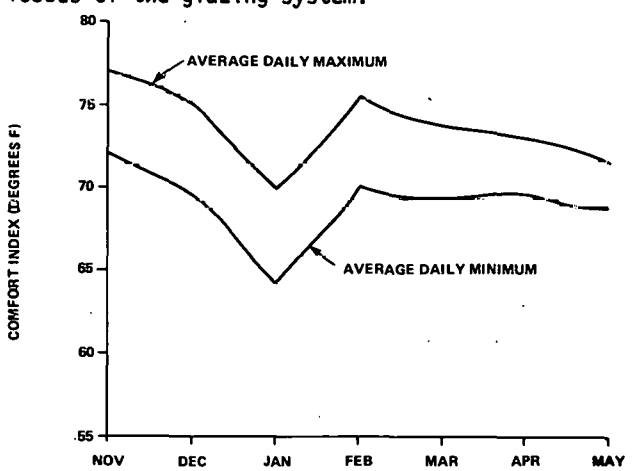


Figure 4. Colorado Sunworks Zone 1 Minimum and Maximum Comfort Index

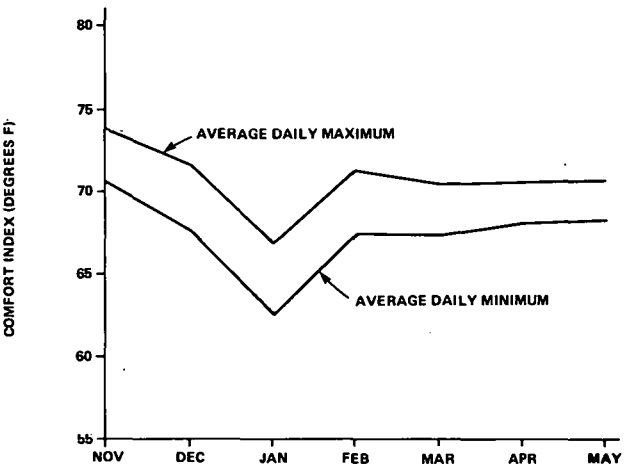


Figure 5. Colorado Sunworks Zone 2 Minimum and Maximum Comfort Level

OBSERVATIONS

Even though satisfactory thermal performance has been observed on most of the systems evaluated, there are some common problems. These problem areas have been observed on several of the monitored systems and as such deserve special mention.

First, adequate system thermal storage should be provided. In addition to providing reserve heating on cloudy days, the storage mass must be capable of providing enough storage capacity to prevent conditioned space overheating on warm, sunny days. Overheating is prevented as the storage mass is able to absorb the majority of the collected solar energy without significant temperature increases as illustrated by the performance shown in Figure 6. Otherwise wider variations in building temperature such as those shown in Figure 7 can be expected.

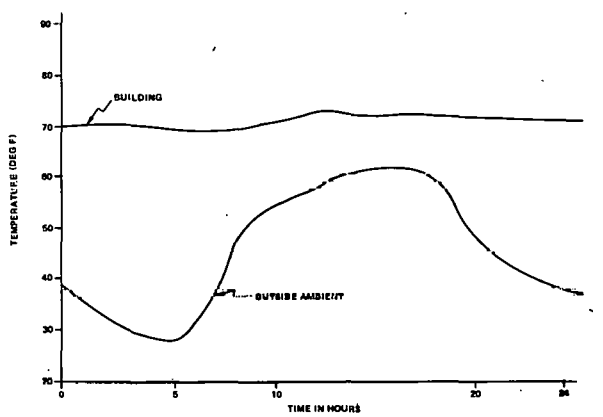


Figure 6. Colorado Sunworks Building and Outside Ambient Temperatures - May 8, 1979

Many designers use some east or west facing window area in an attempt to get some additional solar gains in the morning and afternoon hours. During winter months, particularly in northern latitudes

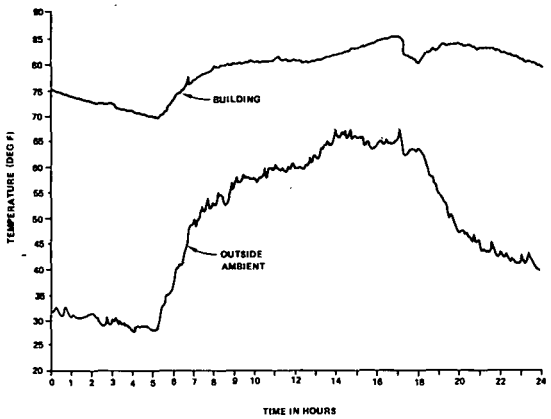


Figure 7. Greenmoss Building and Outside Ambient Temperatures - May 7, 1979

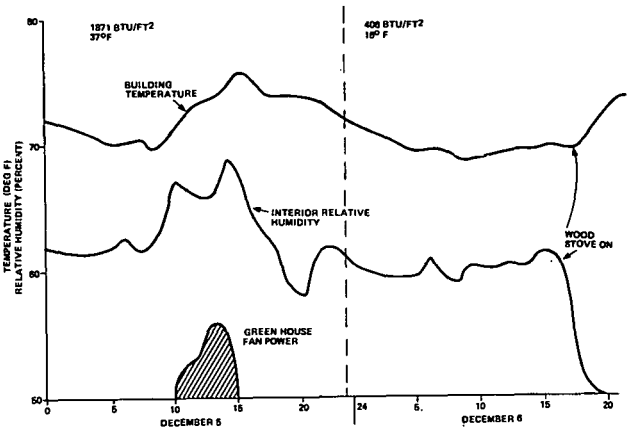


Figure 8. Hullco System Relative Humidity Effects

only a small amount of energy is incident on east and west walls when compared to the south wall, since the sun traverses a low path in the southern sky. In northern United States latitudes during spring and fall months, more than half of the energy incident per square foot is available on the east or west wall. Consequently, systems which use east or west windows for collection could be expected to receive substantial gain from them in spring and fall months. Energy gain through west walls may produce undesirable results. Since building temperatures resulting from south wall solar energy collection generally reach a maximum in mid-afternoon, the effect of additional solar energy collection during the afternoon can be seen in overheating of rooms containing west-facing windows.

Provisions must be made for the control of moisture inside the conditioned space, particularly in greenhouse or sunspace systems. Since a passive structure may be more tightly sealed than a conventional structure, moisture accumulation inside the building may be more of a problem. Interior relative humidity problems at the Hullco building, associated with the greenhouse, were encountered. When a greenhouse is used, additional moisture may accumulate in the greenhouse area due to transpiration from the plants and from water used for the plants. Consequently, provision must be made to prevent the direct circulation of air from a functional greenhouse area to the rest of the house. As illustrated in Figure 8, a definite correlation existed between the operation of the greenhouse fans and increases in interior relative humidity, since the air returned to the greenhouse through the conditioned space. However, other systems, such as the Colorado Sunworks Systems, (Figure 9) do not have problems with excessive moisture inside the house even though they may be tightly sealed as illustrated by the lack of change of interior relative humidity with wind speed changes.

The Greenmoss system, located in an area with a winter climate characterized by very cold winter nights and long sequences of cloudy weather, requires some form of night time insulation. The

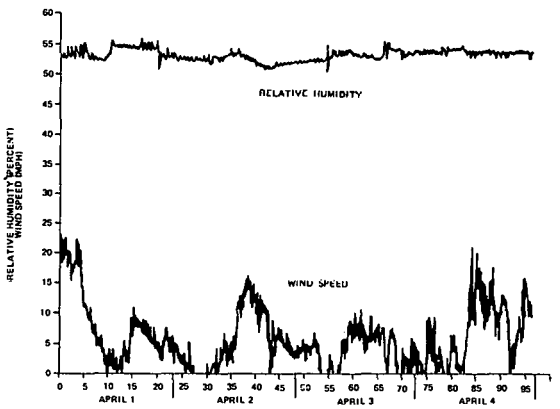


Figure 9. Colorado Sunworks Interior Relative Humidity and Wind Speed

location of the Greenmoss storage wall near to the glazing area has resulted in high thermal losses from the wall area through the glazing. Hourly average storage temperatures have dropped below 40 degrees F during both the winter of 1977-1978 and the winter of 1978-1979. As illustrated in Figure 10, an effect of the high losses through the uninsulated glazed area was a net loss of energy when compared to a conventional building over the winter of 1977-1978. However, as further illustrated in Figure 10, if insulation with a net additional insulating value of R-2 were used only at night, the effect would be to reduce the consumption of auxiliary energy to a point where positive energy savings could be realized.

In some cases, controlled active systems are used to transfer energy within the conditioned space. Controls for this technique, like the south wall circulating fans at the Kalwall system, must be exercised so that gain from the active component operation will offset the power used by the fans. As illustrated in Figure 11, the fans at the Kalwall site operated the most during the winter when available energy in system storage was at minimum values. Consequently, little gain in energy dis-

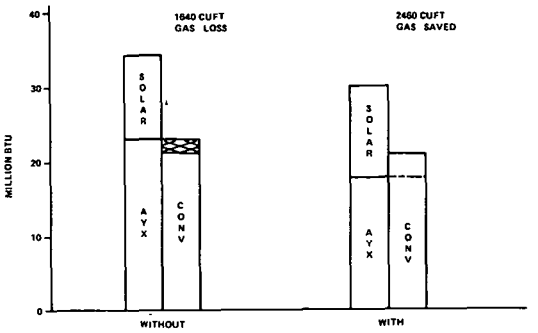


Figure 10. Glazing Insulation Energy Savings - Heating Season

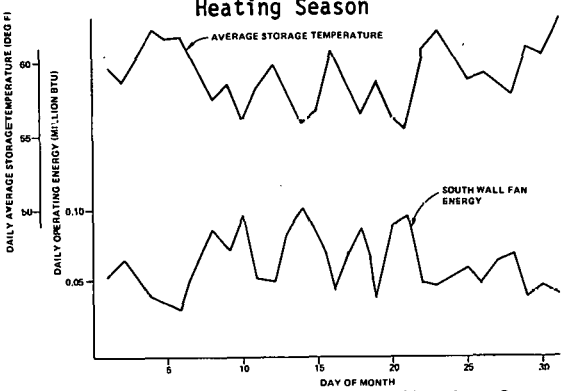


Figure 11. Kalwall Energy Distribution System - January 1978

tribution was realized from operation of the fans, since the usable energy in storage was nearly depleted.

SUMMARY

Through the National Solar Data Network, successful performance of passive solar space heating systems has been demonstrated. Energy savings greater than 300 Btu per square foot of glazed area per day have been observed while comfort conditions acceptable to the occupant have been maintained. Summer overheating can occur but is controllable by proper application of system controls. Some common problems have been observed. However, these common problems have caused only inconveniences rather than substantial degredation of system thermal performance and should serve as lessons learned when considering future passive system designs.

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16. "Results of Thermal Performance Analysis of Passive Space Heating Systems in the National Solar Data Network," M. W. Weston, July 1979. Presented at the U.S. Department of Energy Regional Update Conferences, 1979.

*Copies of these reports may be obtained from Technical Information Center, P.O. Box 62, Oak Ridge, Tennesses 37830.

RESULTS OF PERFORMANCE ANALYSIS OF PASSIVE SYSTEMS IN THE NATIONAL SOLAR DATA NETWORK

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IBM Corporation

Appendix 1 Supporting Data

Presented in this section are some tables which provide supporting data for the paper. The tables presented are for the Hullco, Colorado Sunworks, and Kalwall Systems. The data presented in Tables 1 through 4 reflects the summer and winter performance of the Hullco and Colorado Sunworks Systems. Tables 1 and 3 present energy use and savings data while Tables 2 and 4 present data relative to the comfort conditions maintained. The comfort conditions presentation shows both the average comfort index and the average of the daily minimum and maximum values for the entire building along with the south side of the building (zone 1) and the north side of the building (zone 2). The comfort index used is the average of the space dry bulb and mean radiant temperatures. Interior relative humidity values are presented separately. Presented in Tables 5 and 6 is data for the Kalwall System showing winter energy use (Table 5) and summer building temperatures (Table 6). Comfort values for the Kalwall System are not presented since the building is used as a warehouse and is normally not occupied. Further explanation of the data is contained in some of the sixteen reference papers, particularly those which present seasonal performance evaluations of the systems.

For the Hullco and Colorado Sunworks Systems, three different values of energy savings are presented. The building energy savings is the energy savings for the building as built and as operated. The buildings savings does include the excess energy losses through the glazed areas. The comparison energy savings is the energy savings compared to a model of the passive structure where the glazed south areas have been replaced by insulated walls with thermal characteristics similiar to the other building walls. Finally, the comparison set point savings is the energy savings compared to the comparison building where internal temperatures have not been allowed to vary but are controlled to within a set point range.

TABLE 1

ENERGY SAVINGS - HULLCO CONSTRUCTION

Month (1)	Space Heating Demand (Million Btu)	Auxiliary Electric (Million Btu)	Auxiliary Wood Thermal (Million Btu)	Building Energy Savings (Million Btu)	Building Energy Savings (KWH)	Comparison Energy Savings (Million Btu)	Comparison Energy Savings (KWH)	Comparison Set-Point Savings (Million Btu)	Comparison Set-Point Savings (KWH)	Greenhouse Fan Energy (Million Btu)
May 78	N/A	N/A	N/A	0	0	0	0	0	0	0
June 78	N/A	N/A	N/A	-0.04	-11	-0.04	-11	-0.04	-11	0.04
Aug 78	N/A	N/A	N/A	-0.21	-61	-0.21	-61	-0.21	-61	0.21
Sept 78	N/A	N/A	N/A	-0.07	-20	-0.07	-20	-0.07	-20	0.07
Oct 78	5.95	0.0	0.0	5.78	1,693	3.27	958	0	0	0.17
Nov 78	9.66	0.02	0.89	9.48	2,777	5.58	1,634	3.69	1,081	0.18
Dec 78	10.53	0.02	3.27	10.43	3,056	6.03	1,767	5.49	1,608	0.08
Jan 79	8.32	0.0	5.47	8.31	2,435	4.19	1,227	4.04	1,183	0.00
Feb 79	9.19	0.0	2.29	9.10	2,667	5.30	1,554	4.65	1,363	0.09
Mar 79	8.36	0.0	1.39	8.25	2,417	4.76	1,395	4.17	1,221	0.10
Apr 79	6.43	0.0	0.20	6.30	1,845	3.63	1,063	1.31	383	0.13
Season	58.44	0.04	13.51	57.33	16,798	32.44	9,506	23.03	6,747	1.07

(1) Data not available for July 1978

TABLE 2

COMFORT LEVELS - HULLCO CONSTRUCTION

Month (1)	Building Temperature (°F)			Zone 1 Comfort Index (°F)			Zone 2 Comfort Index (°F)			Interior Relative Humidity (Percent)
	Average	Average Daily Minimum	Average Daily Maximum	Average	Average Daily Minimum	Average Daily Maximum	Average	Average Daily Minimum	Average Daily Maximum	
May 78	75	78	78	75	73	77	73	71	76	35
June 78	79	77	83	78	77	81	78	76	82	21
Aug 78	80	78	83	80	78	81	79	76	82	32
Sept 78	79	76	80	79	77	80	77	74	79	33
Oct 78	78	76	80	79	77	81	76	74	78	33
Nov 78	76	74	78	77	75	80	73	71	75	55
Dec 78	72	69	75	74	71	77	68	66	72	54
Jan 79	72	69	76	73	71	76	69	66	72	47
Feb 79	72	69	76	74	71	77	70	67	73	49
Mar 79	71	68	74	73	71	75	69	66	72	53
Apr 79	72	70	76	74	72	76	70	67	73	55

(1) Data not available for July 1978

TABLE 3

ENERGY SAVINGS - COLORADO SUNWORKS

Month	Space Heating Demand (Million Btu)	Space Heating Load (Million Btu)	Fireplace Energy (Million Btu)	Fireplace Energy (Cu.Ft. Gas)	Building Energy Savings (Million Btu)	Building Energy Savings (Cu.Ft. Gas(1))	Comparison Energy Savings (Million Btu)	Comparison Energy Savings (Cu.Ft. Gas(1))	Comparison Set-Point Energy Savings (Million Btu)	Comparison Set-Point Energy Savings (Cu.Ft. Gas(1))	DHW Energy Savings (Million Btu)	DHW Energy Savings (Cu.Ft. Gas(1))	Beadwall Operating Energy (Million Btu)	Beadwall Operating Energy (kwh)
Nov 78	4.99	6.90	0.40	476	8.29	9,869	6.86	8,166	5.25	6,250	0.68	809	0.14	41
Dec 78	8.25	10.15	0.14	167	13.75	16,369	11.45	13,630	10.57	12,583	0.93	1,107	0.18	53
Jan 79	7.35	9.55	0.55	655	12.26	14,595	9.95	11,845	9.14	10,880	0.91	1,083	0.18	53
Feb 79	6.23	7.69	0.04	47	10.39	12,369	8.50	10,119	7.64	9,095	0.98	1,166	0.15	44
Mar 79	4.37	6.03	0.23	273	7.28	8,667	5.95	7,083	4.92	5,857	0.85	1,012	0.12	35
Apr 79	3.03	4.44	0.02	23	5.05	6,011	4.10	4,880	3.42	4,071	0.94	1,119	0.14	41
May 79	1.71	3.36	0.13	214	2.74	3,262	2.01	2,932	1.91	2,273	0.72	857	0.10	29
Season Total	35.93	48.12	1.56	1,855	59.76	71,142	48.82	58,655	42.85	51,009	6.01	7,153	1.01	296

(1) assumes 840 Btu per cubic foot

TABLE 4

BUILDING COMFORT LEVELS - COLORADO SUNWORKS

Month	Building Temperature (°F)			Zone 1 Comfort Index (°F)			Zone 2 Comfort Index (°F)			Storage Temperature (°F)			Interior Relative Humidity (Percent)
	Average	Average Daily Minimum	Average Daily Maximum	Average	Average Daily Minimum	Average Daily Maximum	Average	Average Daily Minimum	Average Daily Maximum	Average	Average Daily Minimum	Average Daily Maximum	
Nov 78	73	70.5	75.8	74	72.1	77.0	72	70.3	73.9	77.0	74.9	79.2	34
Dec 78	71	67.9	73.9	72	69.7	75.2	70	67.8	71.8	75.1	73.1	77.6	41
Jan 79	66	62.4	68.8	67	64.1	70.0	65	62.5	66.7	69.8	67.5	72.2	39
Feb 79	71	67.9	73.8	73	70.3	75.6	69	67.5	71.3	75.6	73.6	77.7	44
Mar 79	70	67.7	72.5	72	69.5	73.8	69	67.4	70.5	73.6	71.8	75.4	50
Apr 79	70	68.3	72.4	72	69.7	73.2	69	68.0	70.5	72.8	71.6	74.0	48
May 79	70	68.1	71.7	70	68.7	71.6	69	68.2	70.7	70.7	69.8	71.5	48

TABLE 5

MONTHLY HEATING PERFORMANCE - KALWALL SYSTEM

Month	Solar Energy Incident (Million Btu)	Heating System Load (Million Btu)	Solar Fraction of Load (Percent)	Collection Efficiency (Percent)	Average Storage Temp. (Deg. F)	Solar System Operating Energy (Million Btu)	Auxiliary System Operating Energy (Million Btu)	Average Build Temp. (Deg. F)
Nov 77	24.7	23.7	58	46	65	0.544	0.019	64
Dec 77	42.1	36.2	56	49	59	2.065	0.544	60
Jan 78	41.2	40.5	43	48	59	1.965	0.688	61
Feb 78	61.8	42.7	59	44	62	1.263	0.308	64
Mar 78	58.4	36.1	64	40	62	1.155	0.045	64
Apr 78	38.4	21.4	77	44	63	0.688	0.004	64
Season	266.6	200.7	58	45	62	7.78	1.618	63

TABLE 6

SYSTEM THERMAL PERFORMANCE - COOLING SEASON - KALWALL SYSTEM

Month	Ambient Temperature Degrees F	Building Temperature Degrees F	6 PM Average Building Temperature Degrees F	6 AM Average Building Temperature Degrees F	Storage Temperature Degrees F
May 78	61	69	72	66	69
Jun 78	69	76	78	73	75
Jul 78	74	81	83	79	82
Aug 78	73	80	78	82	81

NOTES

PERFORMANCE DATA EVALUATION OF THE BALCOMB SOLAR HOME (SI UNITS)*

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ABSTRACT

Heating season performance data are developed for the Balcomb solar home for the period November 1, 1978 through April 24, 1979. The home is a two-story attached solar greenhouse with a total area of 214 m^2 located in Santa Fe, New Mexico. The measured loss coefficient for the house is in good agreement with the predicted value. The useful solar heat averaged over the season is 301 MJ/day corresponding to 88.3% of the net load and 37.4% of the solar radiation incident on the glazing. The adobe mass wall is effective primarily as heat storage for the greenhouse but is less effective than expected in heating the living areas. The fan-forced rock bed is quite effective, having an overall COP of 11. Energies transported by convection, though not measured directly, are determined by indirect techniques and are found to be quite significant. The need for instrumentation to better understand convective energy flows is discussed.

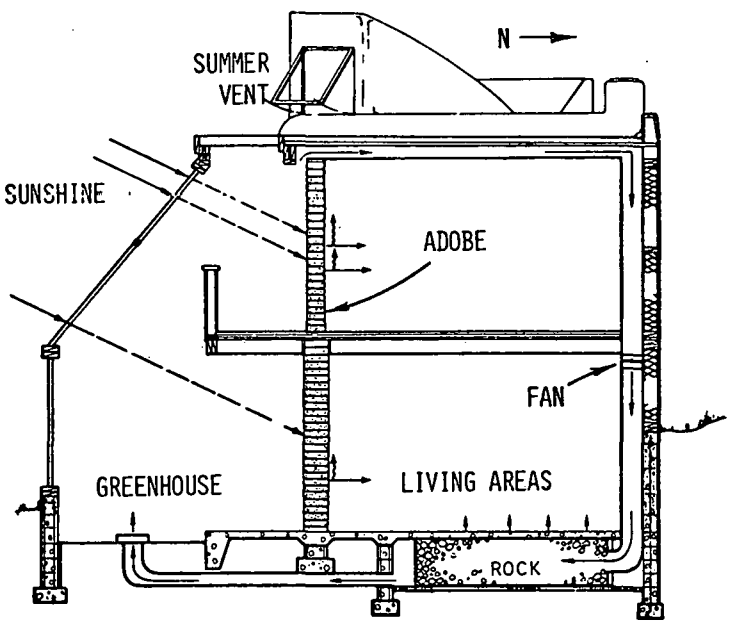
BUILDING DESCRIPTION

The Balcomb home is Unit 1 of an all-solar sub-division of eight homes located in a 3300°C-day climate of northern New Mexico. The 182 m^2 living area is primarily heated by sun captured in a 32 m^2 attached greenhouse. These two areas comprise two very distinct thermal zones separated by an adobe mass wall. All external walls are either bermed or of well insulated frame construction and all glazing is of double glass. The overall predicted heat loss coefficients are 239 W/°C and 200 W/°C for the living area and the greenhouse respectively, accounting for 1/3 ACH and 1 ACH for the respective infiltration losses. The corresponding interzone coefficient is 184 W/°C accounting for 1 ACH between living areas and greenhouse.

The greenhouse glazing, which is oriented 11° west of true south, totals 24.3 m^2 at a 50° slope and 12.7 m^2 vertical. Living area windows total 16.3 m^2 distributed about equally around NE, SE, SW, and NW orientations.

The house contains one active element, a system of air ducts, two fans, and two rock beds located under the living area of the house. Charging of the rock beds is active by blowing warm air from the greenhouse peak through the rock beds and returning to the greenhouse. Distribution of the heat from the rock beds to the living area is passive, by conduction up through a concrete floor slab.

The home was designed and built by Susan and Wayne Nichols.



*Work performed under the auspices of the U.S. Department of Energy, Office of Solar Applications.
Copies of this paper in British units can be obtained from the LASL Solar Group.

INSTRUMENTATION AND DATA ACQUISITION

Total global horizontal solar radiation is measured with a photovoltaic type pyronometer. Wind direction and velocity are measured. Total, auxiliary heating, water heating, and fan electric use is measured using clip-on current meters calibrated to the house main and auxiliary watt hour meters. Thirty temperatures are measured using iron/constantan thermocouples.

Data acquisition and recording is by an Autodata 9 which writes a one-week record of hourly measurements on a magnetic tape cassette. Current readings, solar radiation, outside temperature, and wood stove flue temperature are averaged over hour intervals based on about 240 samples.

ANALYSIS

Permanent data storage is on a magnetic disk and data analysis is performed using expanded Hewlett Packard 9845 calculators.

Although data have been taken for a one and a half years, the interval selected for analysis is the 176-day period from November 1, 1978 through April 24, 1979. There are 26 days in this interval in which there is more than one hour of missing data, generally being associated with the exchange of data tapes. These days, which are scattered throughout the interval, are dropped entirely from the analysis and the average data from the remaining days of the month are substituted into these days for plotting purposes.

Solar radiation gains are calculated hourly through each of six glazing orientations. Of the total solar gain 19.3% is through "non-solar" windows in the living areas. This slightly exceeds the losses through these windows. An experimental correlation was developed which relates the wood stove output to the flue gas temperature. Flow in the air ducts is determined from a detailed one-time scan of velocities across the duct exits using a pitot-tube measurement of velocity head.

The results are 22 m^3/min and 27 m^3/min for the east and west ducts respectively. Heat transferred to the rock bed is calculated using the (flow rate) x (ΔT) integral when the fan is on. Heat fluxes on both sides of the adobe wall are calculated using measured values of the surface temperatures and a six-node solution of the diffusion equation using the following adobe properties: ρc_p = 1206 KJ/°C-m^3 and k = 0.66 W/°C-m. People heat is estimated as 132 W.

The house loss coefficient is determined from the 20-day time period from December 7-26 during which there are no missing-data days or other complications. The overall value is in good agreement with the values calculated using ASHRAE steady-state techniques. This measured loss coefficient is then apportioned between the living areas and the greenhouse in the same ratio as the calculated values. The results are 222 W/°C and 185 W/°C, respectively.

Two major heat flow terms were not measured directly and had to be inferred. These are heat vented from the house by opening windows and doors during warmer periods of the winter and heat released from a woodburning corner fireplace. The times when these occur are quite discrete and are strongly evidenced by a large unbalance in the house energy in excess of what could be explained by thermal storage within the house. It would be much preferable to have a direct measurement of these energy flows but in lieu of such measurement the estimates have been made based on "backing out" the vented and fireplace energy from the other measured values assuming that the integral of the thermal unbalance is bounded. The times when either fireplace or vented energy is indicated by this technique are in good agreement with the climate at these times and the observed way in which the house was operated.

SUMMARY DATA

A summary of the 176-day winter season energy flow is given in Fig. 1, below, and month-by-month summaries of key values are given in Table I on the following page.

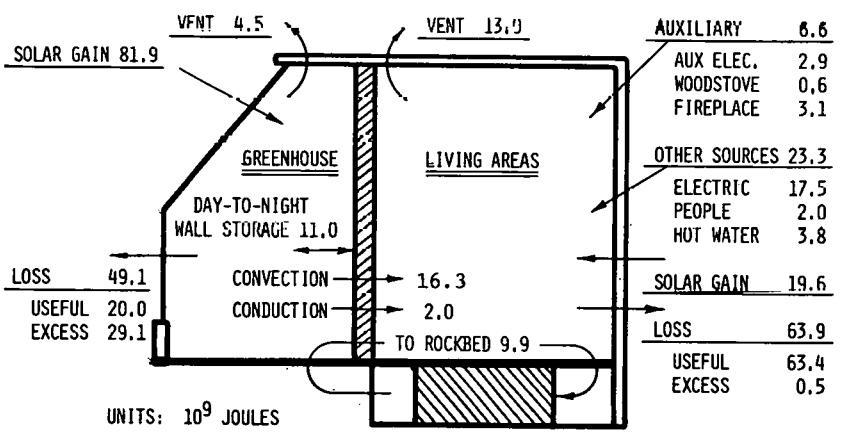


Fig. 1

Total energy flows for the 176-day period from Nov. 1, 1978 through Apr. 24, 1979.

SAMPLE OF HOURLY DATA

Fig. 2 below is a sample of data taken in the Balcomb home during a period of severe but mixed weather. The previous several days had been clear and repetitive so that the house is in a quasi-steady condition. The total measured electrical energy used in the house during this 13-day period is 810 kWh. Of this 299 kWh was for water heating and 82 kWh was for auxiliary baseboard electric heating, used primarily in the dining-kitchen area. The fireplace was used occasionally in the evening. The large spikes in the total electrical power consumption are due to primarily to the water heater which draws 4 kW. During this time period the solar water heater was down for modification. The recorder was off during the data gap in the center. The average outside temperature over this time period is -5.2°C and the calculated heat load, in the absence of all solar and internal sources, is 1982 kWh, using an overall loss coefficient of 314 W/°C (based on considering the greenhouse as an unheated space.)

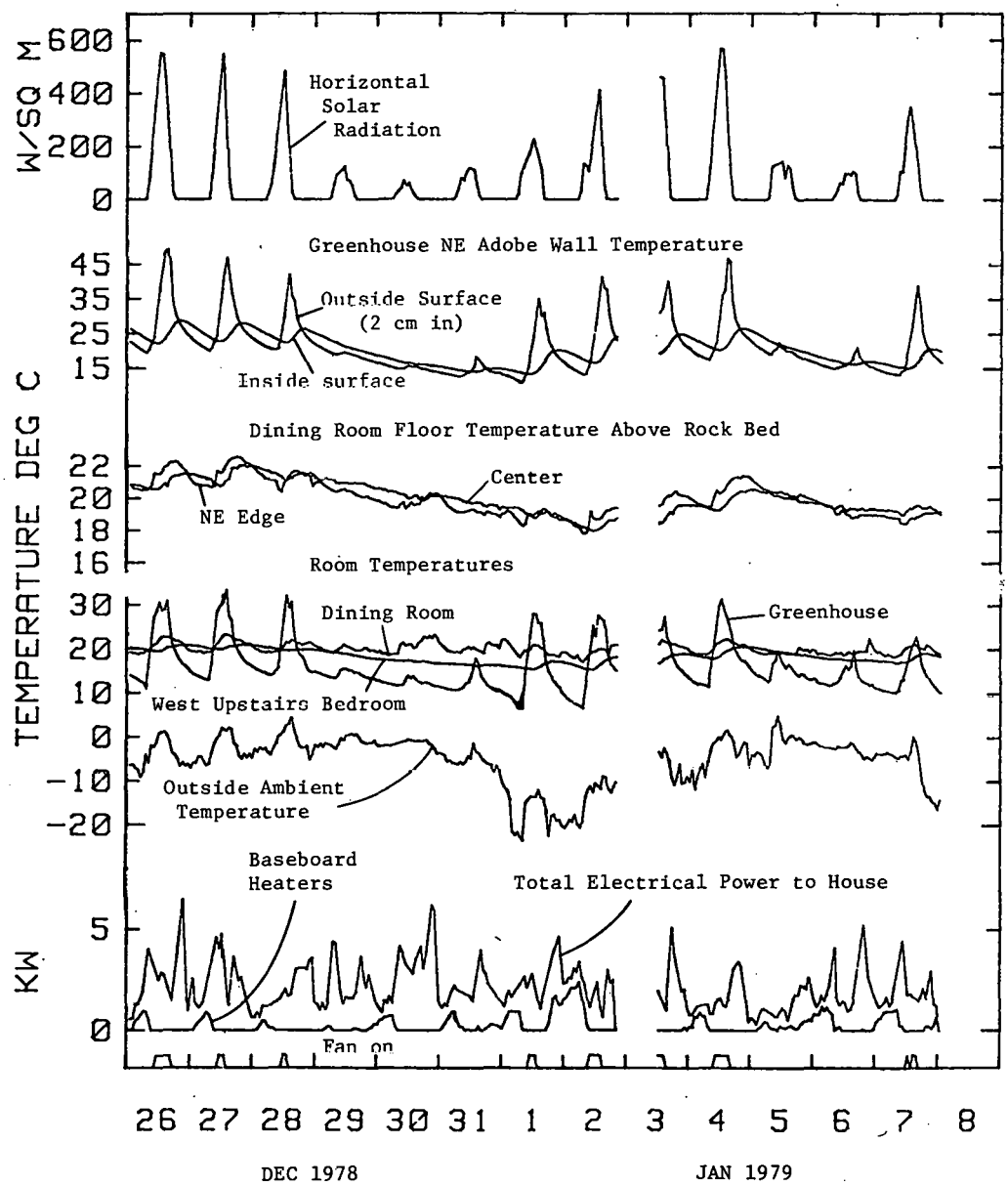
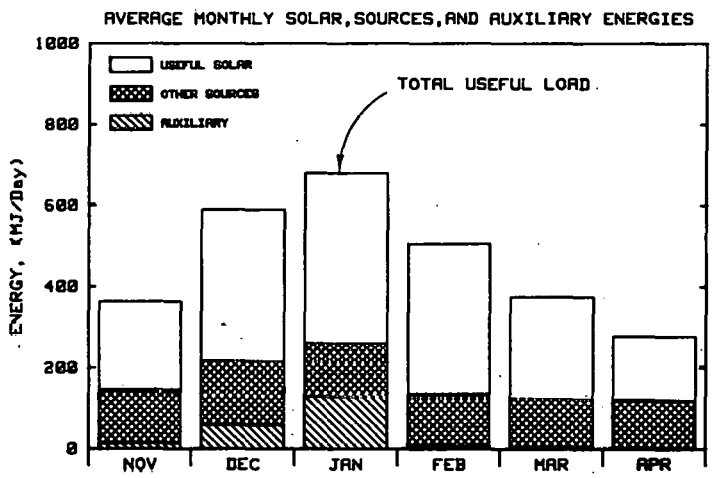


Fig. 2

TABLE I
ENERGY SUMMARY

Month	Incident Solar Radiation, (all windows)	Solar Collected ($\bar{Q}_A$) Transmitted Solar-Venting	Useful Solar ($\bar{Q}^*_u$) - (excess load)	Non-Auxiliary Sources	Auxiliary Heat	Electricity to Operate System	Non-renewable resources displaced, $\bar{Q}_H$	Useful Load, Living Area Only	Useful Load, Greenhouse Only	Rockbed COP	Overall COP, (SCOP')	Collection Efficiency,	System Efficiency,	Non-renewables Eff.	Solar Heating Fraction, Living Areas Only	Solar Heating Fraction, Whole House
	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)
	MJ/day, average											%	%	%	%	%
Nov	679	388	217	133	14	5	205	306	70	10.2	43.7	57.2	32.0	30.1	92.2	94.4
Dec	726	524	372	159	59	5	360	436	163	10.1	78.2	72.2	51.2	49.6	78.7	86.6
Jan	728	534	420	131	129	4	410	463	206	11.5	100.6	73.4	57.7	56.3	61.2	76.1
Feb	1014	573	370	128	8	7	352	388	123	11.9	51.2	56.5	36.5	34.7	96.8	97.8
Mar	890	449	246	125	4	5	235	312	67	10.6	52.6	50.4	27.7	26.3	97.8	98.4
Apr	814	328	160	116	1	5	147	233	41	10.9	31.3	40.3	19.7	18.1	99.0	99.3
Total	806	469	301	133	38	5	289	360	114	11.0	60.2	58.2	37.3	35.9	83.2	88.9



The values from columns 3, 4, and 5 are shown on the left.

The "total useful load" is the sum of columns 8 and 9.

"Useful Load" Calculation

Note on Fig. 1 that the losses from both the house and the greenhouse are divided up between "useful" and "excess" categories. These losses are due to conduction through the walls and windows and normal infiltration and do not include venting. The daily "excess load" is that part of the load which goes to maintaining the average daily living area temperature above 21.1°C (70°F) and the greenhouse temperature above 7.2°C (45°F). All the remaining loss is counted as "useful load".

In the living area 99% of the loss is "useful load" because the temperature seldom exceeds 21.1°C. The extreme range is 15.5°C to 23.9°C and the average is 20.0°C. The average clear-day swing is about 3°C. In the greenhouse only 41% of the loss is "useful load" because the greenhouse is always above 7.2°C and averages about 18°C.

olar Heating Fractions

The solar heating fraction is defined as:

$$SHF = 1 - \frac{\text{auxiliary}}{\text{useful load - other sources}}$$

Two solar heating fractions can be calculated, depending on whether the greenhouse is included or not. The yearly values are:

Living Areas Only:	83%
Entire House:	89%

Energy Savings

The non-renewable energy displaced for the 176-day period is:

50.9×10^9 Joules
$= 48.3 \times 10^6$ Btu
= 14149 kWhr
= $990 (@ 7¢/kWhr)
= 583 gallons of fuel oil (equiv)
= 16.6 barrels of oil not imported
= 8.9 tons of coal left in the ground

Based on the degree-day tabulations, the heating load for Sept., Oct., and May would increase this by another 15%. THe solar heating fraction for these months is usually 100%.

OTHER OBSERVATIONS AND ANALYSES

Electrical Energy to House

Electrical energy delivered to the house is shown in Fig. 3. The major reduction seen in March is due primarily to the start-up of the new, improved (active) solar water heater.

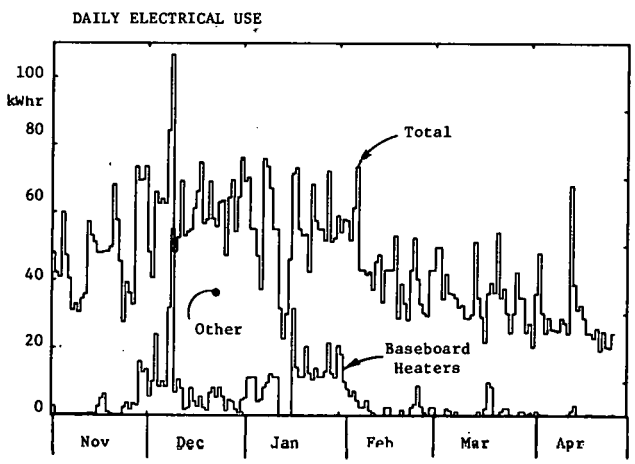


Fig. 3

Energy Sources and Losses

Figs. 4 and 5 show daily energy sources and losses.

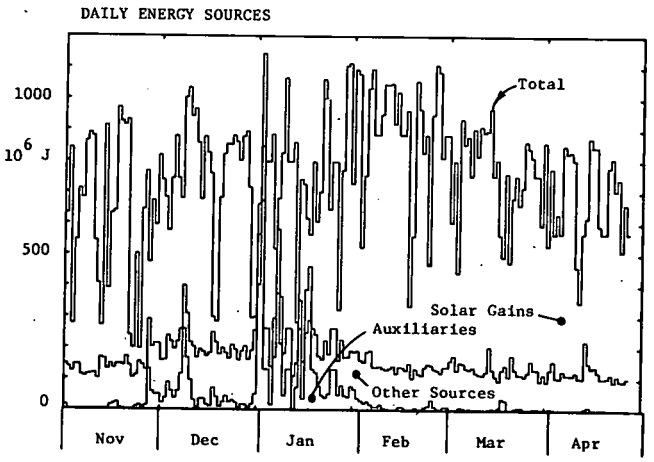


Fig. 4

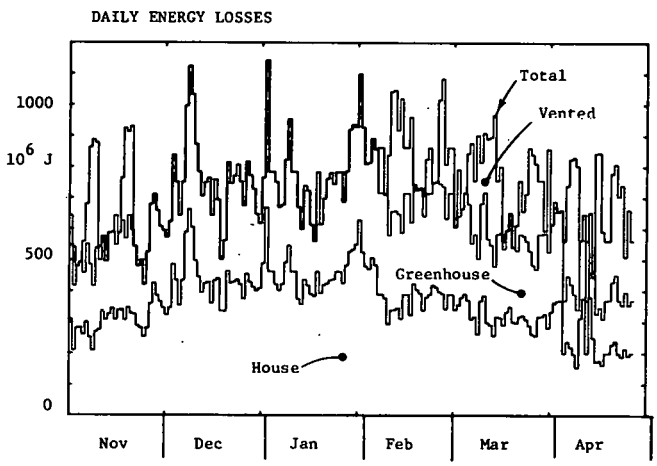


Fig. 5

Auxiliary Energy Use Pattern

The daily auxiliary use is shown in Fig. 3, totaling 790 kWhr for the season. It has been noted that the pattern of auxiliary energy use is predominantly after midnight and prior to 7 a.m. This is important because these are the off-peak hours of the electric utility and thus energy use during this period does not increase the amount of generating and distribution capacity that the utility must install and have on line.

This pattern can be observed in the hourly trace of auxiliary heating on Fig. 2 although it can also be noted there that during the most severe weather there is some heating on-peak. Note though that this does not lead to a correponding peak in the total electric power delivered to the house. Fig. 6 is a composite for the entire year showing the fraction of the total baseboard electric heat used in each one-hour time interval for the entire winter. The curve is normalized so that the integral is unity. Note that the peak integrated use occurs at 7 a.m. just before sunrise. Seventy per cent of the total energy is used during the 11 pm - 8 am off-peak period and only 10% is used on-peak. The house was operated

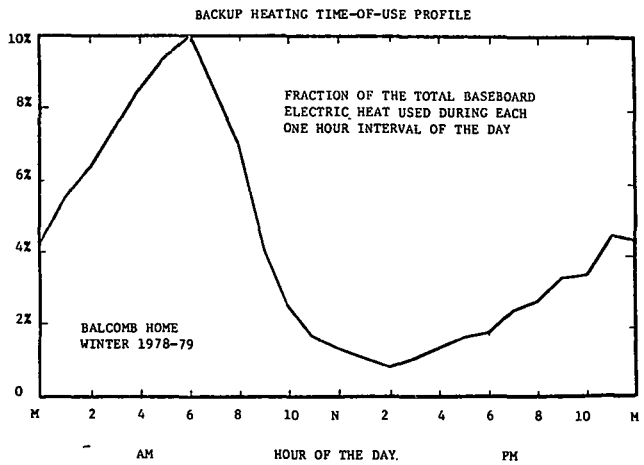


Fig. 6

with the thermostats left set at a constant 18.3°C in the kitchen and dining room area where the great bulk of auxiliary electric heat is used.

An analysis by William Wray has indicated that the use pattern observed in the Balcomb home is almost exactly what would be predicted for a passive solar home of the Trombe wall variety. It is probably characteristic of passive solar buildings in general. It is believed that a totally off-peak auxiliary heating system could be devised, for example, using an under slab heating system with appropriate controls.

Adobe Wall Behavior

The adobe wall which separates the greenhouse from the living areas has a major influence on the thermal dynamics of the house. The wall is 35 cm thick at the lower level and 25 cm thick at the upper level. The wall behaves somewhat like a Trombe wall. Some of the sunlight which is absorbed on the dark-stained south surface is lost directly back to the greenhouse and some is conducted into the wall. Most of the energy conducted into the wall is subsequently lost back to the greenhouse at night and some is transmitted through into the house. The data trace in Fig. 2 show temperatures.

Nine temperature measurements were made at three different wall locations in order to study the thermal behavior. The upper wall apparently receives more sun and experiences larger temperature swings than the lower wall. The phase difference between the west wall (which faces southeast and receives morning sun) and the east wall (which faces southwest and receives afternoon sun) is quite evident.

A technique has been developed for determining wall temperature distributions and surface heat fluxes based only on two measured temperatures by solving the diffusion equation. (If a temperature is measured at a third location this allows for an in-situ measurement of the wall thermal diffusivity.) This technique was used to infer wall heat fluxes for the three measured locations throughout the entire analysis period. A typical result for a sunny day is shown in Fig. 7 for the upper east wall.

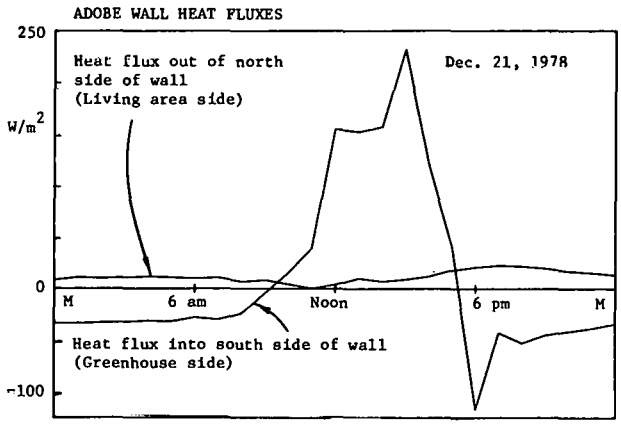


Fig. 7

The analysis indicates that the adobe mass wall is effective primarily as heat storage for the greenhouse but is less effective than expected in heating the living areas. The inner surface of the lower wall is relatively adiabatic but the inner surface of the upper wall provides significant heating to the upstairs bedrooms, especially in the early evening. A correlation fit indicates that above a threshold level of 280 kJ per day that 7% of the solar radiation transmitted through the greenhouse glazing is conducted into the living areas during the winter months (November-February) and 3% in the spring months (March-April).

Another correlation indicates that of the solar radiation which is transmitted through the greenhouse glazing 41% is absorbed by the adobe wall in the winter months and 21% in the spring months. This absorbed energy is particularly important to maintaining temperatures in the greenhouse at night which also helps the living areas indirectly by preventing heat losses to the south.

Fan-Forced Rock Bed

The effectiveness of the fan-forced rock bed as an important thermal element of the system has been emphatically confirmed. This system has the following important characteristics:

1. Heat is transported from the top of the greenhouse, where it would largely otherwise be lost, to the rock bed located under the living areas of the house. This moves heat from an area where it is not needed to an area where it is needed and it also translates it from a time when it is not needed, in the afternoon, to a time when it is, in the evening, by putting it into storage. The fan operated on 134 of the 176 day in the analysis period, transporting a total of

.9 x 10^9 J to the rock bed or an average of 3,800 KJ per day. The fan operates an average 3.3 hours per day drawing a total of 531 W for an overall electric consumption of 0.90 x 10^9 J. Thus the COP of the system is 11.0. Fig. 8 shows the daily energy transported to the rock bed.

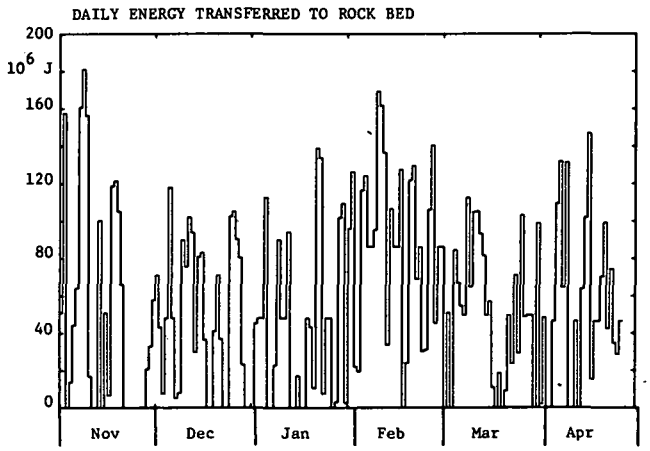


Fig. 8

2. Peak temperatures in the greenhouse are reduced by about 7°C. Normal maximum afternoon temperatures on sunny winter days are about 29°C, which is a reasonably pleasant and comfortable temperature if one is just passing through. On December 22 and 23 the fan controller was turned to the off position for the day resulting in maximum temperatures of 36°C in the afternoon, by far the highest temperatures ever recorded in the house. This is definitely uncomfortable. Despite this higher temperature the heating of the greenhouse adobe wall was not significantly greater indicating that most of the excess heat was lost.

3. The mean radiant temperature of the living areas is increased by the warm floor. This helps to rebalance the room temperature and increases thermal comfort with a lower air temperature. As can be seen in Fig. 2, floor temperatures are typically maintained at about 22°C during sunny weather and drop slowly to 18°C after a long cold spell, whereas they would otherwise be about 14°C without the underfloor heating. The fact that the floor temperatures on the north side are slightly higher than on the south, due to the inlet location, is seen as an advantage since the other principal radiant source in the room is the adobe wall on the south.

The Greenhouse as a Solar Collector

The greenhouse is an effective solar collector. Of the 115 GJ incident in the greenhouse glazing, 71% is transmitted into the greenhouse. Of this, 40% is transferred into the house. The correlation of Fig. 9 shows that there is a threshold level of about 150 MJ/day of solar gain into the greenhouse beyond which 51% is subsequently transferred to the house.

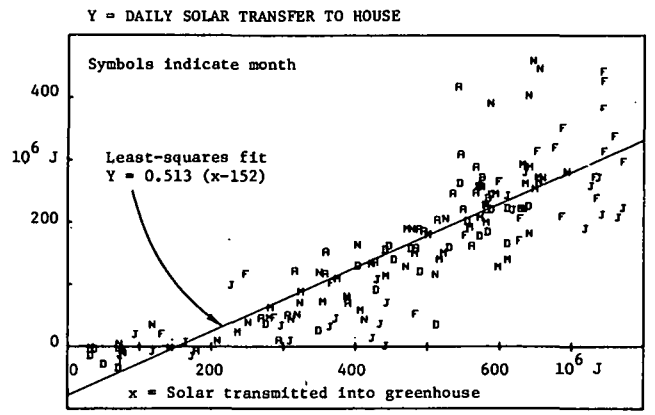


Fig. 9

FUTURE EVALUATIONS

Need to Measure Convection

During the next heating season it is planned to obtain information which will allow a direct measurement rather than relying on the indirect procedures utilized in the present analysis. Since these terms tend to be much more significant than had been originally thought, it is felt necessary to confirm that their magnitudes are approximately correct.

The term of greatest interest is the convection from the greenhouse to the house. The convection was calculated by assuming that the air exchange rate between the greenhouse and house was 69 m^3/min when the greenhouse was warmer than the house and 2.4 m^3/min when the house is warmer than the greenhouse. The former number is consistent with the magnitude of convection velocities which were crudely measured using observations of smoke, assuming that the doors between the house and the greenhouse are open. The latter number corresponds to an air exchange rate between the house and the greenhouse of 1.5 greenhouse air volume per hour assuming that the doors are closed.

Thermal Storage Effectiveness

It is also planned to measure temperature swings in many more locations where heat is stored indirectly.

ACKNOWLEDGMENTS

The authors wish to acknowledge the extensive efforts of others which made possible these performance evaluations, especially B. T. Rogers who performed the steady-state heat load calculations, Michael Ulibarri who picked up and processed the weekly data tapes, Michael Calloway who installed much of the instrumentation, and Peter Stromberg who prepared many of the hourly plots, including Fig. 2. We also wish to acknowledge that the cross-section on the first page was obtained from Sandia Laboratories Report SAND 79-0824 "Passive Solar Buildings".

NOTES

THE SOLAR CLASSROOM AT HAMILTON COLLEGE

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ABSTRACT

The performance of a simple, passive, solar building in the cold, cloudy climate of upstate New York has been tested over two heating seasons: 1977-78 and 1978-79. The thermal conductance of the building has been found to be 112.4×10^5 J/DD($^\circ$C) (5900BTU/DD($^\circ$F)) and the sun's contribution to heating the building has been found to be 69% the first year and 66% the second. The thermal storage is sufficient to provide for several days carry-over and also serves to keep the temperature swings within comfortable limits. Simple steps which might improve this performance to a 85% contribution by the sun are suggested.

INTRODUCTION

A passive, solar building has recently been designed and constructed at Hamilton College for the dual purpose of providing classroom space for astronomy students at the observatory, which is somewhat removed from the other buildings of the campus, and to serve as an experimental building with which to explore the feasibility of solar heating here in upstate New York.

The test of feasibility is a severe one since the heating season is long and hard and the amount of insolation is low. The main purpose of this paper is to explain how feasibility has been tested. The paper is divided into the following parts: A brief description of the building and the relatively simple measurements which were recorded to evaluate its performance; an account of how these data were analyzed in several different ways to yield a common value of the thermal conductance of the building; an account of how the sun's contribution to the heating of the building was found both on a month by month basis and for the heating season as a whole; an investigation of the thermal storage properties of the building; finally, a discussion of ways in which the performance of the building might be brought to its best level.

DESCRIPTION OF THE BUILDING

The dimensions of the classroom are approximately 7.6m x 7.6m x 2.6m (40' x 25' x 8.5'). The long axis lies on an east-west line and the south side has ten windows each of which has the dimensions 1.84m x .80m (6.1' x 2.6' or 73.6" x 31.5"). The east wall has one window of 1.89m x 76m (6.2' x 2.5') and the west wall a window of 1.89m x .55m (6.2' x 1.8') and an entrance door. The east and west windows can be opened but the south windows are sealed. All windows are double glazed. See Fig. 1 for the south elevation.

The building is of concrete block construction with poured footings below the frost line. The floor is a poured .15m (6") concrete slab. The roof is formed with light wooden trusses covered with corrugated aluminum and the rafters are wood of dimensions 5cm x 30cm (2" x 12"). See Fig. 2 for a N-S section.

The concrete block is insulated on the outside with 10cm (4") of expanded polystyrene insulation. Under the floor there is another layer of 5cm (2") of polystyrene below the frost line and above the 1.27cm (1/2") sheetrock ceiling there is 30cm (12") of fiberglass batt insulation. Thus the room is insulated on all six surfaces, including underneath, but with a large thermal mass of concrete, and dirt under the floor, included inside the insulation. The polystyrene is protected on the outside by a synthetic stucco which bonds easily and well to it.

The back up heating is provided by electrical resistance heaters which are rated at 5.25kW. The electrical energy consumed in the building is monitored by a standard kilowatt-hour meter. Some small part of the total electrical energy used in a heating season is because of lights, a fan, a hot water heater and a pump. This part amounts to about 2kWh each day on the average or about 300kWh out of more than 4000kWh in a heating season. This background use is established by finding the rate of energy use during those periods when no heating is required. The temperature inside and outside is measured by means of copper-constantan thermocouples. The intensity of the radiation from the sun is measured by means of a silicon photovoltaic cell mounted at the center of the inside bottom edge of the eastern-most south-facing window. A reference temperature (and a signal zero) is

also measured. For the first year this was the freezing point of water but since January 1979 this has been a temperature controlled reference junction which has been set at 25°C. The three temperatures and the relative intensity of the sun were recorded by means of a switching circuit and a strip chart recorder. The switching circuit caused the recorder pen to respond to one input after another so that each input was sampled about every five minutes. Thus aside from changes which took place in times less than five minutes there is available an almost complete continuous record of these four measurements from February 1, 1978 until the present. The record of the inside temperature includes information about whether the electrical heaters were on or not. When the thermostat was cycling on (at 15°C) and off (at 17°C) this inside thermocouple showed a characteristic wave pattern with a difference of about 2°C between the crests and the troughs. The wave period and shape varied with such conditions as the outside temperature and the amount of insolation for the few days just past. See Fig. 3 for an example of this pattern.

FINDING THE THERMAL CONDUCTANCE (K)

In designing the building we calculated the steady state heat losses by using the known R values of the materials of which the building was to be constructed and R values for sealed double glazed windows, operating double glazed windows, and a door. These losses are given below in a table. Note that the infiltration losses can't be directly calculated. We guessed a small rate of one half exchange per hour because the building was to be made as tight as possible. Even with this small rate of exchange infiltration represented the second highest heat loss.

Windows	53.5×10^5 J/DD(°C)	(2820 BTU/DD(°F))
Infiltration	29.6×10^5	(1560)
Walls	16.4×10^5	(864)
Ceiling	9.76×10^5	(514)
Floor	6.38×10^5	(336)
Door	4.56×10^5	(240)
	$K = 120.0 \times 10^5$	(6300)

However, since the infiltration is only a guess and, for example, could easily be twice as large this value of K is only good to maybe 25%.

It is possible to find K from the data gathered on the performance of the building and two different methods were used: one based on a short term record in which the thermostat duty cycle at night was used to find the energy in and the other based on a long term record over two three month periods during Feb.-April 1978 and 1979 in which the sun's contribution was calculated by using the solar cell records and this was used along with the kWh record to find the total energy put into the house.

The thermostat duty cycle method depends on finding the number of hours during which the heaters were on from the record and knowing that the heaters are rated 5.25kW. A comparison for several different nights with the kWh meter day by day record showed that this technique yielded accurate results. However, the purpose of using this method during this period was to exploit the fact that from Jan. 23 until Jan. 31 there was very little sun and the outside temperature stayed remarkably stable at about the freezing point day and night for the whole period. Thus the energy in could be calculated towards the end of this period with the reasonable expectation that the thermal storage had been drained down to the thermostat setting of 15°C and that the only heat necessary to be considered was that from the heaters. The value of K found averaged over the last few days of this period is 112.4×10^5 J/DD(°C)(5920 BTU/DD(°F)).

The long term method required detailed calculations for the sun's trajectory across the heavens at different times of year and at different times of day (and for the effective area and the shading factor corrections which follow from the sun's trajectory). With these calculations we could use the solar cell records to find the amount of heat put into the building by the sun over the three month period in two different heating seasons. From the records we also found the number of DD's, which numbers taken month by month agreed within a few percent with those recorded locally by the Niagara Mohawk Power Company. The long term, we expected, would serve to average out the thermal storage effects and this expectation was confirmed as we found K = 111.3×10^5 J/DD(°C) (5860 BTU/DD(°F)) in 1978 and K = 119.8×10^5 (6310) in 1979.

THE SUN'S CONTRIBUTION

The method described above for finding K also gives us directly the sun's contribution to heating the building and these results are given below.

	1978	1979
February	56%	53%
March	69	63
April	64	70

Although we have not analyzed the solar cell data for the whole of the 1978-79 heating season we believe that the average value of K found above (112.4×10^5 J/DD(°C)) (5900 BTU/DD(°F)) is a reliable one and hence we can use it along with the kWh readings and the number of degree days to find the month by month contribution of the sun for the whole season. These results are given below

	No. of DD's	kWh	%contribution of sun
October 78	326(587°F)	0	100
November	466(839)	305	79
December	656(1181)	1017	50

nuary 79	766(1379)	1271	47
..bruary	828(1491)	1044	59
March	521(937)	581	64
April	387(696)	353	71
May	150(270)	0	100

N.B. The results for February are 6% high when calculated this way as compared with the direct calculation above. However, the results for March and April are only 1% high and taken altogether the three results agree well enough overall to allow us to claim that the sun's contribution is known to within a few percent for any given month. The number of degree days in a month was the number supplied by Niagara Mohawk which is based on an inside temperature of 65°F. A correction for the fact that our average inside temperature was lower than this brings the contribution of the sun down a percent or two. Also note that the number of degree days shown here is about 95% of the total for the whole year.

It is also possible by the same method to calculate the sun's contribution for the season as a whole and in 1977-78 with 4344 DD's (7820 °F) and 4153 kWh of electrical energy consumed the percentage is 69 and in 1978-79 the numbers are 4331 DD's (7795 °F), 4571 kWh with 66% contributed by the sun.

One other interesting observation based on the solar cell readings is that even on the worst days something like 300 x 10^5J(30,000 BTU) in diffuse radiation enters the building each day. In February over the past two years the average energy required per day is 2,860 x 10^5J(271,000 BTU) so about 11% is contributed by the sun even on the worst days. This diffuse radiation by the way, was taken into account in the calculations to find the sun's contribution which have already been mentioned above.

THERMAL STORAGE

In the data which has been carefully analyzed for Feb.-April 1978 and Feb.-April 1979 we have found that although overall there is an energy balance within our experimental accuracy of a few percent over the three month period or over the period of a month on a daily basis this balance does not hold. For example, on Feb. 6, 1979 there was more energy in than out in the amount of 2,408 x 10^5J(228,000 BTU) and on March 3, 1978 there was more energy out than in with the imbalance being 1,942 x 10^5J(184,000 BTU). These were the largest daily imbalances found but in general the daily energy accounting revealed some imbalance which we attribute to energy going into storage if the imbalance is of the former kind and energy coming out of storage if it is the latter.

If the credits and debits are accumulated over consecutive days they amount to + 4,490 x 10^5J (426,000 BTU) for a sunny period from March 4 to March 10, 1978 and - 2,620 x 10^5J (248,000 BTU) for a cold, cloudy period from Feb. 4 to Feb. 7 1978. These accumulations were the extremes found. A more typical period was that of Feb. 25 to March 3, 1978 just before the extended sunny period mentioned above. The mean daily use of electrical energy for this period of mixed cloud cover and sunshine was 1,030 x 10^5J(97,500 BTU). Thus a thermal storage capability of several days is indicated. A further indication of the same thermal storage capability is the fact that for the five days starting on March 10, 1978 only 503 x 10^5J (47,700 BTU) of electric heat were required, or only an average of 100 x 10^5J (9,500 BTU) each day. See Fig. 4 for a record of the sunny period March 4 to March 10.

An important consideration, which in our case is directly related to thermal storage, is that of comfort. What are the temperature swings which occur? On sunny days the maximum temperature reached was generally 24°C (75°F) or less and the minimum was always greater than or equal to the thermostat setting of 14.7°C (58°F). Even during the extended sunny period during March 1978 the highest temperature reached was 25°C (77°F). The thermal storage capacity of the walls and floor is thus acting to moderate the amplitude of the temperature swings with the result that a swing of about 10°C (18°F) is the largest found and the daily swing is on the average about half this big.

OPTIMIZING THE PASSIVE PERFORMANCE

Two steps suggest themselves as ways in which the Solar Classroom might be improved. First, addition of triple glazing to the windows would cut down on the largest heat loss contribution as seen above. Second, the addition of an air lock at the entrance would cut down on the second largest heat loss contribution.

The first step might seem to be a somewhat timid one but, in fact, more layers of glazing than three (or four) would not be effective as about 8% of the sun's energy is lost for each additional layer. The second step's effect is hard to estimate but could be measured and we intend to carry out these two steps separately as individual experiments. The triple glazing is already in place on the south facing windows and we expect that this change will bring the sun's contribution to heating up to about 75% for the season.

Estimates can be made of the effects of taking the two steps suggested above and also adding insulating shutters over the windows at night. In this case we think it possible that in the worst month (January) we might find the sun's contribution to be ∿66% and for the season the sun's contribution to be ∿85%. We doubt that it is possible to do much better than this with a passive design in our climate.

ACKNOWLEDGEMENTS

The building design was worked out with the advice of Mr. Elihu Root III who also put together the measuring devices, the switching circuit, the temperature reference junction and the strip chart recorder so that continuous data on the building's performance could be recorded. Mr. Frank Marsicane of the Physical Plant Department made the working drawings for the building and drew up its specifications.

FIG. 1
SOUTH ELEVATION

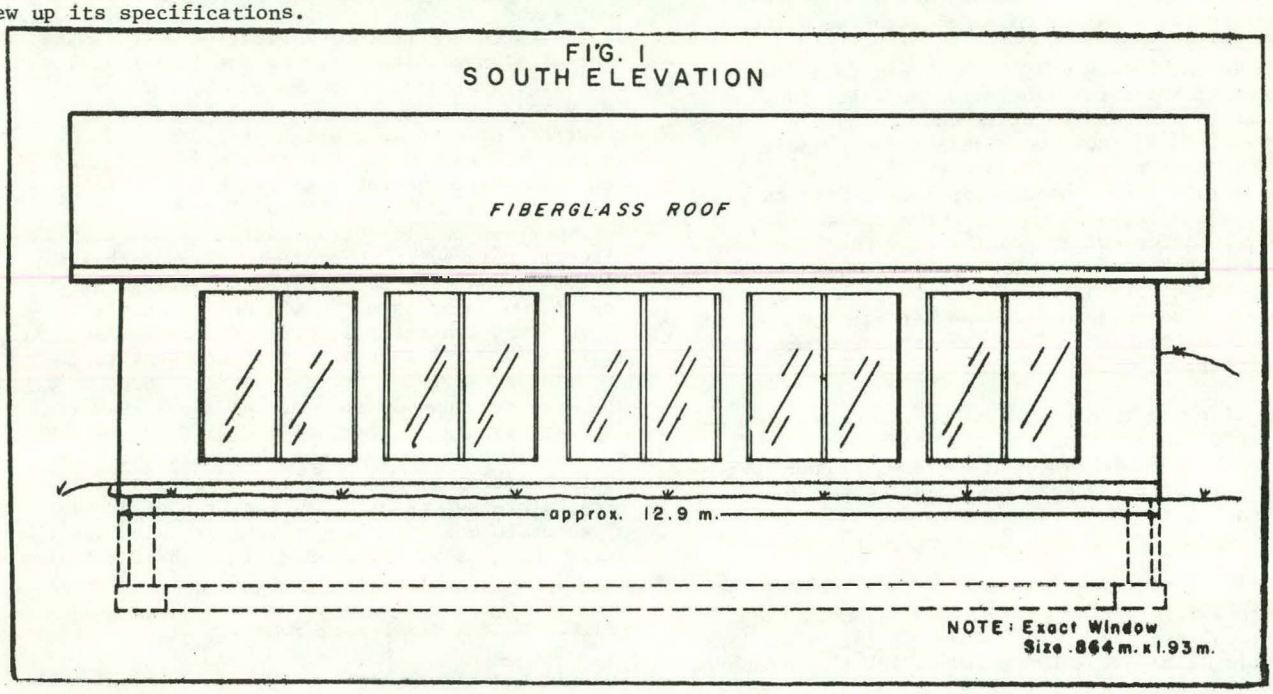


FIG. 2
TRANSVERSE SECTION

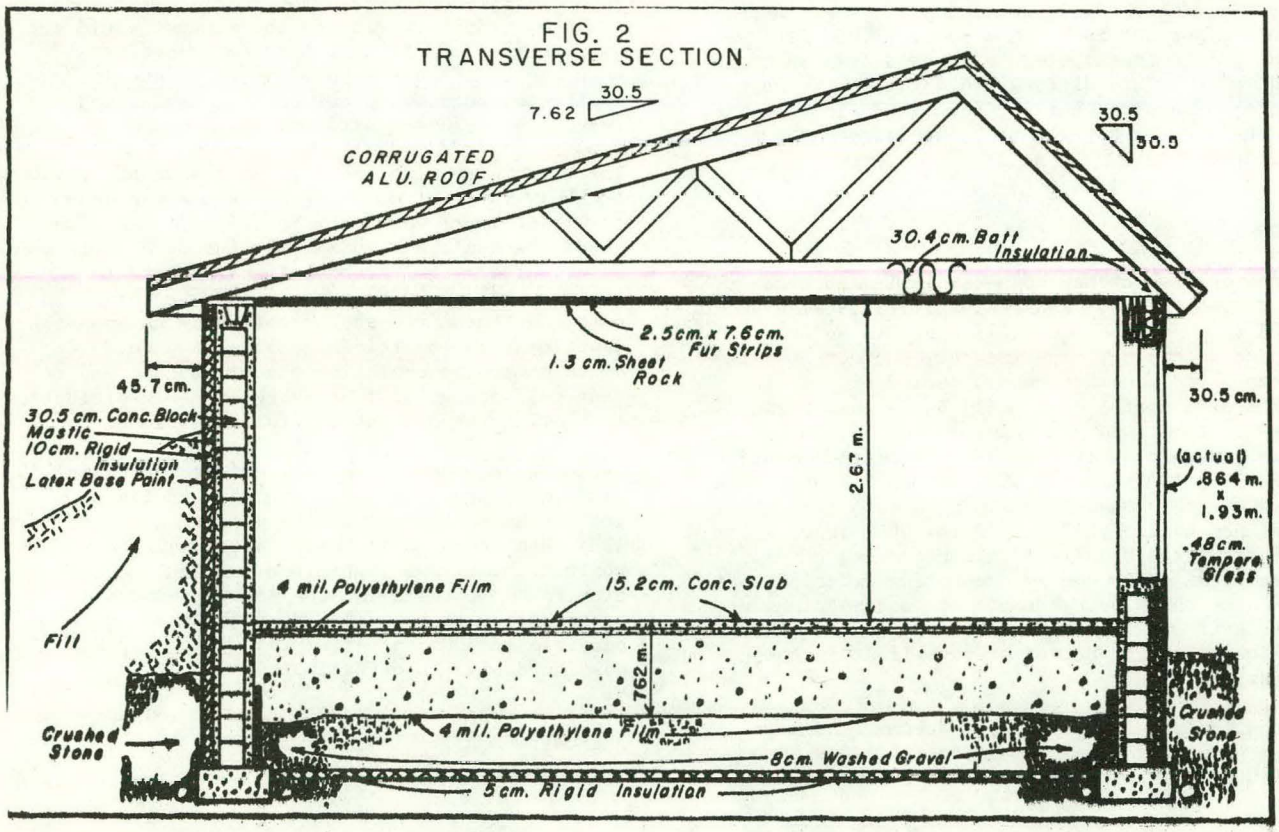


FIG. 3

FEB. 6, 1978
CLOUDY & COLD

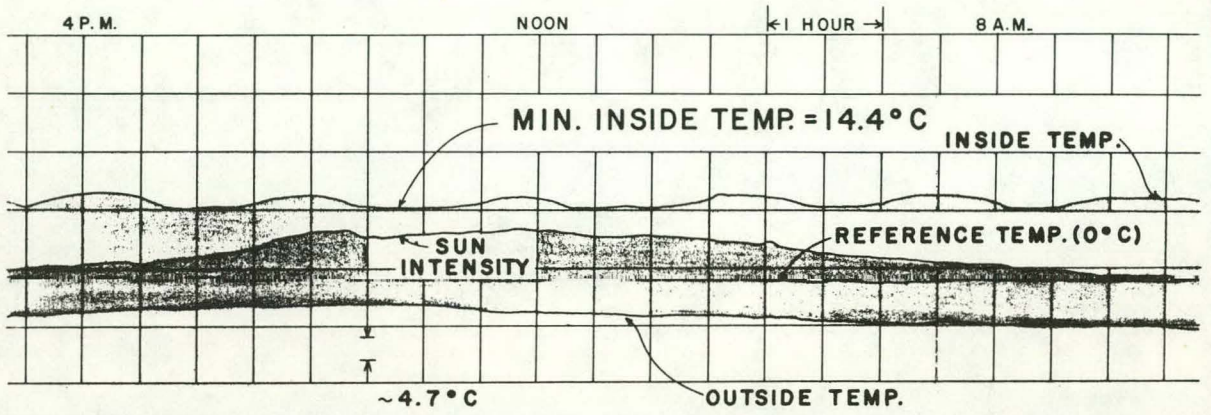
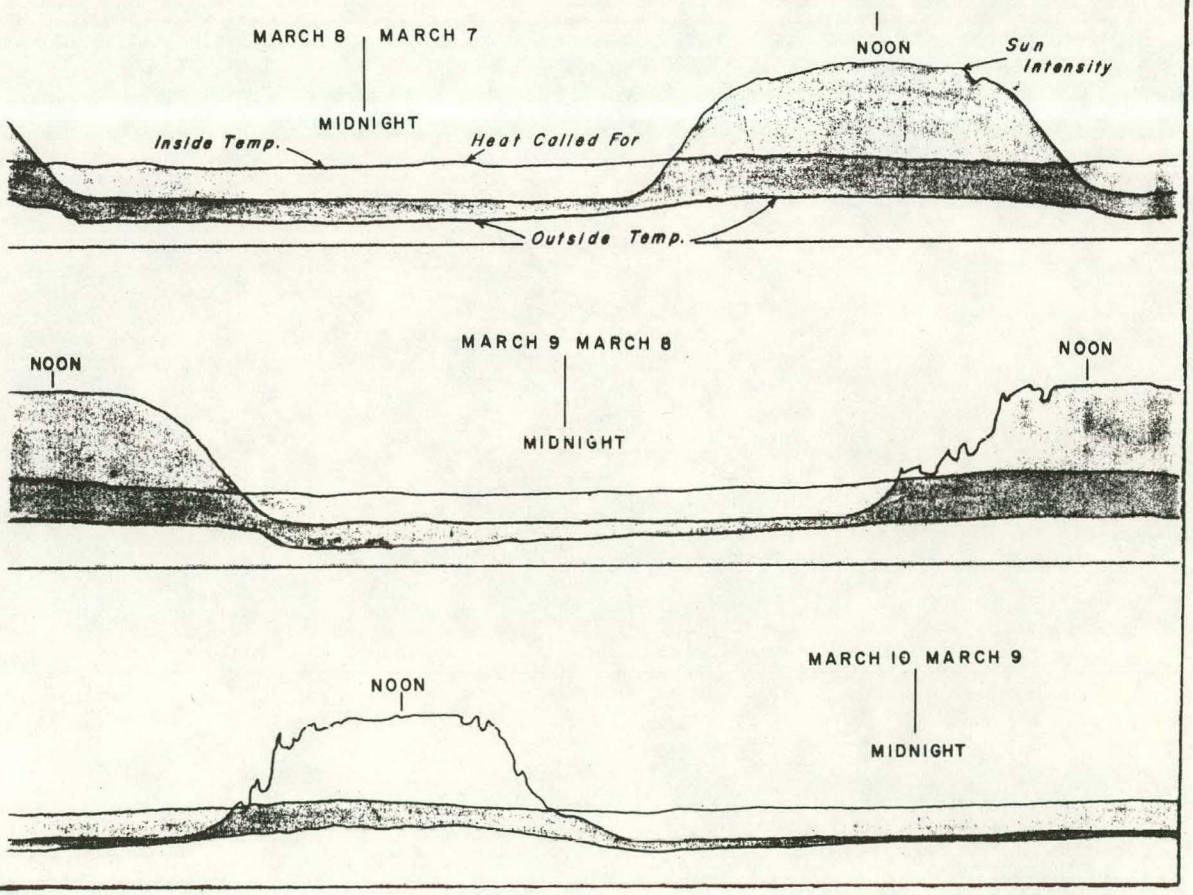


FIG. 4
(1978)



NOTES

RESULTS FROM A PASSIVE THERMAL STORAGE ROOF ON A MOBILE/MODULAR HOME IN LOS ALAMOS*

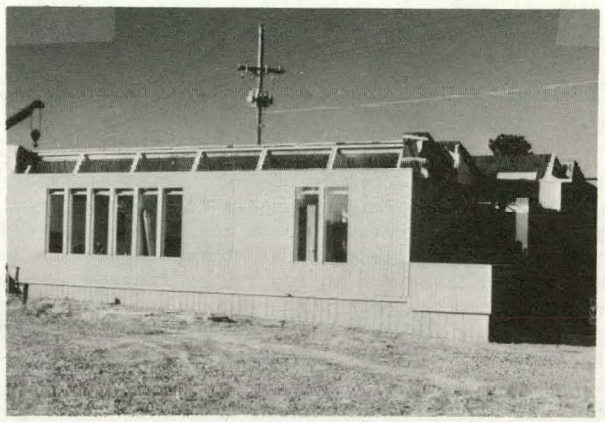
J. D. Balcomb, J. C. Hedstrom, S. W. Moore and R. D. McFarland
Los Alamos Scientific Laboratory
Los Alamos, New Mexico 87545

ABSTRACT

Heating season performance data are developed for the LASL Mobile/Modular II home for the period October 1978 through May 1979. The building was constructed in two sections with a combined floor area of 101 m^2 and transported to Los Alamos, New Mexico. The design incorporates a series of four clerestory windows across the roof, hinged insulation panels located within the clerestory space, and water in plastic bags as the heat storage. The heat storage roof seems to be effective thermally but was overpowered by direct gain through south windows during the early part of the heating season, tending to overheat the space. During the latter part of the season these windows were blocked with insulation greatly reducing temperature swings without apparent reduction in performance. The useful solar heat averaged over the season is 166 MJ/day corresponding to 80% of the net load and 23.5% of the solar radiation incident on the glazing.

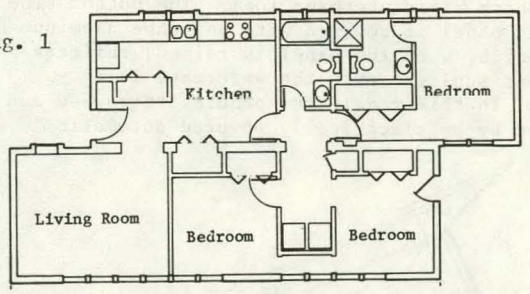
BUILDING DESCRIPTION

This home is the result of a project funded by the United States Energy Research and Development Agency through LASL. The object of the project was to (1) investigate passive solar heating systems for their applicability to the mobile/modular home industry, (2) select a single system, (3) design and build a house, and (4) measure its performance. Of the many concepts studied and analyzed, the roof-mounted system offered the most freedom from possible site restrictions: the roof apertures are relatively clear of shading from neighboring homes, trees, etc.; the design can be altered to face the apertures in any direction with respect to the house orientation; the house does not require any site preparation other than that normally required for a modular home; and the water for thermal storage can be loaded into the plastic bags after the house is established on its permanent site, reducing the over-the-highway shipping weight.



In order to meet the width limitations of highway transportation, the house is constructed in two units, each measuring 4 m by 13.1 m. The two units are bolted together on site to form a house. One unit contains a living room, two bedrooms, and a small utility center for a washer-dryer combination; the second unit contains the kitchen, master bedroom and two baths.

Fig. 1



The walls are standard frame construction using 2 x 6's for studs. The inside is finished with drywall material, and the outside, with wood paneling. The voids between the studs are filled with fiberglass insulation.

The floor joists are 2 x 8's which are sheathed, top and bottom, with plywood. Again, the voids are filled with fiberglass. When emplaced

on poured concrete piers at its permanent site, a plywood skirting is attached around the perimeter to seal off the space between the floor and the ground. This skirting is insulated on the inner side to further reduce heat losses through the floor to the outside.

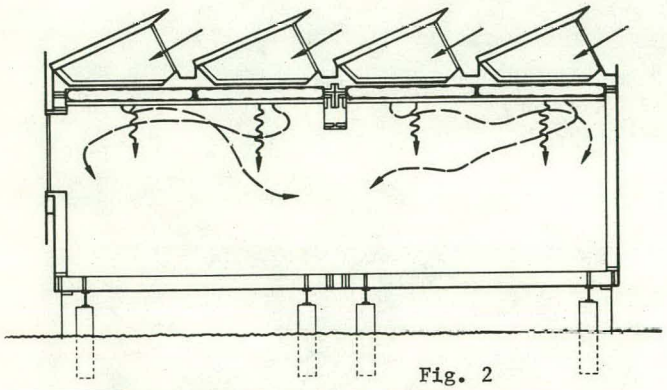


Fig. 2

The roof is supported by open-web metal trusses. Corrugated steel ceiling panels rest on the lower flanges of the trusses and on top of these panels lie the water-filled thermal storage bags. An angle-iron framework on top of the trusses supports the sawtooth roof structure. The south-facing surfaces are double glazed; standard patio doors were used in this model. The north-facing surfaces are sheathed with plywood and covered with Coilzak^{R} aluminum sheeting, which acts as a reflector. A 9 cm layer of urethane foam insulates the underside of the plywood sheathing. Within each of the sawtooth apertures, an insulating panel rests on the top flange of the roof trusses, insulating the water storage bags that lie beneath. These panels are hinged along their north edge and can be lifted by a cable-and-pulley mechanism to expose the waterbags to sunlight admitted through the glazed, south-facing surfaces of the sawtooth roof. These panels are made of plywood bonded to 5 cm-thick rigid urethane foam. The bottom side of each panel is covered with polished aluminum foil which, when the panel is raised, reflects incident sunlight onto the waterbags lying below. In this model, the panels are raised and lowered by an electrically powered automatic system keyed to photo thermal sensors in the apertures.

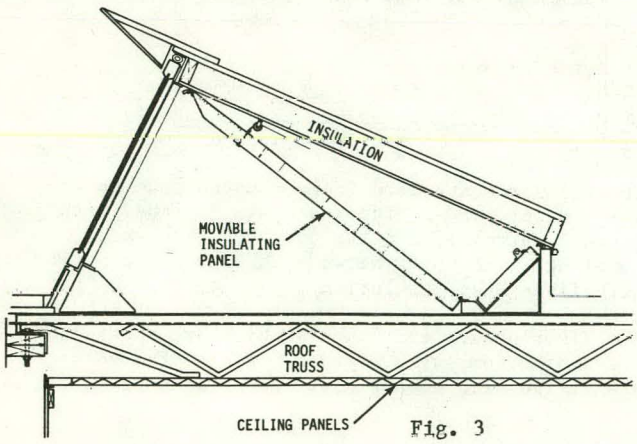


Fig. 3

Operation

This solar system is totally passive in operation except for the action of the movable insulation over the waterbags. When the sun is shining, the insulating panels are raised, allowing the solar energy to fall on, and be absorbed by, the water-filled bags. The reflecting surfaces on the north slopes of the sawtooth roof and on the underside of the insulating panels direct additional sunlight onto the waterbags, increasing the overall gain of the apertures. At night, or on cloudy days, the insulating panel is closed, and thermal radiation from the waterbag is returned to it by reflection. A flexible rubber seal around the perimeter of the insulating panel inhibits warm air around the waterbags from seeping into the aperture space, where the heat would be lost to the outside through the glazing.

The waterbags rest on the steel ceiling panels. Heat is conducted from the waterbag through the panel and transferred by radiation and convection to the living space. Convective heat transfer from the warm ceiling to the air below is rather inefficient, hence the radiative mode accounts for most of the heat transfer.

The geometry of the sawtooth roof was optimized by computer simulation methods. Optimum dimensions were selected based upon minimum annual auxiliary energy requirements as determined from the computer simulation analysis. There are two critical design parameters: the angle of tilt of the glazing and the length of the roof overhang. The transmittance of the glazing the fraction of sunlight which actually passes through both panes of glass varies with the angle of incidence of the sunlight on the glass surface. The more obliquely the sunlight strikes the glass surface, the less of it is transmitted through the glass and the more of it is reflected away. The overhang shades the glass beneath it--the longer the overhang, the more glass is placed in the shade. In summer, the overhang provides more shade than in winter. The effects of this combination of reflecting and shading characteristics was explored by the computer simulation. and the optimum geometry for full-year performance was chosen. The glass tilt is 65 degrees from the horizontal. The overhang extends about 30 cm beyond the front plane of the glass. These parameters would be different for different latitudes and for localities with different annual insolation patterns. The width of the glazing in the apertures is about 66 cm, and each of the four apertures extends the full length of the house. The waterbags are filled to a depth of 15 cm. At this capacity, the thermal storage mass is 12,800 kg of water (about 3370 gallons). This works out to a thermal storage capacity of about 530 KJ/^{o}C for every square meter of floorspace in the house. The design of the house includes additional structural support for this added mass.

The 10 south-facing windows admit direct-gain solar energy to the house interior to supplement the roof-pond aperture system.

TYPICAL DAILY PERFORMANCE

This house was completed and transported to its site at Los Alamos in September 1977. Installation of instrumentation for monitoring the performance of the house was completed in January 1978.

A series of typical data records covering the period March 23-25, 1978, are shown in Figs. 4 through 6. The first two days are cloudy, while the third day is clear and sunny, as is shown by the insolation curves. Outdoor ambient temperatures are about the same for each of the three days and are relatively moderate, so these records do not illustrate a severe test of the solar heating system.

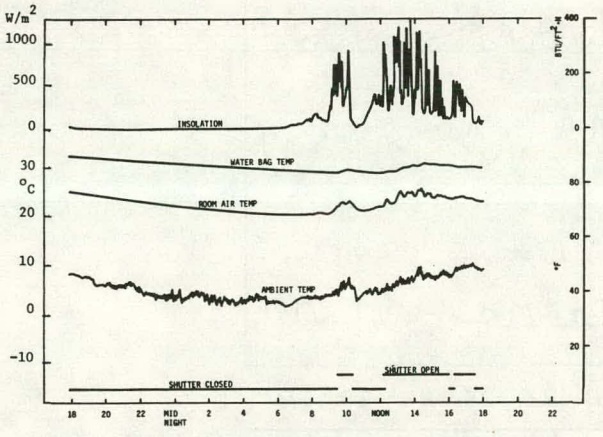


Fig. 4 March 23, 1978

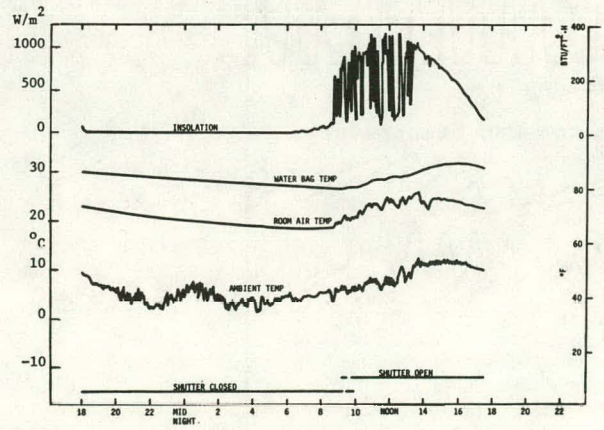


Fig. 5 March 24, 1978

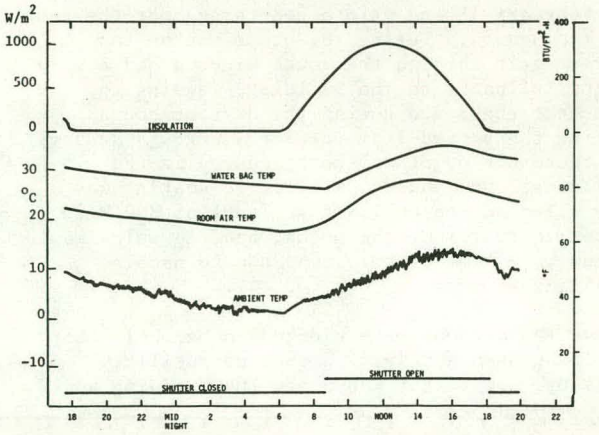


Fig. 6 March 25, 1978

The responsiveness of room temperature to the sun is clearly illustrated on March 23. An early morning burst of sunshine is followed by more than an hour of cloudiness with the sun obscured. The room temperature climbs rapidly in response to the direct gain coming through the large south windows. During the cloudy period, the room temperature initially drops rapidly and then levels out as thermal equilibrium is established. In general, whenever the insolation curve shows sharp variations due to cloudiness, the room temperature curve is correspondingly ragged.

On the sunny day, March 25, the movable insulation panels in the apertures open at about 8:40 in the morning, when the insolation level was already up to about 570 W/m^2. The waterbag temperature begins to climb immediately, as shown by the sharp change in the slope of that curve. The waterbag temperature peaks just before 4 p.m., when the insolation curve is on a decreasing slope, and has reached about 615 W/m^2. The insulating panel does not close as the sun's insolation drops below the 570 W/m^2 mark; instead, it remains open until the sun has apparently set at 6 p.m. During these last two hours that the panel is open, the thermal storage mass is cooling, not warming. The rate at which the waterbags are cooling is promptly slowed when the insulating panel finally closes.

The high interior temperatures shown should not be taken as typical. During the period that the data were taken, the house was unoccupied and kept tightly closed so that heat flow parameters could be accurately determined. House temperatures can easily be reduced to desired levels by venting excess heat to the outside.

It should be noted that the building was not in the same identical condition during the entire analysis period. During the interval from October 26 to November 28, 1978 the direct gain windows on the south side were blocked with foam insulation panels with reflective foil facing outward. This same condition was re-established

on February 15 and maintained throughout the rest of the season. During the other months the direct gain through the south windows had a very major influence on the building, tending to overheat the space during the daytime hours. During the period from October 1 through January 23 there was no other source energy in the building. On January 24 electric heating was installed in the building at level of 800 W in order to represent the normal heating which would occur in a home of this type due to people, appliances, etc.

After the windows were blocked on March 1, the building used a trivial amount of auxiliary heating. It is not known how the building would have operated in this configuration through the mid-winter months but it is noted that the performance in the spring is excellent.

Hourly plots for the months of January and March, 1979 are presented in Figs. 7 and 8 to show the results with and without direct gain. In January, the room temperature is seen to rise above the roof storage temperature and has a daily temperature swing of up to 25°F. In March, with the windows closed, the room temperature tracks the storage temperature and has a daily temperature swing of about 10°F which is much more acceptable. The room temperature did reach a maximum of 90°F which could easily have been lowered by ventilation.

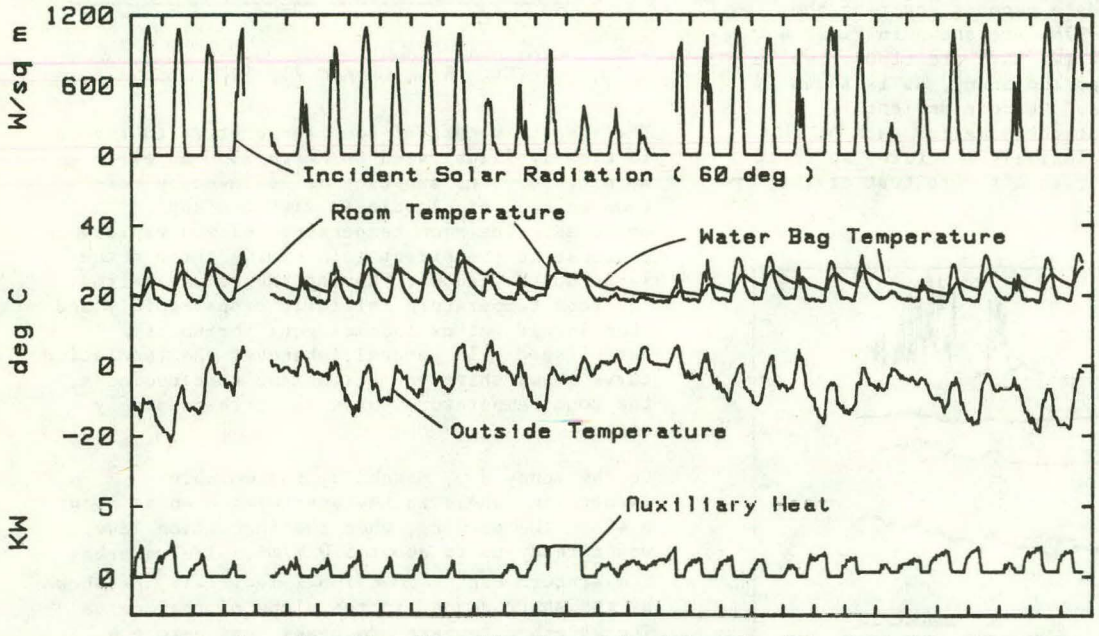


Fig. 7
Jan. 1979
With direct gain

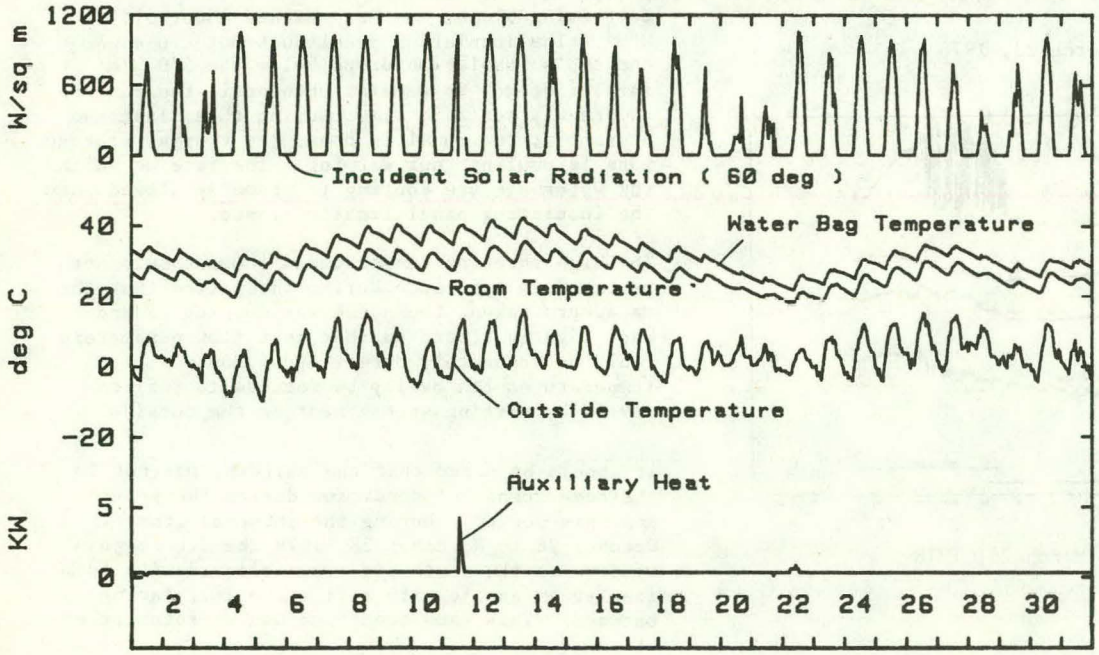


Fig. 8
March 1979
Without direct gain

ENERGY PERFORMANCE

Performance evaluation results are shown in Table I and Figs. 9-11.

The heat loss coefficient for the building was determined experimentally to be 159 W/°C. This coefficient was then used in order to perform the data analysis shown in Table I. The column marked "direct gain" is intended to represent energy which is transmitted in through the south windows. It was not measured directly but was inferred indirectly based on the thermal unbalance of the building. This should be relatively accurate because there is little mass in the building below the ceiling level. The heat fluxes downward from the ceiling were calculated using the experimentally measured coefficient of 527 W/°C based on the difference between the measured ceiling temperature and the average measured room temperatures. That this procedure is relatively accurate can be observed by noting that the calculated direct gain values for the months of November, March, April, and May are relatively small when it is known that the actual direct gain was zero.

TABLE I. MOBILE HOME NO. 2
MONTHLY PERFORMANCE SUMMARY

Month	Incident Solar Radiation on Roof Apertures	Incident Solar Radiation on South Windows	Energy Transfer from Storage to Room	Direct Gain Through South Windows	House Electric	Auxiliary Heat	Useful Load TRM < 21°C	Useful Solar, $\bar{I}^{*}_{u}$ or Q_H	Solar Heating Fraction	Roof Aperture Efficiency	System Efficiency
	MJ/Day, Average								%	%	%
Oct	668	150	65	122	0	11	123	112	90.8	9.8	13.7
Nov	440	10	174	16	0	55	223	168	75.2	39.5	37.3
Dec	589	236	131	102	0	122	328	206	62.9	22.2	25.0
Jan	532	239	91	135	13	107	333	213	66.5	17.1	27.6
Feb	723	157	225	80	67	11	294	216	95.1	31.1	24.6
Mar	649	0	256	0	63	1	239	176	99.7	39.4	27.1
Apr	615	0	166	14	65	2	169	102	98.5	27.0	16.6
May	555	0	120	23	57	0	153	96	100.0	21.6	17.3
	598	107	155	65	31	42	238	166	79.8	25.9	23.5

Roof aperture area = 31.3 m^2 (338 ft^2) South window area = 13.1 m^2 (141 ft^2)

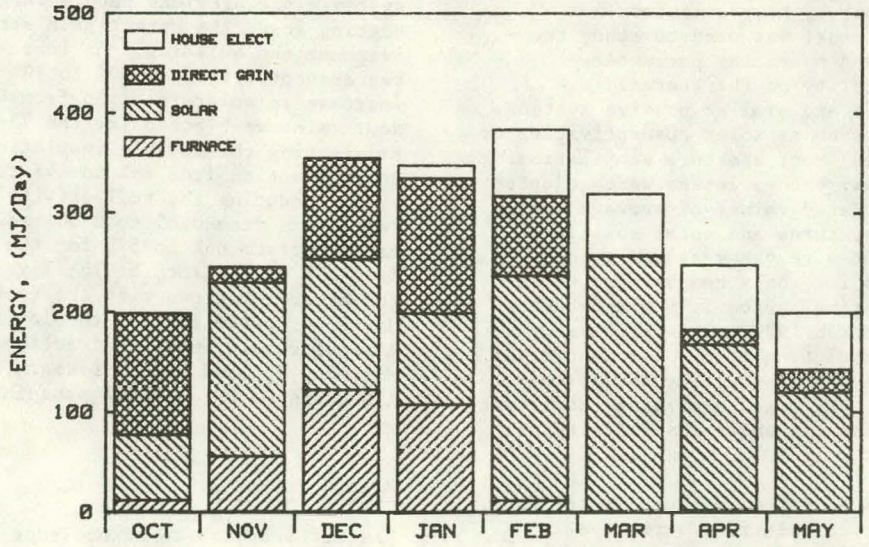


Fig. 9. Monthly average solar delivered through ceiling, direct gain, auxiliary and house electricity used during the 1978-1979 heating season.

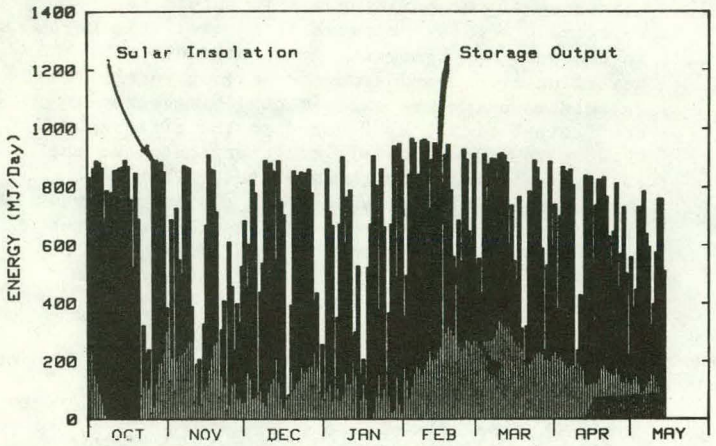


Fig. 10. Daily incident solar radiation on roof apertures and solar delivered through ceiling (from storage) for the 1978-1979 heating season.

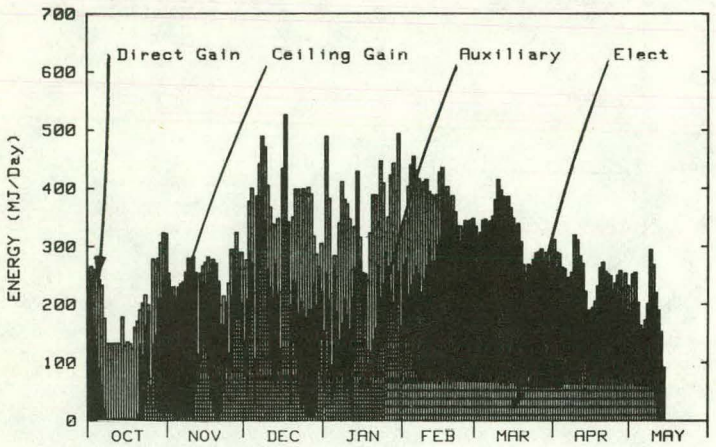


Fig. 11. Daily average solar delivered through ceiling, direct gain, auxiliary and house electricity used during the 1978-1979 heating season.

SIMULATION MODEL

A thermal network model of the mobile home was developed to do hour-by-hour computer simulations. The model was used to study the effect of design and operating parameter variations and locality on the thermal performance of this and similar passive systems. Model parameters, such as solar absorptivities of the varius surfaces, roof aperture ventilation losses and house air change losses were adjusted to bring the calculated values of average water and room air temperatures and total auxiliary heat into agreement with the measured values. The test data used for these comparisons was a nearly continuous time period from mid-December 1978 through mid-March 1979. This process resulted in the prediction of one air change per hour for the house and essentially no roof ventilation loss. The most surprising result was the requirement that the solar absorptivity of the water bags was only 40%.

Using the model as adjusted to the computer data, annual performance calculations were made in which venting was allowed to maintain a maximum room temperature of 75°F, with and without direct gain. Using Los Alamos weather data for the relatively mild 76-77 year resulted in a solar heating fraction of 86% with direct gain and 85% with the south windows blocked with insulation. For the severe 72-73 year the values were 67% and 66%. It appears that since the roof system alone provides such a large portion of the heating load, the direct gain serves only to overheat the building. An increase in the water bag absorptivity from 40% to 70% resulted in an increase in solar fraction from 66% to 83% with south windows blocked for the 72-73 year. Eliminating the movable insulation reduced the solar fraction from 66% to 44% for the same year. Reducing the reflectivity of the roof reflectors from 0.85 to 0.30 reduced the solar fraction from 66% to 57% for the 72-73 year and from 85% to 78% for the 76-77 year. Calculations for other locations with the roof apertures as designed (optimized for the latitude of Los Alamos) result in solar fractions from 39% in Madison, WI; and 34% in Bismark, ND; to 94% in Albuquerque, NM; and 99% plus in Santa Maria, CA.

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THERMAL PERFORMANCES OF SOLAR ENERGY SPACE HEATING SUBSYSTEMS

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ABSTRACT

This paper presents the space heating performance of five National Solar Data Program (NSDP) demonstration sites during the winter of 1978-79 [1-5]. The geographical location of the five installations evaluated vary from Virginia to Wisconsin; the energy transport mediums include air and liquid; one system provides solar energy to a school gymnasium, another to a single-family home. These differences provide the basis for a comparative analysis of solar energy space heating subsystems. The differences notwithstanding, there are underlying similiarities which yield an increased understanding of solar energy system performance. For the five sites in total, approximately 25 percent of the available solar energy was collected and once collected, upwards to 80 percent (of the collected energy) was usefully consumed. The solar fractions, however, vary from 13 percent to 78 percent. An explanation of this variation is found in the arithmetic ratio of heating load to solar energy available - the lower the heating load as a percent of solar energy available, the greater the solar fraction. Based upon an analysis of the five sites reviewed, the overall effectiveness of a solar energy space heating subsystem will be governed by the collector array performance and not by specific space heating methods or designs. This generality, of course, does not consider the (positive or negative) performance effects created by installation techniques, solar energy system control procedures, insulation methods, etc. Refer to Table 1 for an overview of the five solar energy installation designs and reference [6] for additional collector performance information.

INTRODUCTION

The objective of this paper is to present the solar energy performance results based upon operational data analysis of five NSDP demonstration sites. Evaluation or comparison of specific design features or manufacturer's products has not been performed; however, when possible, performance factors are standardized so that generic comparisons can be provided [7,8].

The five sets of results presented were selected because performance data was available for a recent heating season and the data was considered representative of NSDP sites incorporating space heating subsystems [9]. Monthly performance reports have been published for the U.S. Department of Energy as well as seasonal reports which documented long-term (generally six-months) performance results. The seasonal results provide

TABLE 1. SOLAR SITE DESIGN SUMMARY

SITE NAME	LOCATION/ LATITUDE	COLLECTOR AZIMUTH/ TILT	COLLECTOR TYPE/ AREA	STORAGE TYPE/ CAPACITY	HEATING METHOD	AUXILIARY ENERGY
CONCORD LIGHT	CONCORD, MA 42.47°	14°E OF S 57°	AIR 1932 FT^2	ROCK 1210 FT^3	FORCED AIR ONLY	ELECTRICITY
HOWARD'S GROVE	HOWARD'S GROVE, WI 43.13°	0 50°	AIR 2685 FT^2	ROCK 1500 FT^3	FORCED AIR ONLY	ELECTRICITY
T. D. MOSELEY	LYNCHBURG, VA 37.46°	0 50°	LIQUID 400 FT^2	LIQUID 2000 GAL	LIQUID-AIR HX HEAT PUMP	FOSSIL
SCATTERGOOD	WEST BRANCH, IA 41.70°	0 50°	AIR 2496 FT^2	ROCK 1850 FT^3	FORCED AIR ONLY	FOSSIL
M. F. SMITH	JAMESTOWN, RI 41.52°	15°W OF S 45°	LIQUID 512 FT^2	LIQUID 3150 GAL	LIQUID-AIR HX HEAT PUMP	ELECTRICITY

the basis for the data presented in this report.

Table 2 highlights month-by-month performance factors and a seasonal total for each installation. Direct site-to-site comparisons are also included in Table 3.

SOLAR ENERGY SPACE HEATING SUBSYSTEM THERMAL PERFORMANCE

The following sections highlight the design characteristics and solar energy performance measurements for each of the five installations.

Concord Light Plant

The solar energy installation at the Concord Light Plant, Concord, Massachusetts, is an air system providing space heating for a municipal garage and office area. The solar energy collection and storage subsystem (ECSS) consists of 92 flat-plate collectors and a rock storage bin. The solar energy system is designed to aid in maintaining the garage temperature at 50°F, although auxiliary energy can be provided by electrical resistance heaters. If additional solar energy is available, it is delivered to the office area, which is maintained at 68°F. The system has three modes of operation: collector-to-space-heating, collector-to-storage and storage-to-space-heating.

The weather conditions during the 1978-79 winter were below the long-term average for Concord, Massachusetts. Incident solar energy was 7 percent below the average and ambient temperatures were 3°F below average [10].

There were two space heating subsystem problems during the heating season, both of which were resolved: A leaking damper was replaced on January 16 and on February 22 the storage-to-load controls were altered to improve the effectiveness of that mode of operation.

The collector array, arranged in three parallel rows, collected 63.1 million Btu of the 242.0 million Btu of solar energy available. During the same period, 33.4 million Btu were delivered to a 1200-cubic-foot rock storage bin of which 24.1 million Btu were removed [11-13]. The coefficient of performance for the energy collection and storage subsystem was 23.2 which translates to 23.2 Btu being transported to the load for every Btu of operating energy.

The solar energy system at Concord Light satisfied 30 percent of the space heating load. The total heating load was 146.3 million Btu and solar energy contributed 44.5 million Btu directly from the collectors and from storage. The relatively low performance figure for January (15 percent solar fraction) is attributed, in part, to unusually poor weather conditions; e.g., outdoor ambient temperatures were 2°F below normal, and more significantly, incident solar energy availability was 25 percent below normal. Overall the solar energy space heating subsystem provided a net electrical energy savings of 40.5 million Btu.

Howard's Grove

The Howard's Grove solar energy system provides space heating to a 12,330-square-foot addition to the North View Elementary School in Howard's Grove, Wisconsin. The collector array incorporates 138 flat-plate panels arranged in six rows with each row containing 23 collector panels. Air is the medium that transfers the collected solar energy to a crushed-rock storage bin. The auxiliary thermal energy is supplied from a fuel-oil boiler to a liquid-to-air heat exchanger in the space heating supply duct. The solar energy system has three modes of operation: collector-to-storage, storage-to-occupied-classroom (time controlled) and storage-to-unoccupied-classroom. Classroom occupancy is significant because fresh outside air (10 percent) must be provided to each classroom when occupied.

During the October 1978 through March 1979 heating season, Howard's Grove experienced 7025 heating degree-days or 539 more than the long-term average. The 27°F actual weighted average temperature was 2°F below normal, although there was 4 percent more incident solar energy than expected.

Two noteworthy problems occurred at Howard's Grove: In November a half-closed fire damper door was allowing 50 percent of the collected solar energy to escape and, in January, 30 inches of snow accumulated on the collector array. The damper door was quickly fixed, although the snow degraded collector performance for approximately half of January, resulting in an abnormally low collector efficiency as shown in Table 2.

The overall collector efficiency for the six-month period was 14 percent, although January's efficiency of only 4 percent resulted in a reduction of overall efficiency of approximately 3 percent. A total of 75.1 million Btu of solar energy was collected out of the 518.6 million Btu incident. Solar energy delivered to storage totalled 72.3 million Btu (indicating a relatively small loss of 2.8 million Btu during the collector-to-storage operation) and 55.5 million Btu were delivered from storage.

The solar energy space heating subsystem provided 13 percent or 55.5 million Btu of the space heating load. The monthly solar contribution ranged from a high of 23 percent in March to a low of 3 percent in January. Overall, the solar energy space heating subsystem saved 73.1 million Btu of fossil fuel at an electrical expense of 3.3 million Btu.

T. D. Moseley

The Terrell D. Moseley Company solar energy system in Lynchburg, Virginia is installed in a 1780 square foot, single story, office building with an attached warehouse. The site has a collector array of 24 flat-plate collectors. Water is the heat transfer medium throughout the solar energy system and the thermal energy is stored in a 2000-gallon tank. When solar energy is insufficient to

TABLE 2. SPACE HEATING SUMMARY

SITE	PERFORMANCE FACTORS	OCT 78	NOV 78	DEC 78	JAN 79	FEB 79	MAR 79	TOTAL
CONCORD LIGHT	INSOLATION-BTU/FT^2	*	723	955	707	1440	869	(1)982
	COLLECTOR EFFICIENCY	*	.25	.32	.24	.27	.21	(1).26
	STORAGE EFFICIENCY	*	*	.72	.45	.62	.71	(1).65
	SPACE HEATING LOAD-BTU 10^6	*	12.52	34.40	41.99	48.88	21.06	146.33
	SPACE HEATING COP (2)	*	10.50	13.62	19.12	22.86	14.18	16.91
	SOLAR FRACTION-PERCENT	*	12	38	15	33	44	(1) 30
	GROSS SH SAVINGS-ELECT-BTU 10^6	*	1.47	12.94	6.31	16.00	9.22	44.47
HOWARD'S GROVE	INSOLATION-BTU/FT^2	1220	858	767	1090	1392	1062	(1)1061
	COLLECTOR EFFICIENCY	.14	.15	.16	.04	.18	.20	(1) .14
	STORAGE EFFICIENCY	.30	.50	.99	1.05	1.03	.90	(1) .78
	SPACE HEATING LOAD-BTU 10^6	19.83	47.45	88.57	107.22	90.98	66.98	421.03
	SPACE HEATING COP (2)	3.87	5.64	6.21	8.11	8.14	8.55	6.75
	SOLAR FRACTION-PERCENT	20	12	10	3	20	23	(1) 13
	GROSS SH SAVINGS-FOSSIL-BTU 10^6	5.15	7.18	11.85	4.75	23.88	20.26	73.07
T. D. MOSELEY	INSOLATION-BTU/FT^2	1577	946	1196	1046	1049	1447	(1)1210
	COLLECTOR EFFICIENCY	.18	.23	.28	.21	.18	.25	(1) .22
	STORAGE EFFICIENCY	.37	.60	.86	.87	.86	.76	(1) .72
	SPACE HEATING LOAD-BTU 10^6	.87	3.26	8.86	12.83	13.02	5.57	44.41
	SPACE HEATING COP (2)	14.50	8.15	8.12	8.86	9.38	8.77	8.75
	SOLAR FRACTION-PERCENT	100	83	46	19	15	49	(1) 33
	GROSS SH SAVINGS-FOSSIL-BTU 10^6	1.45	4.48	6.76	4.13	3.29	4.54	24.65
SCATTERGOOD	ISOLATION-BTU/FT^2	1313	866	844	868	1486	1025	(1)1061
	COLLECTOR EFFICIENCY	.46	.28	.33	.27	.31	.31	(1) .34
	STORAGE EFFICIENCY	.57	.78	.83	.81	.88	.85	(1) .81
	SPACE HEATING LOAD-BTU 10^6	4.50	12.46	17.27	19.51	29.05	19.58	102.37
	SPACE HEATING COP (2)	47.42	125.16	41.56	44.74	40.92	53.63	44.51
	SOLAR FRACTION-PERCENT	93	92	81	58	73	84	(1) 78
	GROSS SH SAVING-FOSSIL-BTU 10^6	7.50	19.39	25.54	18.86	35.61	27.90	134.80
M. S. SMITH	INSOLATION-BTU/FT^2	1668	985	1060	862	1405	1314	(1)1216
	COLLECTOR EFFICIENCY	.28	.27	.33	.34	.33	.40	(1) .33
	STORAGE EFFICIENCY	.05	.40	1.13	1.36	1.18	.87	(1) .82
	SPACE HEATING LOAD-BTU 10^6	.36	3.84	8.86	9.64	*	*	22.70
	SPACE HEATING COP (2)	5.14	5.60	5.90	4.25	*	*	5.05
	SOLAR FRACTION-PERCENT	100	80	73	67	*	*	(1) 72
	GROSS SH SAVINGS-ELECT-BTU 10^6	.32	1.46	2.56	1.09	*	*	5.43

*DATA NOT AVAILABLE

(1) AVERAGE

(2) COEFFICIENT OF PERFORMANCE (COP) IS DEFINED AS THE TOTAL SOLAR ENERGY DELIVERED TO THE LOAD DIVIDED BY THE SUBSYSTEM OPERATING ENERGY.

TABLE 3. HEATING SEASON TOTALS

SITE NAME	SOLAR ENERGY AVAILABLE (INCIDENT)	EXPRESSED AS % OF SOLAR ENERGY AVAILABLE				SOLAR FRACTION
		SOLAR ENERGY COLLECTED	SOLAR ENERGY FROM STORAGE	SOLAR ENERGY CONSUMED	HEAT LOAD	
CONCORD LIGHT	100%	26%	10%	19%	64%	30%
HOWARD'S GROVE	100%	14%	12%	12%	71%	13%
T. D. MOSELEY	100%	22%	*	17%	50%	33%
SCATTERGOOD	100%	34%	11%	17%	21%	78%
M. F. SMITH	100%	33%	23%	23%	31%	72%

*DATA NOT AVAILABLE

maintain a 105°F storage temperature, a gas-fired boiler provides auxiliary energy to storage. Space heating is provided by circulating storage water through a heat exchanger located in the building's air distribution system. There are three modes of operation: collector-to-storage, storage-to-space-heating and storage-to-heat-pump.

The weather conditions were below normal for Lynchburg during the winter of 1978-79. Incident solar energy was 3 percent below the long-term average and the ambient temperature averaged 43°F or 1°F lower than normal.

There were two significant problems during the reporting period. In November, the gas-fired heater differential control failed to activate the circulating pump and, therefore, the heat pump was the primary source of thermal energy. In December, the pilot light was out on the gas-fired heater, and again, the heat pump was the primary source of thermal energy.

A total of 88.4 million Btu was incident upon the 24 parallel flow, flat-plate panels and 19.6 million Btu were collected. The coefficient of performance, or ratio of solar energy collected to the associated operating energy, was 10.6. Total thermal energy to the 2000-gallon storage tank was 55.7 million Btu and the total energy from storage equalled 44.2 million Btu.

The solar energy system at the T. D. Moseley installation provided 33 percent of the space heating load, ranging from 100 percent in October to 15 percent in February. The space heating load was 44.4 million Btu, for which solar energy provided 14.8 million Btu. The relatively low solar fraction attained in February (as compared to the other months) may be, in part, attributed to the weather conditions. Ambient temperatures were 8°F below normal, increasing the heating degree-days by 30 percent; and the available incident solar energy was 18 percent below average, reducing the system's capability to collect solar energy.

Total system savings for the 1978-79 heating season were 24.7 million Btu of fossil fuel at an electrical energy expense of 2.7 million Btu.

Scattergood

The Scattergood School solar energy system is located in West Branch, Iowa, 35 miles southeast of Cedar Rapids. The system is designed to provide space heating for a gymnasium, hot water for a locker room, and thermal energy for grain drying. The site has an array of 128 flat-plate collectors with a gross area of 2496 square feet. Collected solar energy is stored in a pebble bed for space heating purposes. There are three modes of operation pertinent to space heating: collector-to-storage, collector-to-space-heating and storage-to-space-heating.

During the October 1978 through March 1979 time period, the weather conditions at Scattergood were significantly below average. Daily incident solar energy was 90 percent of the long-term average; there was an average ambient temperature of 27°F versus the long-term average of 33°F, resulting in 6901 heating degree-days as opposed to the long-term average of 5749.

Two problems were identified which degraded the collector array performance efficiency: Snows in January covered the array temporarily, and more significantly, thermal energy leakage from the collector energy transport system persisted throughout the reporting period and resulted in a significant solar energy loss. However, the collector array efficiency was 34 percent, as 161.6 million Btu were collected out of the 482.2 million Btu incident.

The solar energy delivered to the pebble storage equalled 62.5 million Btu and 50.8 million Btu were withdrawn from storage.

The space heating subsystem supported an 102.4 million Btu load, of which 79.6 million Btu or 78 percent was satisfied by solar energy. The fossil

fuel savings for the six-month period totalled 134.8 million Btu at an electrical energy expense for space heating of 2.3 million Btu. The energy collection and storage subsystem consumed an additional 2.8 million Btu of electrical energy, although this expense is necessary to support hot water preheating and grain drying.

M. F. Smith

The M. F. Smith Associates solar energy system is installed at a 1752 square foot, single-family, residence in Jamestown, Rhode Island. The solar energy system is designed for space heating and preheating domestic hot water. Because the house was unoccupied during the 1978-79 heating season, there was no appreciable hot water load. The solar energy system incorporates a 512-square-foot flat-plate collector array. Water transfers thermal energy from the collector array to a 3150-gallon storage tank and to a space heating subsystem consisting of a solar-assisted heat pump and a liquid-to-air heat exchanger. When the storage water temperature is insufficient to satisfy the heat pump requirements, an electrical heating element provides auxiliary energy to the heat pump. The two modes applicable to space heating are collector-to-storage and storage-to-space-heating.

The weather during the October 1978 through March 1979 time period was normal: average actual temperatures of 38°F were identical to the long-term average; there were 806 actual heating degree-days versus the long-term average of 810; actual incident solar energy equalled 1216 Btu per square foot versus the long-term average of 1191 Btu per square foot for this geographical area.

Only one problem related to the solar energy system persisted throughout the reporting period: The freeze-protection control system allowed the drain-down solenoid valves to remain open when the collector subsystem was operational. Because the drain-down system parallels the collector loop, i.e., drains from the collector array to storage, this problem does not significantly degrade the collector efficiency.

The collector array coefficient of performance was 37.2. A solar energy total of 113.1 million Btu were incident and 36.9 million Btu were collected from the 512-square-foot flat-plate collector array. Solar energy delivered to storage totalled 36.6 million Btu and the energy removed from storage equalled 31.2 million Btu. During three months, thermal energy was added to the storage tank from either the surrounding ambient air and/or an auxiliary thermal energy source, making energy balances difficult to compute.

The solar energy space heating subsystem satisfied 72 percent of the heating load for a four-month period when space heating data was available. Because the house was unoccuppied (and the average building temperature was as low as 62°F), the heating load should be considered below normal; however, the solar energy system, satisfied 67 percent of the heating load during the coldest month. For the period October 1978 through January 1979, the electrical energy net savings derived from solar energy totalled 4.8 million Btu.

SUMMARY AND CONCLUSIONS

The five systems reviewed were selected because they represent NSDP installations providing complete, comparable data for the 1978-79 heating season. They also performed above average when compared to 23 other solar energy installations analyzed for the 1978-79 heating season. The five systems differ significantly in all elements of design (see Table 1), however, analysis (see Table 2) reveals no identifiable design advantages or disadvantages. This conclusion suggests that the differences among the varying system performances may be attributable to component sizing, installation techniques and solar energy system control procedures.

Upon reviewing the data presented in the three tables, some comparisons and results, based on this limited sample, can be observed:

Of the two systems with the best solar fraction (Scattergood and M. F. Smith), one is an air system using forced air for space heating and the other is a liquid system using a solar-assisted heat pump. Each, however, has a relatively effective collection and storage subsystem. Thirty-four percent of the incident solar energy was collected at Scattergood and 33 percent at M. F. Smith. The storage efficiency was 81 percent at Scattergood and 82 percent at M. F. Smith.

It was not possible to equitably relate collector size to cubic feet of conditioned space. However, heat load can be expressed as a percent of solar energy available and then compared to solar fraction (see Table 3). Note that the lower the heat load percent, the higher the solar fraction. This numerical relationship suggests that a poorly designed or inadequately installed space heating subsystem could, in effect, be compensated for with an oversized collection subsystem. A comparison of the losses at Howard's Grove and Scattergood reveals further insight -- 20 percent losses at Howard's Grove and 50 percent at Scattergood. The solar fractions, however, were 78 percent at Scattergood and 13 percent at Howard's Grove. The difference was the quantities of energy collected and the collector array efficiencies.

Howard's Grove and Scattergood had identical average insolation and average temperatures (1061 Btu per square foot and 27°F); plus the storage efficiencies approximated each other (78 percent at Howard's Grove and 81 percent at Scattergood). However, the collector efficiency made the difference. Scattergood collected 34 percent and Howard's Grove collected 14 percent.

The data presented in this report highlights the fact that space heating subsystem performance can only be as effective as the collector and storage subsystem performance. There are many variables and economic considerations; however, the collect-

or array efficiency will generally govern the effectiveness of a solar energy space heating subsystem -- regardless of space heating equipment type, storage methods or energy transfer media.

ACKNOWLEDGEMENT

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THERMAL PERFORMANCE OF A COMMERCIALLY AVAILABLE STATE-OF-THE-ART SOLAR AIR-HEATING SYSTEM

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ABSTRACT

Performance of a "state-of-the-art" air-type solar heating system in CSU Solar House II is evaluated for the 1978-79 heating season. The system includes 32 modules (57.9 m^2) of Solaron series 3000 collectors, two blowers (one to collect and store solar heat and a second to distribute thermal energy from storage), 10.3 m^3 of pebble bed storage, domestic water preheating capability, and a 3-ton Carrier air-to-air heat pump for auxliary space heating. An average daily collection efficiency of 33 percent was achieved, with operating collection efficiency of 38 percent (based on solar irradiance while the system was operating). Overall solar COP for useful heat delivered to space and hot water was 13.5 consisting of COP for collection of 26.3 and COP for distributing solar heat of 36.0. Average heating COP for the heat pump was 1.60 which is low because the heat pump is required only after solar storage is nearly exhausted, hence it operates when ambient air temperature is low.

INTRODUCTION

Solar House II, in the Solar Village at Colorado State University, has been utilized to develop and evaluate solar air heating systems. The first system was constructed at the site and installed in 1975 and thermal performance of that system has been reported previously [1]. During September and October 1978 the site-built collectors were replaced with "state-of-the-art" manufactured collectors and the single blower design was changed to a dual blower arrangement, utilizing one air handler to collect and store solar heat and the air handler of a heat pump to distribute solar heat from storage to the building. Thermal performance data of the system were taken from December 1978 through mid-May 1979. This paper is concerned with the experimental performance results of the state-of-the-art solar air-heating system.

SYSTEM DESCRIPTION

A schematic drawing of the system in CSU Solar House II is shown in Fig. 1. There are 57.9 m^2 (gross area including perimeter insulation of 5 cm thickness) of Solaron series 3000 collectors on the of of the building. Excluding perimeter insulaon, the gross area is 55.7 m^2 and the aperture area is 50.8 m^2. There are 10.3 m^3 of pebble-bed storage located in the basement. Interconnecting ducts are well-sealed and insulated. There are two air-to-water heat exchangers for preheating domestic water; one is for winter operation and the other is for summer use. The auxiliary heater is a 3-ton Carrier air-to-air heat pump with 11.5 kW of back-up resistance heating capability.

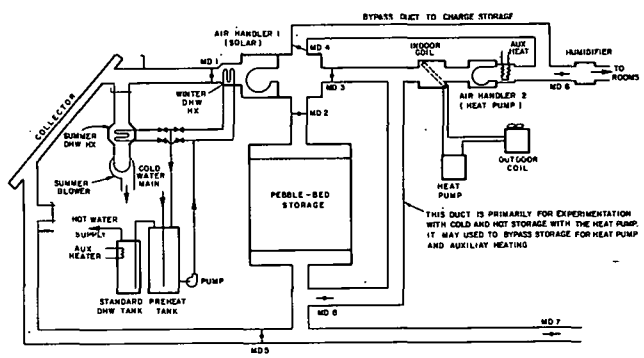


Fig. 1. Schematic Arrangement of the Solar System in CSU Solar House II

Collectors

The series 3000 Solaron collector consists of a steel absorber plate, which is part of the air channel in the collector, with a selective coating of black chrome over nickel. Representative absorptance of the coating is 0.95 and emittance is 0.15. The single glazing is double strength, low iron, tempered glass, which is lightly patterned, and has a transmittance of 0.89. The back side of the collector is insulated to achieve an R-value of about 13.

Installation - The collectors are assembled in a two-high array as shown in Fig. 2. To minimize ducting and associated costs, Solaron collectors are designed with interior manifolding as illustrated in Fig. 3. There are also lateral connections across the array below the air channel. The patented interior manifold design requires only one supply duct connection and one return duct connection for an array with as many as 30 modules. This

simplification in duct connections to the collector array represents a trade-off of both material and installation labor cost for ducts against added material cost of a deep collector design to accommodate the internal manifold.

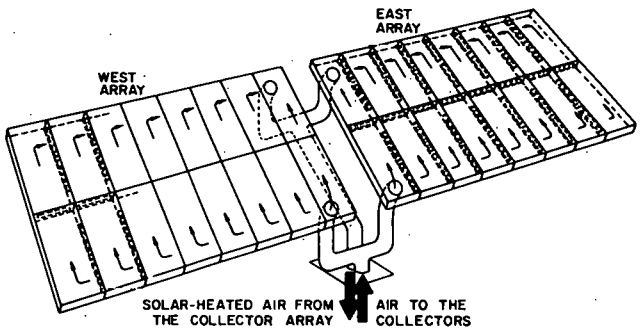


Fig. 2. Collector Array on CSU Solar House II

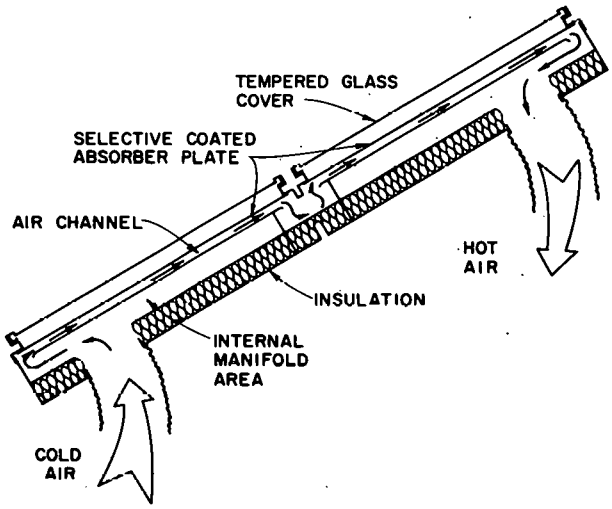


Fig. 3. Cross-section of Solaron Collectors in a Two-high Array

In the Solar House II system, the collector array is divided into two equal sections of 16 collector modules each and separate supply and return connections are made to the two arrays. The division was made for experimental reasons, and is only a slight variation from manufacturer's recommendations.

There were two methods used in assembling each 16-module collector array. In the west array all the collector modules (16) were lifted to the roof first, the cherry picker, which was used to lift the collectors, was discharged and the modules were then moved into position by the installers and fastened to the roof. For the east array, collector modules were lifted to the roof as needed and fastened into position. The first method minimizes use time of the mechanical lifting equipment, but the second minimizes overall installation time. Including pre-installation preparation time, for such functions as unpacking the collectors, cleaning the outer glazing, etc., the 16 collector modules in the west array were lifted to the roof in 2½ hours, and the array was completed in 10 additional hours The east array was totally installed in 9½ hours.

Performance - Air leakage rates into the two collector arrays are different, and the leakage rates vary with the mode of operation because of different pressure balances in the system. During December, 1978, and through January 6, 1979, the total air flow rate at the outlet from the collectors was about 45.4 m^3/min (referenced to local blower conditions) with a variation of 2.5 percent depending on mode of operation. Unit flow rate was 13.1 ℓ/sec per square meter of collector, which is between 10.2 and 15.3 ℓ/(sec·m^2) recommended by the manufacturer. Under this condition, leakage in the west collector array was 9 percent when heating the rooms directly from the collector and 5 percent while collecting heat in storage. Air leakage rate in the east array was 22 percent while heating the rooms and 13 percent while storing heat.

On January 7, 1979 the air flow rate through the collectors was reduced by 27 percent to 9.6 ℓ/(sec·m^2) which raised the temperature of air delivered to storage and decreased collection efficiency. With the reduced air flow rates, leakage in the east collector array was 18 percent during direct heat delivery and 6 percent during heat storage. The west array was less at 7 and 5 percent respectively.

Scatter diagrams of measured "instantaneous" collector efficiencies (5 minute interval of integration) are shown in Fig. 4. The lines on the figure are drawn from values of F_R, τ, α and U_L (collector heat removal factor, transmittance, absorptance, and heat loss coefficient respectively) developed by the manufacturer which are based on collector test data obtained at the Desert Sunshine Exposure Test Center (DSETC), from tests that were conducted in accordance with recommended procedures described in ASHRAE 93-77. Values of $(\tau\alpha Aa/Ag)$ for the Solaron series 3000 collector is given as 0.744, where Aa is aperture area and Ag is gross area. The ratio Aa/Ag is 0.912 for the Solaron collector and the heat loss coefficient, U_L, is 7.12 W/(m^2·°C).

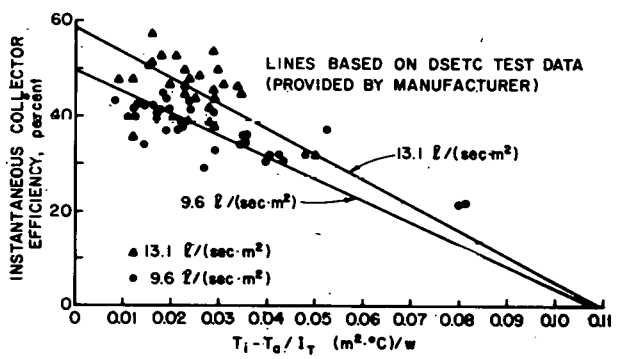


Fig. 4. Measured Instantaneous Collector Efficiencies Near Solar Noon

orage

A pebble bed storage unit consisting of 10.3 m^3 of 2- to 4-cm diameter gravel was used in the system. Air flows vertically through the pebble bed. The top plenum is 25 cm high and the bottom plenum is 20 cm high which also contains bond beams and heavy screen to support 1.68 m depth of rock above the screen.

Vertical temperature profiles at the center of the pebble bed are shown in Figs. 5 and 6 for typical winter days when all the thermal energy stored during the day is utilized during the night. Air flow rates through storage while being charged and discharged are listed in Table 1.

TABLE 1. STORAGE AIR FLOW RATES, SUPERFICIAL VELOCITIES AND PRESSURE DROPS

ITEM	BEFORE 1/7/79	AFTER 1/7/79
Flow rate while charging storage	47.5 m^3/min	35.1 m^3/min
Superficial velocity	7.74 m/min	5.72 m/min
Pressure head drop	4.6 mm water	2.5 mm water
Flow rate while discharging storage	31.0 m^3/min	31.0 m^3/min
Superficial velocity	5.05 m/min	5.05 m/min
Pressure head drop	2.0 mm water	2.0 mm water

Maximum temperatures of heat delivery from storage on a typically cold day in December when ambient air temperature was -10 °C, was about 44 °C. After reducing the flow rate through the collector in January, higher collector and storage temperatures resulted and after a comparable day of heat collection at -10 °C ambient air temperature, the maximum temperature of the circulating air to the rooms from storage was 54 °C.

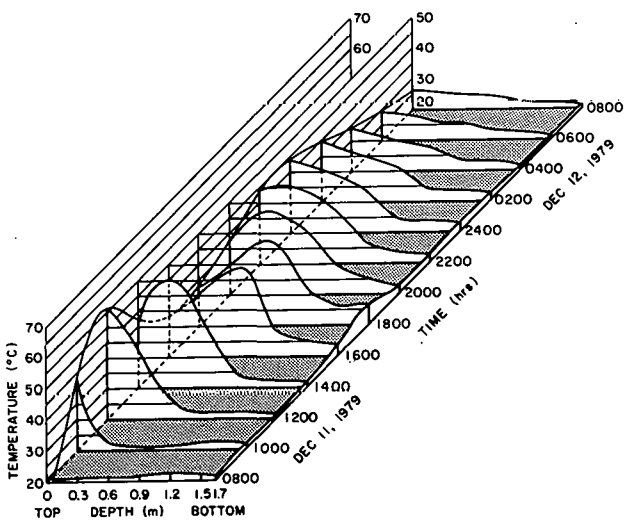


Fig. 5. Temperature Profiles in the Pebble Bed During Charge/Discharge Cycle. Air Flow Rate 13.1 ℓ/(sec·m^2)

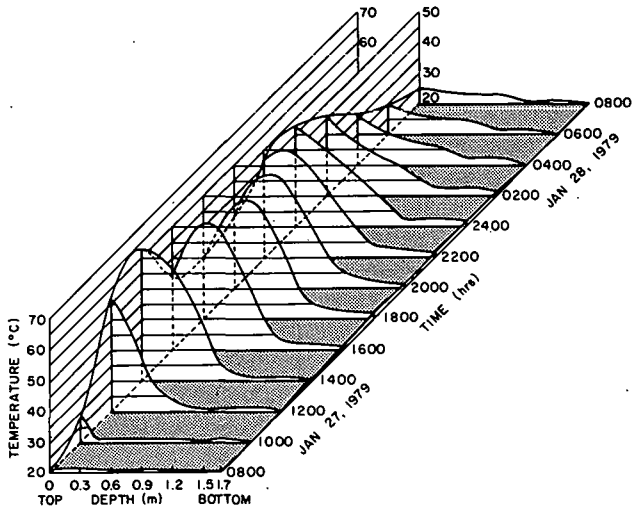


Fig. 6. Temperature Profiles in the Pebble Bed During Charge/Discharge Cycle. Air Flow Rate 9.6 ℓ/(sec·m^2)

SYSTEM PERFORMANCE

Collection of Solar Heat

System performance summary for 5½ months of the 1978-79 heating season is provided in Table 2. The amount of solar heat collected in relation to total measured solar radiation on the plane of the collector, and radiation while collecting solar heat, are shown in the bar graph of Fig. 7. Collection efficiency during December is higher than in other months because of the relatively high air flow rate through the collector. However, COP for solar heat collection is lowest in December at 20.5.

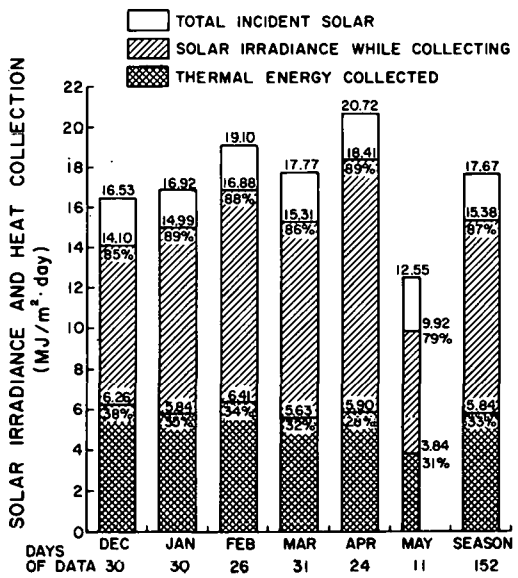


Fig. 7. Solar Irradiance and Thermal Energy Collected

TABLE 2. CSU SOLAR HOUSE II PERFORMANCE SUMMARY HEATING SEASON 1978-79

No.	Item	Units	Dec.	Jan.	Feb.	Mar.	Apr.	May	Season
1	Incident Solar Radiation	MJ/(m^2·day)	16.53	16.92	19.10	17.77	20.72	12.55	17.67
2	Incident Solar While Collecting	MJ/(m^2·day)	14.10	14.99	16.88	15.31	18.41	9.92	15.38
3	Thermal Energy Collected*	MJ/(m^2·day)	6.26	5.84	6.41	5.63	5.90	3.84	5.84
3a	Thermal Energy Collected	MJ/day	362.59	338.14	371.22	325.89	341.35	221.16	338.27
4	Collector Operating Efficiency	(3)/(2)	0.44	0.39	0.38	0.37	0.32	0.39	0.38
5	Collector Daily Efficiency	(3a)/(1)	0.38	0.35	0.34	0.32	0.28	0.31	0.33
6	Total Heating Load	MJ/day	476.83	597.54	439.92	358.61	248.50	304.59	421.71
6a	Space Heating Load	MJ/day	436.69	538.74	375.79	287.01	176.50	246.50	361.04
6b	DHW Load	MJ/day	40.14	58.80	64.13	71.60	72.00	58.09	60.67
7	Solar Energy Utilized	MJ/day	307.88	303.27	330.47	267.88	206.74	190.35	278.20
7a	Space Heating	MJ/day	288.01	275.77	292.17	229.55	159.92	170.29	245.64
7b	DHW Preheating	MJ/day	19.87	27.50	38.29	38.32	46.82	20.05	32.56
8	Auxiliary Thermal Energy	MJ/day	168.95	294.27	109.46	90.74	41.76	114.35	143.52
8a	Heat Pump, Space	MJ/day	72.96+	117.32	52.63	47.25	14.67	51.10	61.92
8b	Electrical Resistance, Space	MJ/day	75.72	145.65	30.99	10.21	1.91	25.21	53.20
8c	DHW, Electrical Resistance	MJ/day	20.27	31.30	25.84	33.28	25.18	38.04	28.11
9	Electrical Energy to Collect	MJ/day	17.67	12.49	11.91	10.84	13.10	8.24	12.87
10	Electrical Energy to Dist. Solar	MJ/day	10.54	11.66	7.30	5.96	2.40	7.05	7.73
11	Electrical Energy for Heat Pump	MJ/day	55.42+	79.37	28.24	23.32	8.10	24.52	38.81
12	Electrical Energy for HP Blower †	MJ/day	10.54	11.66	7.30	5.96	2.40	7.05	7.73
13	Solar System Efficiency	(7)/(1·A_c)	0.32	0.31	0.30	0.26	0.17	0.26	0.27
14	COP to Collect Solar	(3a)/(9)	20.5	27.1	31.2	30.1	26.1	27.0	26.3
15	COP to Distribute Solar	(7)/(10)	29.2	26.0	45.3	45.0	86.1	24.2	36.0
16	COP Overall Solar	(7)/(9/10)	10.9	12.6	17.2	13.7	13.3	12.5	13.5
17	COP Heat Pump	(8a)/(11)	1.32	1.48	1.86	2.03	1.81	2.08	1.60
18	COP Auxiliary Heating	(8a+8b)/(11+8b+12)	1.05	1.11	1.26	1.46	1.34	1.34	1.15
19	Overall Heating COP	(6)/(8b+8c+9+10+11+12)	2.51	2.05	3.94	3.20	4.68	2.77	2.84
20	Number of Days of Measurement		30	30	26	31	24	11	152
21	Solar Fraction, Total Load	(7)/(6)	0.65	0.51	0.75	0.75	0.83	0.62	0.66
22	Solar Fraction, Space Heating	(7a)/(6a)	0.66	0.51	0.78	0.80	0.91	0.69	0.68
23	Solar Fraction, DHW	(7b)/(6b)	0.50	0.47	0.60	0.54	0.65	0.35	0.54
24	F-Chart Solar Fractions		0.65	0.50	0.75	0.80	1.00	0.63	0.68
25	Difference of Calculations from Measurements	Percent	0	-2	0	+7	+20	+2	+3

* Based on gross area of collector, A_c, 57.9 m^2
\+ Based on 26 days
† One-half of total electrical energy to distribute heat to the rooms is "charged" to the solar system

Solar Contribution to Load

The contribution of solar energy to the total load for the season was 66 percent. As expected, solar fraction is lowest in January, which is historically the coldest month of the year. The solar fraction increases steadily from January through April as the total heating load decreases while the average daily solar heat collected varied only slightly through the season. Figure 8 is a bar chart of the solar and auxiliary contributions to the total load for the season.

Space Heating

Contributions of solar, heat pump and electrical resistance heating to the total space heating load are shown in Fig. 9. While solar heating provides a major portion of the load, electrical resistance heating was also dominant in December and January. The average COP of the heat pump was 1.60 (EER=5.46) for the season, with a low in December of 1.32 (EER=4.51) and high in May of 2.08 (EER=7.10). The COP for the heat pump is low because it is called to operate during unfavorable conditions when solar heat has been depleted from storage and outside air temperature is generally low. An overall "COP" for auxiliary heating (line 18 in Table 2) is 1.15 for the season. From a low of 1.05 in December, conditions improve slightly in the spring, but was never greater than 1.46. The heat pump COP as a function of ambient air temperature is shown in Fig. 10. Overall COP for the combined solar and auxiliary heat pump with electrical resistance back up was 2.84.

Comparisons with f-Chart

Measured and predicted monthly and seasonal solar fractions with the f-chart method are shown in Fig. 11. The calculated solar fractions are based on measured values of solar radiation, ambient air temperatures and heating loads. Collector parameters described by the lines in Fig. 4 were used for the calculations. The agreement between measured and calculated values is excellent. When average solar radiation and average ambient air temperatures for Fort Collins, Colorado is used [2], and heating loads calculated from average heating degree days (12.8 °C base) for Fort Collins [2] are used to predict solar fractions, the differences are more pronounced.

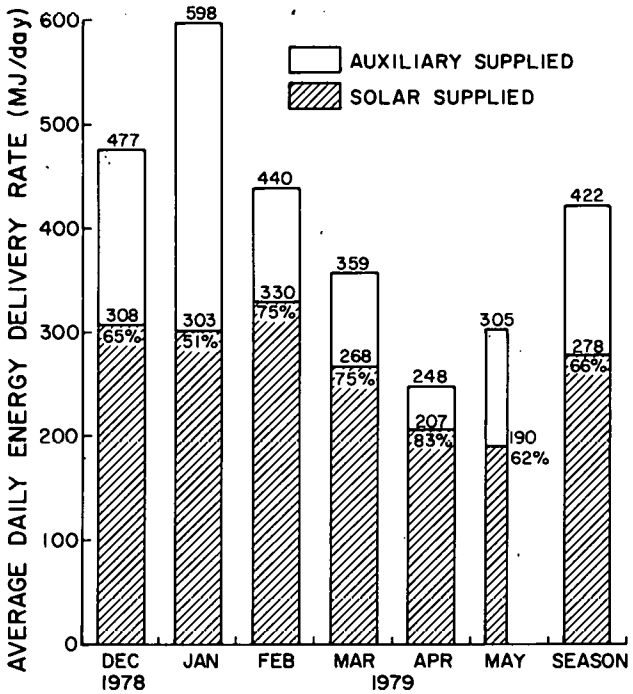


Fig. 8. Solar and Auxiliary Contributions to Total Space and Water Heating Loads

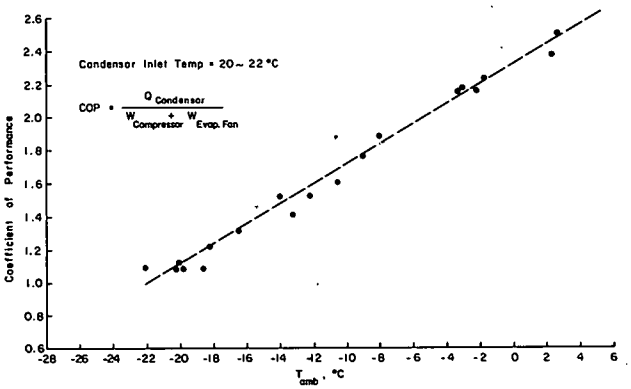
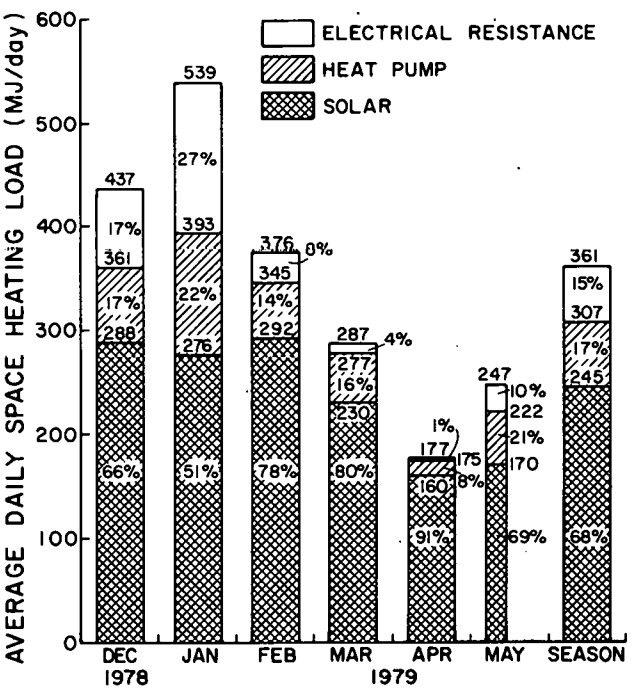


Fig. 10. Heat Pump COP as a Function of Ambient Air Temperature



g. 9. Solar, Heat Pump and Electrical Resistance Heating Contributions to Space Heating Load

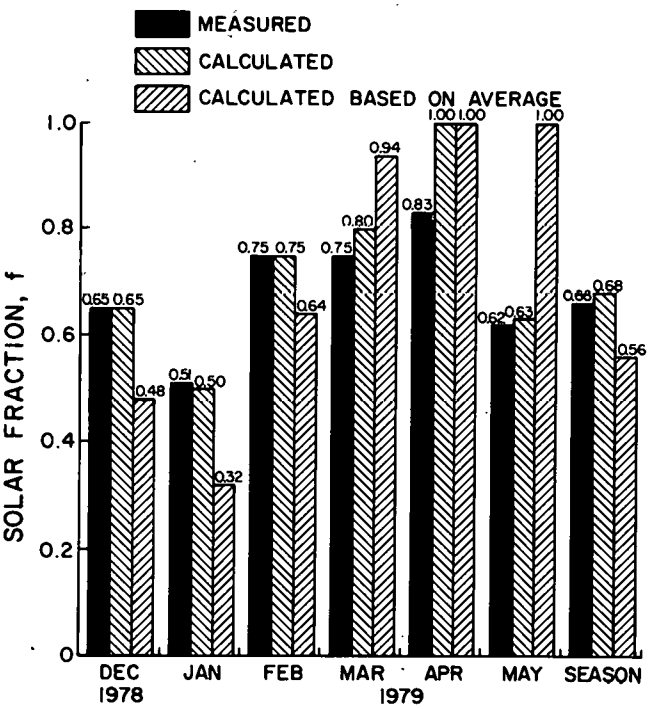


Fig. 11. Comparison of f-Chart Predicted and Measured Solar Fractions for CSU Solar House II (1978-79)

COMMENTS

A state-of-the-art solar system has been tested and evaluated in CSU Solar House II during the 1978-79 season. Collector efficiency is a function of air flow rate with efficiency increasing with air flow rate. For a given system, with fixed duct and storage sizes, there is a trade-off of collector efficiency with added parasitic energy costs to circulate greater air flow rates. Also the depth (length of flow path) in storage must be sufficient to enable effective heat transfer to the pebbles with minimum temperature of the air which recirculates to the collectors. As air flow rate increases, the temperature of the air delivered from the collector decreases which could influence the auxiliary heating required.

Leakage of air into collector arrays does not adversely affect an air-heating system. Air leakages of 5 and 20 percent existed in the west and east halves respectively. Leakages across closed dampers are significant, particularly when a damper is installed in a bidirectional duct and the reverse side of the damper is subjected to increased pressure. In such instances it is recommended that two dampers be placed in the duct section so as to be effective for flows in both directions.

A standard heat pump installed as an auxiliary heating unit with a solar system will tend to have low seasonal COP because the heat pump must operate during generally adverse conditions. The back-up resistance heating unit is called to service frequently in the cold months because the heat pump operates when ambient air temperature is low, energy extraction from the air is poor and there is frequent frosting of the outdoor coil.

CONCLUSIONS

Evaluation of a state-of-the-art air-heating solar system with an air-to-air heat pump auxiliary in CSU Solar House II indicates that:

1. A monthly average daily collection efficiency of 38 percent can be achieved and operating efficiency of 44 percent is attainable with a state-of-the-art solar air heating system.

2. Solar system efficiency for the season was 27 percent.

3. Overall solar COP for collection and distribution of 13.5 was attained, with a collection COP of at 26 and distribution COP (from storage) of 36.

4. Overall heating COP for the season was 2.84.

5. Average heat pump COP for the season was 1.60.

ACKNOWLEDGMENT

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AIR LEAKAGE EFFECTS ON ACTIVE AIR-HEATING SOLAR COLLECTOR SYSTEM PERFORMANCE

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Abstract

Air leaks into air-based solar-heating systems are a reality and it does not appear that total elimination of the leaks is economically feasible or desirable in some applications. This paper presents a method of determining the worst case effect of collector duct leakage on instantaneous system performance and suggests a procedure for estimating the worst case effect of leakage on long-term performance.

Although air leaks into a collector array can be beneficial in some applications, depending on the nature of the load, analysis is carried out in this paper without these beneficial effects included.

Introduction

In recent years increased attention has been paid to the effects of air leakage on solar system performance.

The DOE-HUD Solar Demonstration Program has discovered rather large amounts of air leakage at various points in solar air-heating collector systems. The effect of these leaks on system performance have not yet been determined, but efforts are underway to determine effects on system performance.

Close and Yusoff [1], studied the effects of air leakage of collector performance and collector performance testing. They considered heating of ambient air as a legitimate load on the collector. Using this consideration, collector air leaks increased the efficiency of the collector.

Rask [2], et al., studied leakage effects on a solar heating system and measured leak rates of 30% in the collector before doing additional caulking. A computer simulation of the system showed a decrease in solar contribution with leakage.

Elkin and Cash [3], modeled a solar heating system in a study of leakage effects. Collector outputs were calculated using the equations of Close [1] which considers heating of leaking air part of the load.

Karaki [4], reported no noticable change in system performance after modifications on CSU House II which resulted in increased leakage. FCHART underpredicted performance of the house by 12 to 14% with 40% leakage in the collector.

Shingleton [5], in a TRNSYS simulation with leakage studied the effect of air leaks on performance and concluded that external air leaks can have an appreciable effect on performance. In this study the mechanical room was completely outside the heated space which is rarely the case.

Performance Reporting

The net effect of air leakage on system performance has been reported in a variety of ways including:

1. Increase in auxiliary energy.
2. Percent increase in auxiliary energy.
3. Decrease in solar contribution.
4. Decrease in solar fraction.
5. Percent decrease in solar fraction.

The reporting method used makes a big difference in how the final results appear. For example, if a house had a heating load of 100 MBTU and the solar fraction was 25% with no leakage, the auxiliary would supply 75 MBTU. A leak which caused a 10 MBTU loss would increase auxiliary energy use 13.3% and decrease solar fraction to 15%. But, if the original solar fraction was 90%, a 10 MBTU leak would increase auxiliary energy by 100% from 10 MBTU to 20 MBTU and decrease solar fraction to 80%.

The performance quantity of interest to consumers and engineers is the actual auxiliary energy consumption which may also be derived from the solar fraction and overall heating load. Expressing leakage effects in terms of percent changes in auxiliary energy use or the percent decrease in solar fraction is essentially taking a percent of a percent and can greatly skew the results depending on what the original solar fraction was. The best measure of leakage effects appears to be the actual decrease in solar fraction.

A second rule of judging effects of leakage is to keep the load constant. If the leakage through the solar system is to be considered as part of the load, it should be consistant with the make-up air requirement of the space and not be changed every time the collector leakage rate is changed.

Theoretical Analysis

Consider the solar heating system in Figure 1, which depicts a standard solar air system with leakage into the collector and out from the heated space. Thermal storage and mechanical equipment are assumed to be within the thermal envelope which is the case in 90% of Solaron installations and is the condition assumed by FCHART [7].

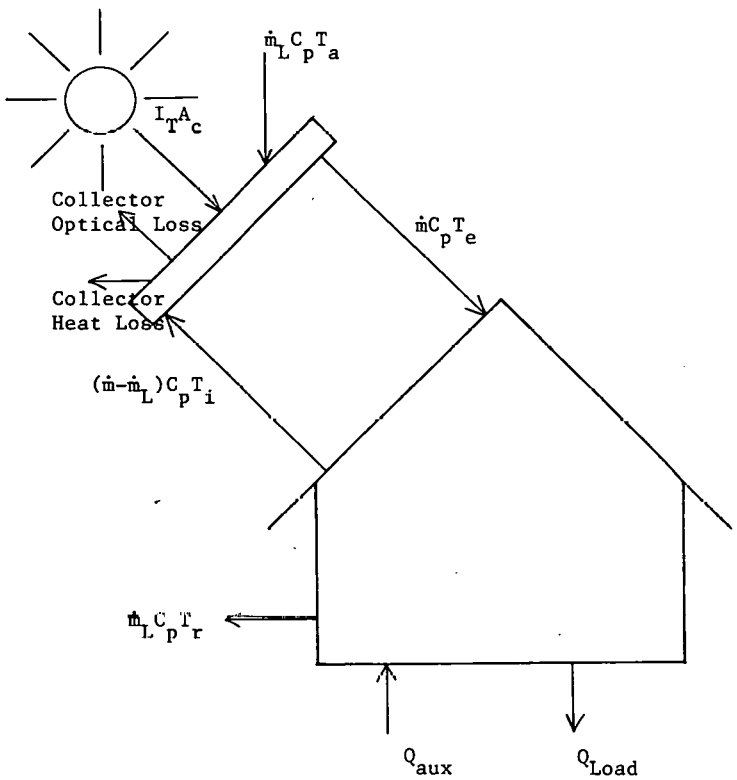


Figure 1.
Solar Heating System Schematic

Symbols used are generally based on [8]. Doing an instantaneous heat balance on the heated space:

$$\dot{m} C_p T_e + Q_{aux} - Q_{load} - (\dot{m}-\dot{m}_L) C_p T_i - \dot{m}_L C_p T_r = \frac{dU}{d\theta} \quad [1]$$

where, $\dot{m}$ is flow rate exiting collector.

C_p is specific heat of air.

T_e is collector exit fluid temperature.

Q_{aux} is auxiliary energy used.

Q_{load} is heating load.

$\dot{m}_L$ is leak flow rate.

T_i is collector inlet fluid temperature.

T_r is heated space temperature.

$\frac{dU}{d\theta}$ is rate of change in internal energy.

Useful energy collected (Q_u) may be defined as:

$$Q_u = \dot{m} C_p T_e - (\dot{m}-\dot{m}_L) C_p T_i - \dot{m}_L C_p T_r \quad [2]$$

Rearranging and dividing by collector array area:

$$q_u = GC_p (T_e-T_i) - \% GC_p (T_r-T_i) \quad [3]$$

where, q_u is useful energy collected per unit area.

G is mass flow rate per unit area.

% is ratio of leak flow rate to collector exit flow rate.

Q_u accounts for all energy flows associated with the solar system including increased exfiltration from the heated space induced by leaks into the collector array.

ASHRAE has devoted much effort to understanding infiltration and their findings are summarized below [6]. Without the effects of a heating system, infiltration air flow rate depends on the pressure difference between inside and outside and on the resistance to air flow through openings in the building. Pressure differences are due to wind or to density differences (often called chimney or stack effect). Wind around a building creates regions of different pressure resulting in inflow of air on the windward side and outflow on the leeward side. Stack effect also results in both inflow and outflow from a space. Mechancial ventilation which brings in outside air (or leaks into the collector array) increase the pressure inside the building. When inside and outside pressures on the windward side are balanced, the outdoor air required will be roughly equal to the infiltration that would otherwise occur, unless building interior separations are tightly sealed. ASHRAE states that where mechanical ventilating systems are designed to produce positive pressure in an enclosure, if the specified rate at which air is supplied exceeds the calculated infiltration rate, it is common practice to use the greater value in calculating heating load.

Therefore, it may be concluded that air leaks into a collector will displace the natural infiltration which would otherwise occur during collector operation. Thus, during collector operation, the natural infiltration component of the heating load should be reduced or eliminated depending on the leak rate. The relative magnitude of the infiltration-collector leakage interaction may be illustrated with an example. Consider a 1200 $ft.^2$ house containing 9600 $ft.^3$ of air. If natural infiltration is equal to one half air change per hour the infiltration flow rate will be 80 cfm. If the house were equipped with 400 $ft.^2$ of solar collectors with a flow rate of 800

n, a 10% leak rate would result in a forced ventilation rate of 80 cfm. Following ASHRAE common practice performance calculations would be based on 80 cfm and not 160 cfm. Therefore, a leak of 10% would not significantly effect the performance of the system. Close [1] points out that in this case where leakage does replace natural infiltration, collector leaks will result in better performance over collectors which don't leak. Of course, in the above example leaks greater than 10% would be detrimental.

In terms of thermal comfort bringing in outside air through the heating system where it is heated before reaching the occupants is more comfortable than letting it come through as cold drafts through cracks.

Ignoring the last three paragraphs, this study will assume a worst case condition, that the solar system leaks do not displace natural infiltration. Energy loss due to leaks will be considered an energy loss on the collector.

The amount of useful energy collected with leaks into the system is dependent on the leak location.

Leak Upstream of Collector. For a leak into the collector supply duct, the standard collector performance equation may be used with a reduced collector inlet temperature (T_i*):

$$T_i^* = (1 - \%)\ T_i - \%\ T_a \qquad [4]$$

The temperature rise across the collector may be calculated using the collector performance equation:

$$GC_p\ (T_e - T_i^*) = F_R(\tau\alpha)\ I_T - F_R U_L\ (T_i^* - T_a) \qquad [5]$$

Combining equations [4] and [5] and solving for T_e:

$$T_e = F_R[(\tau\alpha)I_T - U_L(1-\%)(T_i-T_a)]/GC_p + (1-\%)\ T_i + \%\ T_a \qquad [6]$$

Substituting into equation [3]:

$$q_u = F_R\ [(\tau\alpha)I_T - U_L\ (1-\%)\ (T_i-T_a)] - \%\ GC_p\ (T_R-T_a) \qquad [7]$$

The resulting expression for useful energy is exactly what one would expect. The useful energy collected (q_u) is the energy gained in the collector less the energy required to heat the leakage air from ambient to room temperature. Heat gain in the collector is increased over the no-leak case due to lower inlet temperatures, but the losses due to the required heating of outside air are greater.

Again, it should be noted that natural infiltration losses were not reduced for this analysis as recommended by ASHRAE standard practice. If
HRAE standard practice were followed and leakage
s greater than the calculated infiltration rate, the natural infiltration loss would be subtracted from the heating load or alternatively added to the useful energy collected as expressed in equation [7] resulting in increased performance. If the leak rate were less than the infiltration rate and standard practices followed solar system performance would actually increase over the no-leak case.

Leak Downstream of Collector. For a leak in the collector return duct, the collector performance equation may be used with a modified collector heat removal factor (F_R*). It is assumed that the plate efficiency factor (F') changes little with flow rate in the range of leakages expected. F_R* is the heat removal factor for a collector whose flow rate has been reduced by the percent leak rate (%).

$$\frac{F_R^*}{F_R} = \frac{GC_p(1-\%)}{F_R U_L}\left\{1-\left[1-\frac{F_R U_L}{GC_p}\right]\frac{1}{1-\%}\right\} \qquad [8]$$

Heat gain in the collector is then:

$$(1-\%)\ GC_p\ (T_e^*-T_i) = F_R^*\ (\tau\alpha)\ I_T - F_R^*\ U_L\ (T_i-T_a) \qquad [9]$$

where T_e* is the fluid temperature exiting the collector upstream of the leak. The fluid temperature downstream of the leak (T_e) is:

$$T_e = (1-\%)\ T_e^* + \%\ T_a \qquad [10]$$

Solving [9] and [10] for (T_e) and substituting into [3] yields:

$$q_u = F_R^*\ [(\tau\alpha)\ I_T - U_L\ (T_i-T_a)] - \%\ GC_p\ (T_r-T_a) \qquad [11]$$

Again, the resulting expression is what one would expect. Heat gain in the collector is reduced due to a reduced flow rate through the collector and additional energy is required to heat ambient air.

A similar analysis may be applied to a system with leakage divided between inlet and exit ducts resulting in:

$$q_u = F_R^*\ [(\tau\alpha)\ I_T - U_L\ (1-\%_i)\ (T_i-T_a)] - (\%_i+\%_e)\ GC_p\ (T_r-T_a) \qquad [12]$$

where, F_R* is based on $\%_e$.
$\%_i$ is leak fraction at inlet.
$\%_e$ is leak fraction at exit.

If ASHRAE standard practices were followed, q_u in [11] and [12] would be increased by the natural infiltration energy loss.

Furthermore, equation [7] with 100% leakage represents a make-up air heater or a collector with an ambient air source. The resulting performance is:

$$q_u = F_R\ (\tau\alpha)\ I_T - GC_p\ (T_r - T_a)$$

For a make-up air heater application, outside air heating would probably be considered a legitimate load which would result in:

$$q_u = F_R\ (\tau\alpha)\ I_T$$

Applicability to FCHART

Up to this point only the instantaneous performance of the system has been considered. What is of interest to the solar community is the effect of air leakage on long term solar system performance. There appears to be movement towards imposing maximum allowable leakage requirements on solar air-heating systems which would have the effect of stifling the solar industry and possibly reducing thermal performance of existing systems. A low-cost leaky collector may be more cost effective than an expensive leak-tight collector. Clearly leakage allowable in a given design should be an economic trade-off between performance and cost. Regulatory agencies should concern themselves with making sure system performance claims are correct and should stay out of dictating system performance.

FCHART [7] is a solar thermal performance prediction method derived from year long hourly simulations of solar heating systems by a large scale digital computer. Monthly solar fractions are calculated from two parameters:

$$X = F_R U_L\ (F_R'/F_R)\ (T_{ref} - T_a)\ t\ A/L$$

$$Y = F_R\ (\tau\alpha)_n\ (F_R'/F_R)\ (\overline{\tau\alpha})/(\tau\alpha)_n\ \overline{H}_T\ A/L$$

X is related to the ratio of a reference collector energy loss to the heating load. Y is related to the ratio of the solar energy absorbed by the collector to the heating load.

If the effects of collector leakage on instantaneous system performance can be expressed as altered collector performance parameters, the same equations can probably be used to alter the FCHART method simply by putting in altered collector parameters.

A set of collector parameters for FCHART modification has been derived based on the preceding theoretical analysis of the effect of air leaks on instantaneous system performance. To derive these parameters it was assumed that the collector inlet duct fluid temperature is equal to the heated space temperature which is the case most of the time in air based space heating systems. An example of the procedure applied to a standard Solaron system has been done with a 15 percent leak rate assumed. The factors are based on a worse case condition with no credit given for reduced natural infiltration. The suggested parameters and the resulting parameter numbers for the example and the resulting solar fractions are included below:

No Leaks

intercept $F_R\ (\tau\alpha) = .52$

slope $F_R U_L = .85$

solar fraction = 74.2%

Leaks Into Collector Inlet Duct

intercept $F_R\ (\tau\alpha) = .52$

slope $[F_R U_L\ (1-\%) + \%\ GC_p] = 1.05$

solar fraction = 70.0%

Leaks Into Collector Exit Duct

intercept $F_R^*\ (\tau\alpha) = .50$

slope $F_R^*\ U_L + \%\ GC_p = 1.14$

$$\frac{F_R^*}{F_R} = \frac{(1-\%)\ GC_p}{F_R U_L}\left\{1-\left[1-\frac{F_R U_L}{GC_p}\right]\frac{1}{1-\%}\right\}$$

solar fraction = 65.5%

Leaks Equally Divided Between Inlet and Exit Ducts

intercept $F_R^*\ (\tau\alpha) = .51$

slope $F_R^* U_L\ (1-\%/2) + \%\ GC_p = 1.10$

F_R^* based on %/2

solar fraction = 67.7%

Ambient Air Heater (No Collector Inlet Duct)

intercept $F_R\ (\tau\alpha) = .52$

slope $GC_p = 2.16$

solar fraction = 53.8%

FCHART Parameters

Code	Variable Description	Value	Units
1	Air SH+WH=1, Liq SH+WH=2, Air or Liq WH only=3.	1.00	
2	(Flow Rate/Col Area)(Spec Heat)?	2.16	Btu/H-F-F2
3	If 2, what is (epsilon)(Cmin) /(Ua)?	2.00	
4	Collector area	.500.00	FT2
5	FR-Tau-Alpha Product (Normal Incidence)	0.52	
6	FR-UL Product.	0.85	Btu/H-F-F2
7	Incidence Angle Modifier (-2.0 for SLRN 3000)	-2.00	
8	Number of Transparent Covers	1.00	
9	Collector Slope.	53.00	Degrees
10	Azimuth Angle (E.G. South=0, West=90)	0.00	Degree
11	Coll/Stor Heat Exchande Effectiveness	0.70	

Storage Capacity	15.00	Btu/F-FT2
13 Effective Building UA.	14400.00	Btu/F-Day
14 Constant Daily Bldg Heat Generation	0.00	Btu/Day
15 Room Set Temperature	73.00	F
16 Hot Water Usage.	0.00	Gal/Day
17 Water Set Temp (To Vary By Month, Input Neg.#)	.140.00	F
18 Water Main Temp (To Vary By Month, Input Neg.#)	51.80	F
19 City Call Number	40.00	
20 Print Out: Detailed=1, Summary=2 .	1.00	
21 Economic Analysis? Yes=1, No=2 . .	2.00	

The example illustrates the relatively small effect leaks have on a typical system performance even under worst case conditions. With a 15% leak equally divided between inlet and exit duct, solar fraction is reduced from 74% to 68%. If a system were configured such that ambient air were the sole source of inlet air to the collector, solar fraction would drop from 74% to 54%. Use of FCHART in this way as a means for determining solar fraction has not yet been verified.

To verify this hypothesis, TRNSYS must be run with collector leakage properly modeled and the results compared with FCHART runs with the altered collector parameters. Such a study is currently being considered at NBS as part of their investigation into leakage effects.

Honeywell did some simulations on one of their solar air-heating systems [2] in order to study the effects of collector leakage. The study was done on a house with a large passive gain in addition to the active system and only limited documentation was produced. A 50% solar fraction was predicted for a no-leak system. At collector leakage rates of 10, 15 and 20% the solar fractions were 42, 37 and 32% respectively. Applying FCHART with altered collector parameters, solar fractions at 0, 10, 15 and 20% collector leakage were 49, 44, 42 and 40%. These results are encouraging, but inconclusive. The role of the passive gains in the results were not clear.

Conclusions

Obviously much more analysis is required before the equations presented here can be used to predict the effects of air leaks on system performance. A large scale computer effort is required to validate the suggested modifications to FCHART.

The role of air leaks into a solar system is to decrease the efficiency of the system over the case where no leaks occur (assuming leaks do not displace natural infiltration). This is the worst case condition. A sample calculation shows performance degradation with leakage is not great (6.5% drop in solar fraction for a 15% leak in the example) even with assuming worst case conditions. The effect on performance is dependant on the location of the leak.

On the other hand, if ASHRAE standard practices are followed for load calculation and solar system leakage does not exceed the natural infiltration flow rate, effects of leakage are small and may be positive or negative depending on the location of the leak.

The effect of leakage in the collector itself was not considered since collector leaks would result in reduced collector performance during performance testing and thus would already be inherent in the collector performance parameters.

The results of this study are very encouraging. Depending on the effect solar system air leakage has on natural infiltration and the location of the leaks, the effects on system thermal performance range from a relatively small decrease in performance in the worst case and a slight increase in performance in the best case. It should be fairly easy to develop a calculation procedure to predict the effect of leakage.

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NOTES

THE PERFORMANCE OF SOLAR ENERGY SYSTEMS IN THE RESIDENTIAL SOLAR DEMONSTRATION PROGRAM

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ABSTRACT

This paper analyzes performance information derived from the instrumentation of twenty-three solar projects built in the Residential Solar Demonstration Program.

Four exercises were performed on data supplied by instrumentation.

1. The apparent solar fraction was calculated by dividing the measured solar contribution by the sum of the measured solar contribution and the measured auxiliary thermal output energy.[A]

2. The measured electric or fossil input was compared with the home's predicted electric or fossil input assuming solar performs as expected.[B]

3. Measured losses from solar energy storage and transport subsystems were tabulated and the effects of adding measured losses to the measured solar contributions were calculated.

4. The measured electric or fossil input was compared with the home's predicted conventional energy use assuming no solar contribution.[B]

The following results were obtained:

1. Solar Fraction - Table 1

   Solar fractions for many active space heating systems were respectable, with eight of sixteen projects showing fractions around 40%. These were below F-CHART predictions for most of the projects, however. Domestic hot water systems had fractions ranging from 24% to 91% with four of five projects near or exceeding predictions. Two passive space heating projects had high fractions of 75% and 81%.

2. Electric or Fossil Energy Input - Table 2

   Nine of sixteen active space heating systems and three of five domestic hot water systems used amounts of electric or fossil input equal to or less than predicted levels.

3. Uncontrolled Losses - Table 3

   In eight active space heating projects heat losses from storage and transport were more than 50% of the measured solar contribution. Solar fractions in most of these projects could improve significantly if heat losses were reduced.

4. Comparisons with Equivalent Non-Solar Houses - Table 4

   Nearly all projects used less electric or fossil input for space and domestic water heating than predictions for equivalent houses without solar.

INTRODUCTION

The U.S. Department of Housing and Urban Development is responsible for managing the residential demonstration portion of the National Solar Heating and Cooling of Buildings Program. Since the program's inception HUD has funded the construction of over 600 single or multi-family projects with space and/or domestic water heating provided by solar energy. About one third of these have been completed. Most of these have been sold or rented and the solar homes and systems are providing a variety of information. Some projects are supplying instrumented results through the National Solar Data Network. In addition, construction, institutional and market acceptance information is being collected from a subset of projects by HUD contractors. This paper focuses on results obtained from analysis of data from instrumented projects in the first three grant cycles.

Twenty-three projects, representing 44% of the instrumented projects and 7% of all projects awarded grants in the first three cycles, are occupied and are supplying data within reasonable parameters. It should be emphasized that analysis of this performance information is constrained by the small size of the sample and the fact that most of these projects represent early program efforts - most were built in cycle one or cycle two. The following numbers and types of projects are included in this analysis:

System Application	# of Projects
Space and Domestic Water Heating - Liquid	11
Space and Domestic Water Heating - Air	5
Domestic Hot Water	5
Passive Space Heating	2
TOTAL	23

RESULTS

Solar Fraction

Apparent and predicted solar fractions are shown for the four types of projects included in this sample in Table One. For most active space heating systems, apparent solar fractions were below those predicted. Fractions were generally higher for liquid than for air systems. For liquid systems, solar fractions ranged from a high of 52% to a low of 12% with an average of 33%. For air systems, the range was between 45% and 4% with a 17% average.

Four of the five domestic hot water systems had apparent solar fractions near or above predictions, although the range was wide, varying from 24% to 91%. Both passive systems had high solar fractions at 75% and 81%.

Lower solar fractions in active space heating systems were usually correlated with significant design or installation problems. Many poorly performing air space heating systems reported problems with leaky ducts and dampers. One air system with a very low measured solar fraction reported multiple problems at collector units, dampers, blowers and storage. Instrumentation also shows that this system is losing a significant amount of heat from collectors, storage and transport.

Liquid space heating projects with performance significantly below predictions have reported problems with collector flow rates, valve and joint leaks, pump cycling and heat loss from badly insulated or uninsulated piping and storage. (1)

One multi-family hot water project with mediocre performance lost heat in long pipe runs from central storage to individual storage tanks and at individual storage. Heat loss here was furthered by irregular use patterns, however.

Although the solar fractions of both passive projects were very high, occupants have allowed temperatures to vary more than normal. Further, at least one has had minor problems with heat distribution. At times the rooms at the north side of the house were noticeably cooler than those adjacent to collection areas. In addition there has been some overheating because of inadequate venting. An analysis of passive demonstration projects done by the International Business Machines Corporation has suggested that control devices need further attention to assist south-north energy transfer in passive design. (2) HUD reached much the same conclusion in its analysis of many of the submissions to its recent Passive Design Competition.(3)

AUXILIARY ELECTRIC OR FOSSIL ENERGY INPUT[B]

Ratios of measured to predicted electric or fossil input with solar are presented in Table 2. Results should be interpreted in the context of the actual solar contribution. A few projects report substantial solar contributions but not quite at predicted levels. Since predicted auxiliary input with solar is low for most of these projects a small deviation from predicted performance can result in the use of twice as much auxiliary as predicted[C] even though consumption is still low. Project C-5 is a good example of this phenomenon. The project is performing well as shown by the measured solar contribution of 119.66 million BTU (MMBTU) for the period. Even though it is using about twice the predicted electrical energy, the MMBTU of auxiliary consumed are still low and the apparent solar fraction is 86%.

Table 1 - APPARENT SOLAR FRACTION

SYSTEM TYPE	PROJECT	PERIOD	MEASURED SOLAR CONTRIBUTION (MMBTU)	MEASURED SOLAR CONTRIBUTION + MEASURED AUXILIARY OUTPUT (MMBTU)	APPARENT SOLAR FRACTION	PREDICTED SOLAR FRACTION
A.- Heating and Liquid Collectors	A-1/22041 Denver, CO	Apr,1978-Mar,1979	20.41	51.38	40%	84%
	A-2/23101 Gainesville, FL	June-Oct, 1978, Dec, 1978	11.05	21.29	52%	93%
	A-3/26910 Medway, MA	Mar-Apr, 1979	7.75	19.39	40%	47%

SYSTEM TYPE	PROJECT	PERIOD	MEASURED SOLAR CONTRIBUTION(MMBTU)	MEASURED SOLAR CONTRIBUTION + MEASURED AUXILIARY OUTPUT (MMBTU)	APPARENT SOLAR FRACTION	PREDICTED SOLAR FRACTION
A. Heating and Water—Liquid Collectors	A-4/22072 Malta, NY	Dec,1978-Apr,1979	6.53	52.64	12%	58%
	A-5/23097 Escondido, CA	Aug,1978-Apr,1979	8.09	16.58	49%	64%
	A-6/22012 Big Fork, MT	Feb,1979-Apr,1979	13.60	31.36	43%	52%
	A-7/21510 Virginia Beach VA	Feb,1978 May-June, 1978, Sept-Nov,1978	2.73	8.45	32%	78%
	A-8/22080 Huntsville, AL	May,1978 July-Nov 1978	3.96	9.74	41%	71%
	A-9/22053 Albequerque, NM	May,1978 Sept-Dec 1978,Mar-Apr,1979	12.64	34.84	36%	75%
	A-10/22063 Columbia, MD	Oct-Dec, 1978,Mar-Apr,1979	9.97	26.84	37%	53%
	A-11/22063 Columbia, MD	Oct-Nov 1978,Feb-Apr,1979	5.52	33.59	16%	61%
		TOTAL	102.25	306.1	33%	
B. Heating & Water—Air Collectors	B-1/23126 New Haven, CT	Dec,1978-Feb,1979	1.76	31.66	6%	37%
	B-2/22069 Greenville, SC	Jan-Apr, 1978,Oct-Dec,1978	25.71	56.64	45%	82%
	B-3/22023 Kirkland, WA	Nov,1978-Apr,1979	5.95	35.85	17%	8%
	B-4/22007 Greenwood, IN	Dec,1977-Feb,1979	10.24	116.35	9%	60%
	B-5/22087 Chester, NY	Oct-Dec, 1978	.95	24.61	4%	0%
		TOTAL	44.61	265.11	17%	
C. Domestic Hot Water	C-1/22030 San Diego, CA	Mar,1978-Apr,1979	113.31	474.47	24%	51%
	C-2/26909 Kaneche, HI	Feb-Nov, 1978,Jan-Apr,1979	8.78	9.65	91%	75%

SYSTEM TYPE	PROJECT	PERIOD	MEASURED SOLAR CONTRIBUTION(MMBTU)	MEASURED SOLAR CONTRIBUTION + MEASURED AUXILIARY OUTPUT (MMBTU)	APPARENT SOLAR FRACTION	PREDICTED SOLAR FRACTION
C. Domestic Hot Water	C-3/26910 Medway, MA	Nov,1978-Mar,1979	3.80	15.39	25%	17%
	C-4/26910 Medway, MA	Nov,1978-Apr,1979	3.33	9.45	35%	27%
	C-5/22018 Honolulu, HI	Apr-Dec, 1978,Feb-Apr,1979	119.69	138.84	86%	95%
		TOTAL	248.91	647.80	38%	
D. Passive	D-1/23061 Longmont, CO	Nov,1978-Apr,1979	37.41	50.22	75%	N.A.
	D-2/23110 Prescott, AZ	Oct,1978-Apr,1979	58.44	71.95	81%	N.A.
		TOTAL	95.85	122.17	78%	

Table 2 MEASURED VS. PREDICTED AUXILIARY ENERGY INPUT[B]

SYSTEM TYPE	PROJECT	SOLAR CONTRIBUTION (MMBTU)	MEASURED AUXILIARY INPUT (MMBTU)	AUX FUEL TYPE	PREDICTED AUXILIARY INPUT WITH SOLAR(MMBTU)	RATIO OF MEASURED TO PREDICTED AUXILIARY INPUT WITH SOLAR
A. Heating and Water Liquid Collectors	A-1	20.41	50.07	Gas	21.24	2.4
	A-2	11.05	10.24	Elec	4.76	2.2
	A-3	7.75	17.99	Oil/Elec	15.31	1.2
	A-4	6.53	68.92	Gas/Elec/Wood	30.87	2.2
	A-5	0.09	13.38	Elec	26.11	.5
	A-6	13.6	2.05	Elec	18.24	.1
	A-7	2.73	2.32	Elec	3.32	.7
	A-8	3.96	2.59	Elec	3.29	.8
	A-9	12.64	14.10	Elec	6.33	2.2
	A-10	9.97	8.26	Elec	10.45	.8
	A-11	5.52	12.48	Elec	5.31	2.4
	TOTAL		202.40		145.23	1.4
B. Heating and Water-Air Collectors	B-1	1.76	29.90	Elec	70.36	.4
	B-2	25.71	51.14	Gas	11.91	4.3
	B-3	5.95	50.10	Gas	173.31	.3
	B-4	10.24	106.11	Elec	41.41	2.6
	B-5	.96	12.47	Elec	10.81	1.2
	TOTAL		249.72		317.62	.8
C. Domestic Hot Water	C-1	113.31	361.16	Elec	131.86	2.7
	C-2	8.74	.87	Elec	7.38	.1
	C-3	3.80	19.32	Gas	16.61	1.2
	C-4	3.33	10.20	Gas	12.46	.8
	C-5	119.69	21.33	Elec/Gas	9.20	2.3
	TOTAL		412.88		177.51	2.3
D. Passive Heating	No Predictions Available					

spite of the fact that many active space heating systems had apparent solar fractions that were lower than predicted, six of eleven liquid and three of five air space heating systems used electric or fossil energy equal to or less than what was predicted, assuming solar performance at F-Chart 2 levels. These projects are actually saving more conventional fuel than predicted. HUD believes that measured solar contributions are being supplemented in many of these projects by the effects of uncontrolled heat losses as well as by passive design features which help satisfy the overall heating load. Passive features supplementing active solar heating systems became more common as the cycles progressed. In addition some projects have supplemented both solar and conventional energy sources by using other renewable energy sources. For example, the conventional contributions have been relatively low in project A-6 because of the contribution of a hydro heater fireplace which is integrated into the overall energy supply design.

However, part of the difference between measured and predicted auxiliary usage may also be due to the different conditions and methodologies involved in measuring loads on the one hand and predicting them on the other. For example, actual weather conditions may differ considerably from the historical average weather data used in making predictions. Calculation procedures are discussed by Sparkes in another paper.(4) The potential contribution of uncontrolled losses is discussed below.

Effects of Uncontrolled Losses

Table 3, shows results for space heating projects where losses from storage and transport have been measured for months between September and April. Losses from collectors to storage have been excluded because it is assumed that most of these were to the ambient air at the collectors. Besides storage and transport losses, the table shows the solar contribution in MMBTUs, the apparent solar fraction and the ratio of actual to predicted electric/fossil energy use. Of the nine projects where storage and transport losses could be recorded, eight measured losses over 50% of the measured solar contribution. Three of the eight lost more solar heat from storage and transport than was delivered to the load through the regular distribution system. If most of the uncontrolled heat loss in storage and transport went to conditioned space, the solar fractions would improve noticeably for most of the projects. The solar fraction obtained by adding the measured storage and transport losses to the apparent solar fraction is presented as the potential solar fraction in Table 3.

The table also suggests that the contribution of uncontrolled losses varies widely from project to project. Project B-3, for example, has an apparent solar fraction of only 17% but it also is using only about 30% of predicted gas auxiliary The project also reports relatively large losses from storage and transport. The difference between actual and predicted auxiliary use here is too large to be explained by the effects of losses alone. However, it is reasonable to assume that at least some of the heat lost from

Table 3 - PROFILE OF MEASURED LOSSES

PROJECT	PERIOD*	MEASURED SOLAR CONTRIBUTION (MMBTUs)	MEASURED LOSSES (MMBTUs)	RATIO OF MEASURED TO PREDICTED ELECTRIC OR FOSSIL INPUT WITH SOLAR	APPARENT SOLAR FRACTION	POTENTIAL SOLAR FRACTION
A-1	Apr,1978, Sept,1978-Mar,1979	16.07	29.20	.7	36%	61%
A-3	Mar-Apr,1979	7.75	5.45	1.2	40%	53%
A-4	Dec,1978-Apr,1979	6.53	4.03	2.3	12%	19%
A-5	Sept,1978-Apr,1979	7.63	4.44	.5	49%	59%
A-8	Sept-Nov,1978	2.26	5.69	.5	43%	73%
A-10	Oct-Dec,1978 Mar-Apr,1979	9.97	2.33	.8	37%	42%
A-11	Oct-Nov,1978,Feb-Apr,1979	5.52	5.74	2.4	16%	27%
B-3	Nov,1978-Apr,1979	5.95	8.36	.3	17%	32%
B-4	Dec,1977-Apr,1978, Sept, 1978-Feb, 1979	8.66	5.10**	2.5	8%	12%

* Figures here are for the period listed only
** Losses are probably much more extensive for this project but there has been difficulty measuring them.

storage and transport is helping to satisfy the heating load. If all losses are assumed to go to the conditioned space, the apparent solar fraction changes from 17% to 32%. The actual solar fraction is probably somewhere between these numbers. Project B-4, however, has a measured solar fraction of only 8%. And this project is using over twice the amount of predicted electric energy. Losses in this project have apparently not been as helpful. The project has reported heavy losses from collectors to the outside air. Storage is also buried underground with the consequence that most storage losses do not go to heated areas.

Although uncontrolled losses are apparently helping to satisfy heat loads in some projects, they also add to cooling loads in summer unless there is a hot water only mode that by-passes storage. In project A-1, for example substantial heat losses have added significantly to utility costs for cooling. In surveys, several purchasers of grant houses have complained about summer overheating and high electric bills caused by the need for additional air conditioning in projects with extensive losses.(5)

Comparisons with Equivalent "Non-Solar" Homes

Nearly all of the houses in the sample are using less conventional energy for space heating and domestic hot water than they would without solar. In Table 4, the electric or fossil energy input used is compared to the predicted utility use of the house without solar. Of the active space heating systems three of five air systems and nine of ten liquid systems were net space heating energy savers. Since this comparison includes space and domestic water heating energy use only, additional cooling loads caused by solar heat released in summer are not computed. As noted above, however, these can be significant in projects with heavy heat loss and no summer storage by-pass. Projects which lose extensive heat to the conditioned space during summer months would be expected to save noticeably less energy on a net basis than what is indicated here if auxiliary for cooling were included in this calculation.

Three of five domestic hot water systems were net energy savers. It should be noted that predicted loads for one of the two that used more proved to be inaccurate because of extremely unpredictable usage patterns.

Consistency of Results

This general analysis has suggested that while measured solar fractions of many active space heating systems were below design specifications, uncontrolled losses in some projects helped to satisfy heating loads and thus save conventional energy. However, it is still not clear exactly how much this is occurring at individual sites.

Although we have suggested possible ways to help determine this we are also aware of the limitations involved in dealing with a small sample which is supplying data whose reliability is still being verified.

One way to assess the overall reliability of the results for each project is to compare different parameters to see if they present a consistent picture of system performance. If results are not consistent, some additional digging into the details of the project itself is necessary for an understanding of system performance. Numbers supplied by instrumentation or calculation may be erroneous or some design, installation or operational anomaly may be influencing the results.

Table 5, compares the Apparent Solar Fraction, the Measured/Predicted Electric/Fossil Usage with Solar and Measured Electric/Fossil Usage/Predicted Electric/Foss Use Without Solar for all projects in the sample for which data are available. If we look at these three results on a project by project basis we can separate consistent from inconsistent project descriptions.

For project A-3, for instance, the three performance measures appear to be relatively consistent. The apparent solar fraction is 40% - respectable, although below design specifications. The house is also using about what was predicted for conventional energy consumption with solar and about 60% of the predicted consumption of an equivalent non-solar house.

In project B-1, on other hand, there is an apparent solar fraction of only 5%. However, results also indicate that the home is consuming only 37% of its predicted utility consumption with solar and 22% of the conventional energy that would be consumed by an equivalent home without solar. Clearly this project needs additional study to explain or reformulate performance results.

le 4 - MEASURED AUXILIARY COMPARED TO PREDICTED AUXILIARY USE WITHOUT SOLAR

SYSTEM TYPE	PROJECT	MEASURED ELECTRIC OR FOSSIL USE (MMBTU)	PREDICTED ELECTRIC OR FOSSIL USE WITHOUT SOLAR (MMBTU)	RATIO OF MEASURED ELECTRIC OR FOSSIL INPUT TO PREDICTED INPUT WITHOUT SOLAR
Heating & Water-Liquid Collectors	A-1	50.07	133.74	.4
	A-2	10.24	27.83	.4
	A-3	17.99	28.90	.6
	A-4	68.92	74.02	.9
	A-5	13.38	71.65	.2
	A-6	2.05	37.87	.1
	A-7	2.32	17.60	.1
	A-8	2.59	14.17	.2
	A-9	14.10	27.04	.5
	A-10	8.26	22.42	.4
	A-11	12.48	13.11	1.0
	TOTAL	202.4	468.06	.4
Heating & Water-Air Collectors	B-1	29.90	112.55	.3
	B-2	51.28	67.13	.8
	B-3	50.10	187.77	.3
	B-4	106.11	104.19	1.0
	B-5	12.47	10.81	1.2
	TOTAL	249.86	482.45	.5
Domestic Hot Water	C-1	361.16	266.48	1.4
	C-2	.87	29.68	.03
	C-3	19.32	20.07	1.0
	C-4	10.20	17.14	.6
	C-5	21.33	189.21	.1
	TOTAL	412.88	522.58	.8
Passive Heating		No Predictions Available		

Table 5 - SUMMARY

SYSTEM TYPE	PROJECT	APPARENT SOLAR FRACTION	RATIOS OF MEASURED TO PREDICTED AUXILIARY ELECTRIC OR FOSSIL INPUT WITH SOLAR	WITHOUT SOLAR
Heating & Water-Liquid Collectors	A-1	40%	2.4	.4
	A-2	52%	2.2	.4
	A-3	40%	1.2	.6
	A-4	12%	2.3	.9
	A-5	49%	.5	.2
	A-6	43%	.1	.1
	A-7	32%	.7	.1
	A-8	41%	.8	.2
	A-9	36%	2.2	.5
	A-10	37%	.8	.4
	A-11	16%	2.4	1.0
Heating & Water-Air Collectors	B-1	6%	.4	.3
	B-2	45%	4.3	.8
	B-3	17%	.3	.3
	B-4	9%	2.6	1.0
	B-5	4%	1.2	1.2

SYSTEM TYPE	PROJECT	APPARENT SOLAR FRACTION	MEASURED ELECTRIC/FOSSIL USE/PREDICTED ELECTRIC/FOSSIL USE WITH SOLAR	MEASURED ELECTRIC/FOSSIL USE/PREDICTED AUXILIARY USE WITHOUT SOLAR
Domestic Hot Water	C-1	24%	2.7	1.4
	C-2	91%	.1	.03
	C-3	25%	1.2	1.0
	C-4	35%	.8	.6
	C-5	86%	2.3	.1
Passive Space Heating	D-1	75%	N/A	N/A
	D-2	81%	N/A	N/A

RESULTS OF OTHER PERFORMANCE MEASURES

Additional information on residential solar system performance and operations is being supplied by sources other than instrumentation. In general these data are consistent with the conclusions discussed above. They also suggest that instrumentation results may improve as later projects begin to report data.

Utility Bills

For projects where both instrumentation results and utility bills are available, results have been relatively consistent. Utility bill analyses for several residential projects are presented in another paper.(6)

Reports on Problems

Reports on serious problems occuring during system operation are forwarded to HUD and NBS. Serious problems in all hardware areas seem to be declining, although many of the more recent projects have not been operating long. While collectors are still cited most frequently as the focus of solar operational problems, the incidence of these problems appears to be going down between cycle two and three as manufacturers and builders gain more field experience. As noted above, however, problems in non-collector areas such as insulation and controls can have significant effects on solar system performance.

Purchaser Reactions

In addition, purchaser reactions seem to be more positive for cycle three than for cycle two. The most recent survey of solar purchasers indicates that for projects built in cycles one through three about 2/3 of those surveyed were satisfied with their solar energy systems. However, 92% of those cycle three purchasers surveyed were satisfied. Dissatisfied purchasers generally cited higher than expected utility costs or uncomfortable temperatures as the reasons.

Many of the complaints expressed by the dissatisfied consumers were also consistent with instrumentation and problem reports. For example, a number of dissatisfied purchasers have complained about heat losses from the system or malfunctioning control devices.(5)

The previous two sources of information suggest that the measured performance of solar systems may improve as some of the more recently completed projects begin to report results, but this of course is speculative at this time.

CONCLUSIONS

With only a few exceptions the projects reported here are performing neither exceptionally well nor exceptionally poorly. While both passive systems in the sample performed well, most of the active space and/or domestic water heating projects could be performing better. The key factors affecting the performance of these systems appear to be quality of equipment, soundness of design, and most importantly, competence of installation.

At last year's Operational Results Conference HUD suggested that uncontrolled heat losses from storage and transport were substantially degrading performance in active space heating systems.(7) This still appears to be the case at least for this sample of projects.

In many active space heating systems uncontrolled heat losses are helping to meet heating loads. But this is an inefficient use of solar energy unless the house is designed to take advantage of these losses in the winter and not be adversely affected by them during non-heating periods. Otherwise, to perform well, solar energy systems must be designed and installed to reduce uncontrolled heat losses as much as possible. In most cases this means that installation quality must meet or, in the case of air systems, exceed current good HVAC practice.

HUD believes that the performance of solar houses currently being built can be improved based upon lessons learned in earlier projects. This is particularly true in the area of installation. As hardware and design techniques improve, installation quality will become even more critical to good system performance.

To help builders learn as much as they can from these experiences HUD is continuing to produce publications describing demonstration experiences and outlining good solar installation practice. These include Installation Guidelines for Solar Domestic Hot Water Systems in One and Two Family Dwellings, a new Installation HUD Solar Status Report which outlines good practice for installing air and liquid active systems and the First Passive Solar Home Awards which includes a chapter on design and installation mistakes which can hurt performance in passive systems. Additionally updated versions of Building the Solar Home and Selling the Solar Home are currently being prepared. All of these publications are or will be available from the National Solar Information Center.

ACKNOWLEDGMENTS

Anne Peters of the National Solar Information Center did the bulk of the data collection for the paper using instrumented reports from the National Solar Data Network as her primary sources. In addition, a number of individuals have made important contributions to the analysis. H. Robert Sparkes and James McCall of Dubin Bloome Associates provided F Chart results for all active systems in the sample. Mike Nash and Gordon Cunningham of International Business Machines, Inc., Federal Systems Division, gave substantial assistance in interpreting instrumentation results. Valuable comments on earlier drafts of this paper were provided by Mark McGrath of the Boeing Aerospace Company and Frank Weinstein, Ray Yaeger and Lance Hoch of the National Solar Information Center. However, the authors are responsible for the conclusions presented here.

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[7] William Freeborne, "Performance of HUD Residential Solar Systems," Proceedings of Solar Heating and Cooling Operational Results Conference, Colorado Springs, 1978, pp. 1-5.

APPENDIX [NOTES]

[A] This measurement was used instead of the measured load recorded by instrumentation because measurements for solar and auxiliary subsystems were deemed to be more consistent and reliable. Additionally, the auxiliary input energy measure was inappropriate in heat pump projects therefore was estimated by adjusting auxiliary electric fuel consumption by the following formula:
(Electric MMBTU consumed x 1.2708) + .0255 x (monthly average ambient temperature in $^{\delta}$F). This formula is used in HUD's Residential Solar Viability Program (RSVP), a computer package designed to project solar technical and economic performance.

[B] Note that auxiliary electric or fossil input is a measure of the BTU content of the fuel consumed by the heating and hot water subsystems as opposed to the auxiliary thermal output figures used in other parts of this paper to measure the BTUs delivered from the subsystems to the loads. Predicted auxiliary input with solar was calculated by dividing the predicted load minus the predicted solar contribution by the relevant burner efficiency or coefficient of performance (COP). Predictions for utility input without solar were calculated by dividing the predicted load by the same number. Burner efficiencies used were 1.0 for electricity and 0.6 for oil and natural gas. The COP of heat pumps was calculated using the RSVP formula discussed in Appendix A. Predicted loads and solar contributions were computed by Dubin Bloome Associates using F-Chart 2 method.

[C] It should also be noted that both load and solar contribution predictions could be too high.

NOTES

Luncheon Speaker

THE SOLAR BANK AND OTHER SOLAR LEGISLATION IN THE 96TH CONGRESS

Terry Adlhock, Legislative Assistant to Senator Robert Morgan (D-N.C.) U.S. Senate, Washington, D.C.

Thank you for having me here today. You're probably wondering what a legislative assistant to a junior U.S. Senator from North Carolina who is not on the Senate Energy Committee, not on the Senate Commerce, Science and Transportation Committee is doing here in Colorado. I'm reminded, in answer to my own question, of what another North Carolina Senator has been doing lately along these same lines: "advertising". The differences being, however, Sam Ervin is on national television, he's being paid handsomely, and he's selling a proven and widely accepted product. If solar applications were the same as American Express, could you even imagine where we'd be if everyone wasn't leaving home without them? Or, for that matter, if everybody simply had them at home to leave?

THE SOLAR BANK INTRODUCED

Far be it from American Express, however, My "advertising" is for some solar legislation - the sponsors of which feel is going to do a great deal to get more solar applications into homes and businesses in this country - the Solar Energy Bank. This legislation originated in the Senate this year with Senator Morgan, my boss, and in the House of Representatives with Representative Steve Neal, also of North Carolina. As I'm sure you all are aware, President Carter, in his June 20th Solar Message to the Congress, enthusiastically endorsed the Solar Bank. The respective authorizing committees of each House reported out favorably and unanimously in the legislation. The Federal Budget for Fiscal 1980 has provided ample funding for the Bank's first year of operations.

However, when Dan Ward talked with me in mid-August about my - and I quote from his follow-up letter - "speaking November 30th at the Second (Annual Solar Heating and Cooling Systems Operational Results) Conference on the status of solar legislation passed and pending", I realize now that we then completely disregarded the one sure and fast law in Washington: the law of uncertainty. It doesn't take a physicist, a professor, a commission or even a consultant to determine that little, if anything, is certain - or even predictable - in our nation's Capital, particularly in Congress.

I would be less than accurate to tell you that the Solar Energy Bank is a certainty this year and will definitely be in operation in the foreseeable future. I can tell you in all sincerity that, in mid-August, I fully and reasonably, I think, expected this by now, however. The legislation was roduced simultaneously in identical form in both ses. It is straight-forward, uncomplex legislation which relies on financial incentive and the private marketplace, with minimal government involvement. We wrote the bill such that only one committee in each chamber of Congress would have to deal with the bill. We started the Bank off modestly but provided for real growth and significant funds in the second and subsequent years of operation. We designated existing government personnel for the Bank's manpower, so as to utilize people with a proven track record and also to avoid the charge of "just another Washington Agency to employ 'umpteen' more federal bureaucrats".

I talk about all this connected with the proposed Solar Bank in the conditional, if not the past tense, because of what has happened in Washington with the Bank since mid-August, almost none of which we "plain ol' country lawyers", as Sam Ervin likes to say, foresaw.

What we foresaw eight months ago when the Solar Bank was introduced was a (Solar Energy Development) Bank as a separate legal entity but under the umbrella of the federal Department of Housing and Urban Affairs (HUD). The Bank would be a corporation akin to Ginnie Mae, the Government National Mortgage Association, also under the HUD umbrella. The Solar Bank would subsidize the interest rate on loans for installation of solar energy systems by making a lump-sum payment to lenders to make up the difference between a below-market interest rate and the market rate. At the time, we were looking at roughly a 12-percent market rate, and our bill would have paid a subsidy of half that cost, or a flat six percent.

Either owners or builders of both residential and commercial structures would qualify for the subsidy, simply provided that they purchased and installed solar energy systems in such structures. 'Solar energy systems' was defined essentially as it is found in both Section 2(a) of the National Housing Act and in the National Energy Policy and Conservation Act:

> "Any addition, alternation or improvement to an existing or new structure which is designed to utilize wind energy or solar energy either of the active type based on mechanically-forced energy transfer or of the passive type based on convective, conductive or radiant energy transfer or some combination of these types to reduce the energy requirements of the structure from other energy sources, and which is in conformity with such criteria and standards as shall be prescribed by the Secretary of Housing in consultation with the Secretary of Energy."

Warranties would be required on solar devices financed by a subsidized loan. These warranties would come from either the manufacturer or installer, or

both. Loan repayment would be limited to not less than five, nor more than 30 years. Sixty percent of the amount of Solar Bank subsidy payments in any given year would have to go to residential loans, as opposed to loans to business places. Four hundred and fifty million dollars was to be provided over three years, starting this year, for the (Solar) Bank.

THE SOLAR BANK IN NOVEMBER

Where are we in late late November with this bank? In the inevitable give-and-take of the legislative process, there have been changes - necessary changes, I might emphasize - in order to see the proposal through. Yet, I don't believe we've ended up with a camel - by committee - when we started out intending to design a horse. As passed out of the Senate three weeks ago and now in conference with the House, we have a solar and Conservation Bank. The Solar Bank portion has been funded for four years, at over $700 million. A provision has been added whereby builders and developers are availed of interest subsidies to assist them in their short-term or interim construction borrowing, as long as solar energy systems are actually installed in their buildings. "Wood-burning appliances, solar-process heat, solar-electric devices, and earth shelters" have been added to the definition of "solar energy systems". Federal interest subsidies and federal tax credits both can be utilized under express language in this most recent version of the Solar Bank bill, but only to the extent of 50 percent of the overall cost of any eligible solar energy system. The interest rate subsidy will be flexible - up to 6 percentage points below the Federal Housing Administration (FHA) interest rate. This flexible interest rate is considered both necessary and desirable to ensure that different solar technologies are subsidized only to the degree necessary to stimulate their use by considering (1) prevailing market interest rates, (2) the availability of other government subsidies for solar energy, (3) the costs of non-renewable energy resources and systems, (4) the costs of solar energy systems, and (5) the amount of inducement necessary to ensure installation of solar energy systems. Unfortunately, we couldn't figure out a way simultaneously to ensure a Bank board of Solomons and Merlins to make the balance of these five considerations come out just right every time.

The 'give-and-take' precess extracted a few costs, however. Units of local government which own and operate low-income housing are now eligible for lump-sum interest-subsidy payments for the purchase and installation of solar energy systems. Persons with "adjusted" incomes exceeding $50,000 annually will not be eligible for Solar Bank assistance, except that this provision will not take effect until 1981 and must be measured in "real" dollars, whatever those are.

I should mention, also, among the 'costs' extracted in the legislative process to date connected with the Solar Bank has been the necessity to see it included in the massive Synthetic Fuels bill, S.932, which occupied the news headlines several weeks ago. In the unfathomable wisdom of the United States Senate, the $88-billion effort for syn-fuel plant development, the creation of a government Synthetic Fuels Corporation, and a Solar and Conservation Bank all go together. Accordingly, the Solar Bank proposal I have been discussing with you was included in the SynFuels bill, and it is in that vehicle the Bank is now before the House of Representatives. Efforts to keep the Solar Bank separate, and thereby simpler, we felt, were thwarted in the Senate essentially on the issue of committee jurisdiction.

THE BANK'S IMPACT

Thinking ever-optimistically - or, at least speaking that way - let me assume successful final passage and enactment of the Solar Bank this year. What has been estimated for its impact?

The Banking Committee, which held the hearings this year in the Senate on the Solar Bank bill, estimates the Bank should have minimal regulatory impact, because it relies heavily on existing regulations. New regulations would be necessary to coordinate Solar Bank activities with other federally-assisted loan programs, to define eligibility for subsidized solar loans, and to require warranties on solar energy devices financed with subsidized loans.

The solar industry would be impacted by these new regulations requiring one-year warranties. However, borrowers and lenders should benefit to the extent these warranties guarantee sound solar equipment and competent installation. I might point out that a recent General Accounting Office (GAO) study entitled "Commercializing Solar Heating: A National Strategy Needed" (July 20, 1979) found that warranties are lacking within the solar industry in many cases.

"For example, of the 56 solar manufacturers GAO contacted, only 7 (12 percent) would guarantee in any way the performance of their collectors." Also according to GAO, "at the present time, consumers have little assurance that any performance claims will be met and no adequate assurance of a system's reliability".

We Solar Bank backers believe these assurances are necessary to protect consumers and to ensure the widespread acceptance and development of solar technologies. Fortune magazine, in its September 24th article on solar energy, capsulated these concerns in discussion in the article about this very conference - on the performance of operating solar heating and cooling systems - last November: "The majority of the systems simply delivered a lot less energy than they had been expected to because of seemingly inconsequential errors." Dan Ward, our conference chairman both then and now, was quoted to say, "What we learned was - the importance of systems design - that you can take a system that's able to meet 50 percent of a building's heating load and reduce its contribution to zero by being sloppy about the details."

All the witnesses the Senate and House Banking Com-

mittees heard this year endorsed a Solar Bank, from Professor Maidique of Harvard Business School, who wrote the solar energy chapter of Energy Future, the Business School's cogent Energy Project Report, to Denis Hayes, then head of the Solar Lobby in Washington:

> "The residential sector currently consumes 22 percent of U.S. energy; an additional 16 percent is consumed in the commercial sector. Almost all of this energy is employed to heat space and hot water. The residential sector used almost 12 quads of energy last year for space and water heating. At least 6 quads were expended by the commercial sector for the same purpose. That makes 18 quads for space and water heating --the equivalent of more than 9 million barrels of oil per day, which is more oil than we imported last year.
>
> "The figures, rough as they may be, leave little doubt that the extent to which solar technologies are used bears a direct relationship to the nature of the consumer incentives extended by the Federal government. The program outlined in (the Solar Bank bill) must be adopted if solar systems are to penetrate the residential and light commercial markets significantly. The costs are clearly in line with subsidies granted to users and developers of other energy sources." (Hayes, before the House Banking Committee.)

It was most reassuring yesterday to hear Denis still quoting these figures and urging this action. DOE told us at the time that roughly 200 companies are now selling or attempting to sell solar-space-and-water-heating systems. Clearly, the successful and timely commercialization of these systems becomes extremely important if solar is to begin helping the nation meet its energy needs.

Mayor Bradley of Los Angeles gave us what I feel is a good example of the Bank's potential impact:

> "A $2,000-solar-hot-water system paid for over a ten-year period with a 12 percent home improvement loan would actually cost a total of $3,443. However, the same $2,000 system paid for over 10 years at an interest rate of six percent would actually cost $2,664. This lower interest rate would save the purchaser almost $800, while costing the government approximately $450 at the beginning of the loan period. The bank or other private lender would have tied up not $2,000, but $1,500 or so.
>
> "With current prime commercial lending rates at 12 to 13½ percent, most consumers are dissuaded from making the initial investment to purchase a solar energy system, even though it will more than pay for itself within a few years of purchase."

Using the figure of an average loan per residential dwelling at $5,000 and assuming a 25-percent subsidy of the total cost of that loan under the Solar Bank program, the Senate Banking Committee staff estimates over 1,000,000 solar loans of the same amount could be supported under the Solar Bank's provisions, as they stand today.

EXISTING INCENTIVES AND PROGRAMS

What of existing programs to encourage solar energy use? Clearly, some solar energy use is already being spurred by existing federal and state programs. According to GAO, 22 states have passed laws since 1974 "exempting solar equipment from state sales taxes or offering income tax incentives to homeowners and businesses installing solar heating systems, for example, California's 55 percent income-tax credit up to $3,000 or Arizona's 35 percent state income-tax credit. Twenty-nine states have laws reducing the amount of increased property taxes resulting from installing solar heating devices."

While state efforts have focused on financial incentives, federal programs, as you know far better than I, have been geared primarily to research and development of standards and certification procedures, notwithstanding the 1978 National Energy Act which provided limited financial incentives for installation of solar energy equipment, such as the non-refundable income tax credit for individuals who install solar energy devices in their principal residences; a business tax credit for solar investment, and a $100-million subsidized loan program for installation of solar devices in one-to-four-family residential structures. The Solar Bank proponents in the Congress have viewed these incentives as simply not enough. I might add that this time around, we don't see an 18-month lag between proposals of the incentive and its adoption, as occurred in 1977, when the proposed tax credits for solar installation did not get through Congress for 18 months after they were proposed. The results then - the industry's upward sales curve collapsed as consumers waited to see if the government would actually do what it promised - were disastrous. This time, we've gotten from submission of the bill to pending conference in seven months; the Administration is in agreement; and money is in the budget. However, to cite again that first law of Congress: nothing is ever certain.

Speaking of uncertainties, I am almost afraid to mention the solar tax credits in the Senate Finance Committee's "windfall profits" bill now on the floor of the Senate. You will recall that Senator Packwood of Oregon was releatively successful in getting his fellow committee members to expand these credits about two months ago. Whether his efforts will survive on the Senate floor, then in conference and finally under the President's review is a matter of, at the least, some uncertainty.

OTHER SOLAR LEGISLATIVE ISSUES

The Congressional Solar Coalition, at last count including 44 House members and 11 Senators, this fall has enumerated solar issues for Congressional action or oversight, although without any set dead-

lines:

- DOE solar energy programs reorganization
- Solar energy employment and training
- Solar access and sun rights review
- Local energy management
- Solar energy in federal buildings
- Federal power marketing and Rural Electrification Administration (REA) renewable energy requirements
- International solar energy research, development and demonstration
- Global market survey

This list has surprisingly and pleasingly close parallels to that developed in September by Resources for the Future of Washington, DC, for the Ford Foundation: Energy: The Next Twenty Years, in its "Recommendation Nine: Remove Impediments to Use of Solar Energy".

To close, let me cite again my first Congressional law: uncertainty. People vastly more knowledgeable and experienced in these matters than I, such as the Solar Lobby, the Energy Group at the Congressional Office of Technology Assessment (OTA), and at DOE, HUD and other federal agencies directly involved on these matters, can bear witness to the seeming inevitability that, in Congress, just when you think you've got it is when you find out precisely you don't. Fortune puts it definitely more eloquently, if somewhat more wordily:

> "To date, the hisotry of solar has been the story of Tantalus: year after year the prize has remained, maddeningly, just beyond reach. Each new rise in the prices of conventional energy has been accompanied by fresh revelations of seemingly intractable problems or diseconomies. It has proved harder than the pioneers imagined to overcome the inherent difficulties of harnessing an energy form that is stupendous in the aggregate, but dilute in any given setting. The kind of progresss that is evident today suggests that the period of solar frustration is drawing to a close."

When Senator Proxmire, the Banking Committee chairman reknown for fiscal frugality, Senator Lugar of Indiana and Senator Armstrong of Colorado - both formidable foes of increased federal spending - and Southern moderates such as Alabama's Senator Stewart and North Carolina's Senator Morgan all can readily come together and agree, as they did on this Solar Bank - they actually almost doubled the funding the President proposed - then, perhaps, we really are drawing to that close.

Session III: Poster Presentations

SOLAR HEATING RESULTS FOR THE NAMBE COMMUNITY CENTER* 01000A/

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ABSTRACT

The Nambe Community Center is a solar heated adobe community building for the Nambe Indian Pueblo, located twenty miles north of Santa Fe, New Mexico. The 279 m^2 building makes use of several passive solar heating features. The active solar heating system is an air system using a 15,420-kg rock storage bed. The solar collector system consists of a 41 m^2 vertical collector with a 98 m^2 reflector. The system also allows night evaporative cooling of the rock bed for summer cooling operation. The heating season of 1978-79 was the first full season of operation for which complete operational data were obtained. These results are presented along with a discussion of the use of the rock bed for summer night cold storage.

SYSTEM DESCRIPTION

The Nambe Indian Pueblo Community Center consists of an existing building with a structure added to provide space for tribal offices and a classroom. The solar energy system was designed by the Los Alamos Scientific Laboratory and funded by the Department of Energy as part of the solar heating demonstration program.

Building Description

The 279 m^2 building is of single story construction, built of adobe with 0.36-m walls. Foam insulation protected by a stucco exterior is used to improve heat retention of the exterior walls. The mass of the adobe provides a large heat capacity to reduce inside temperature fluctuations, and large window areas act as passive solar collectors with the adobe walls serving as heat storage.

Some fixed architectural considerations affected the design of this system. The size of the addition to the existing building limited the size of the collector array, and the building faces considerably west of south (approximately 45°). Reorienting the collector on top of the building would have resulted in substantial system performance gains, but could not be justified due to the weight of the aesthetic losses.

Solar Heating System Description

The heating system is an air system using rock bed storage. The Nambe system uses a vertical collector with a horizontal reflector. The 41 m^2 solar collector comprises the south wall of a narrow second story room running the length of the north edge of the building. The collector consists of four 8.9 by 51 cm air ducts stacked on edge. The air flows in through two of the ducts and returns through the other two. Solar heat is absorbed on the black-painted surface of the air ducts. One sheet of tempered glass covers the air ducts. The reflector is 4.9 by 20.1 m and constructed of Coilzak aluminum.

Air flows from top to bottom in the 15,420-kg rock storage bin when solar heat is being collected and stored. Thermal energy is removed from storage to heat the building by reverse air flow through the rock bin. The rock bed is located under the classroom floor, as shown in the cross section of the building in Fig. 1. A gas furnace is in series with the rock bed for auxiliary energy.

Controls

In the heating mode, the solar collectors are controlled by a temperature differential between the collector face and the rock bin. The collector supply fan is turned on when a 8°C temperature differential is established, and off when the differential drops below 1.7°C.

One dual setpoint thermostat is used in the building to control delivery of heat. When the room temperature drops below the higher setting, the building supply fan is turned on, supplying heat from storage. If the room temperature drops below the lower setpoint, the furnace is turned on.

In the cooling mode, only one thermostat setpoint is used. If the building calls for cooling, the room fan is turned on to remove cold air from storage. If the rock bed average temperature rises above a preset level, the evaporative cooler runs on low speed. After midnight, the cooler runs at high speed for a predetermined time to charge the rock bed.

*This work performed under the auspices of the Department of Energy, Office of Solar Applications.

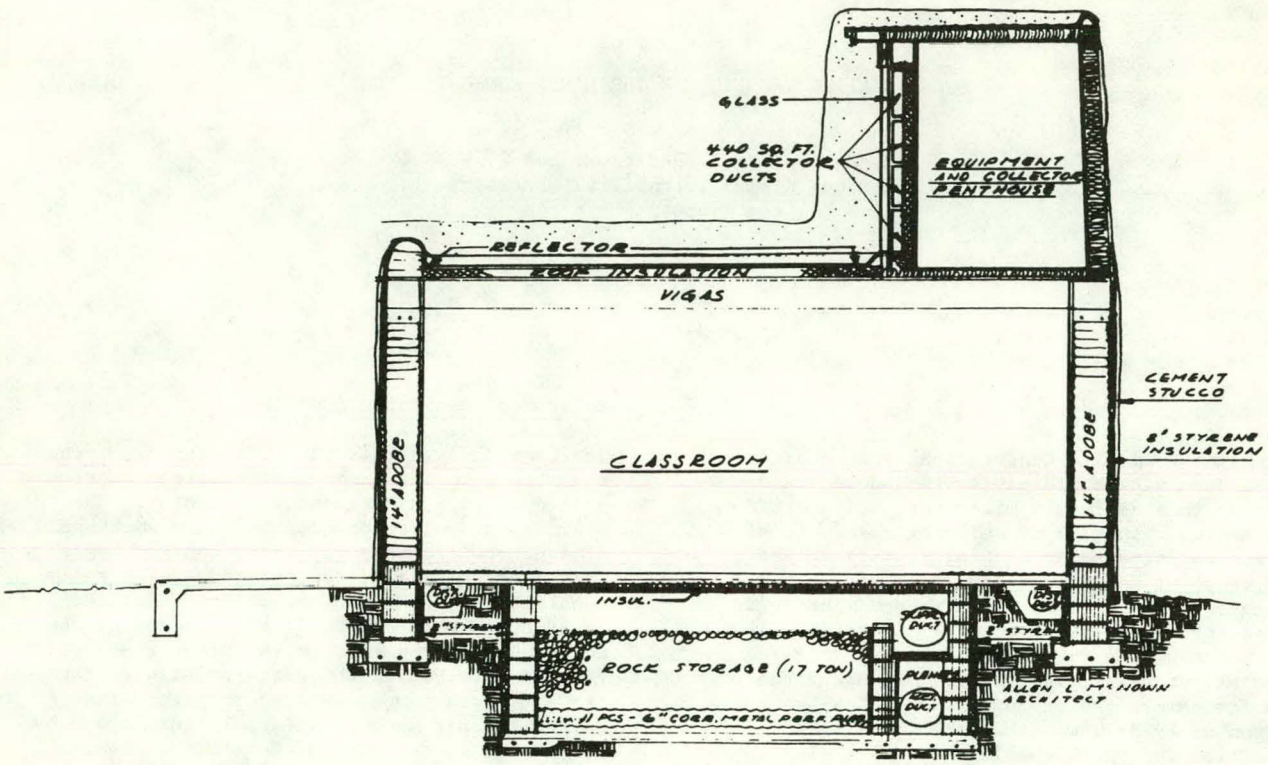


Fig. 1. Building section at rock bed.

SYSTEM PERFORMANCE STUDIES

The Nambe building is instrumented with thermocouples and air duct monitors to measure energy flows in the solar energy system. Three pyranometers are used to measure solar flux--one for the horizontal radiation, and two on the vertical face of the collector. Additional thermocouples are placed in the rock bin to study the charactertistics of the storage medium.

The data are collected and reduced to temperatures, flow rates, and energies in engineering units via an HP-9830 calculator-based system with a power-fail/auto-restart module. Data are recorded on cassettes, which accommodate one week's data. A negligible amount of data was lost during the heating season.

Heating System Performance

Heating for the Nambe building was required from October 1, 1978, through the middle of May, 1979. The mean daily measured system energies are summarized in Table I. Tabulated are:

QHORZ	Horizontal radiation.
QINC	Collector incident radiation.
COLL	Collector output energy.
STOR	Energy actively removed from storage.
AUX	Auxiliary energy.
DEL	Energy delivered to the building.
ELECT	Electrical (fan) energy.
PS	Per cent solar heating.

TABLE I

ENERGY SUMMARY FOR NAMBE COMMUNITY CENTER

MON	QHORZ	QINC	COLL	STOR	AUX	DEL	ELECT	PS
Oct	640	785	216	53	25	60	14	89.7
Nov	405	596	173	124	216	329	25	44.4
Dec	391	680	234	127	375	482	20	38.5
Jan	408	601	218	107	583	670	18	27.3
Feb	597	828	341	213	337	512	31	50.3
Mar	724	730	254	127	208	319	27	55.1
Apr	854	635	194	66	130	195	19	59.8
May	948	583	147	39	64	100	17	69.6
Yr	598	685	226	111	254	348	22	47.1

Units: MJ/day

The term DEL results from an independent measurement, in addition to measurements on the storage outlet side and furnace. Agreement between DEL and (STOR + AUX) is within 5 per cent for the year.

Per cent solar is calculated assuming that all of the energy delivered to storage is useful in heating the building. It is seen that over the year, only half of this energy is delivered actively to the building. However, losses from the rock bed are to the building because of its placement underneath the heated space.

The occupants of the building frequently caused the furnace to run when it was not required.

:ten after a weekend the thermostat would be .aced at its maximum setting to warm the building, causing the furnace to run even though storage was hot from two days of charging. The building was also generally kept at a fairly high temperature by current standards. Thirty per cent of the winter's auxiliary energy was used when the air temperature was higher than 23°C, and auxiliary was frequently used when the temperature was as high as 26°C.

Table II gives the system performance parameters for the 1978-79 heating season. The nonuseful loss fraction, β, is assumed to be unity if the ambient temperature is over 10°C, and zero if the temperature is less than 10°C. Monthly values of β are calculated by averaging daily values. The other monthly values tabulated are:

$\overline{Q_u}^*$	Average daily useful solar heat delivered to load
$\overline{Q_H}$	Average daily nonrenewable energy savings.
SCOP'	Solar COP.
SCOP	Solar controlled COP.
η_c	Collector efficiency.
η_{st}	Solar heating thermal efficiency.
η_s	Solar heating system efficiency.

Collector output divided by collector incident, regardless of whether the collector is running, is used to calculate η_c. For the purpose of calculating the nonrenewable energy savings, conventional furnace efficiency is assumed to be 0.65, and overall electrical power generation efficiency is assumed to be 0.26, to give a ratio of 2.50.

TABLE II

MONTHLY PERFORMANCE PARAMETERS

	β	$\overline{Q_u}^*$ MJ/DAY	$\overline{Q_H}$ MJ/DAY	SCOP'	SCOP	η_c %	η_{st} %	η_s %
Oct	.23	178	144	13.0	3.9	27.4	22.7	18.3
Nov	.07	170	107	6.7	4.9	29.0	28.5	17.9
Dec	.00	234	184	11.6	6.3	34.4	34.4	27.0
Jan	.00	218	174	12.2	6.0	36.4	36.4	28.9
Feb	.00	341	264	11.0	6.9	41.2	41.2	31.8
Mar	.00	254	187	9.5	4.8	34.8	34.8	25.7
Apr	.57	122	73	6.2	3.4	30.6	19.2	11.5
May	.67	75	32	4.4	2.3	25.2	12.9	5.5
Yr	.16	206	152	9.6	5.1	33.0	30.1	22.2

Daily Performance

Bar graphs of daily collector input and output and of daily solar and auxiliary load are given in Figs. 2 and 3, respectively. Figure 4 shows a scan of rock bed temperatures for February, 1979. The periods when no charging of storage can be observed are those periods when the collectors are directly heating the enclosure.

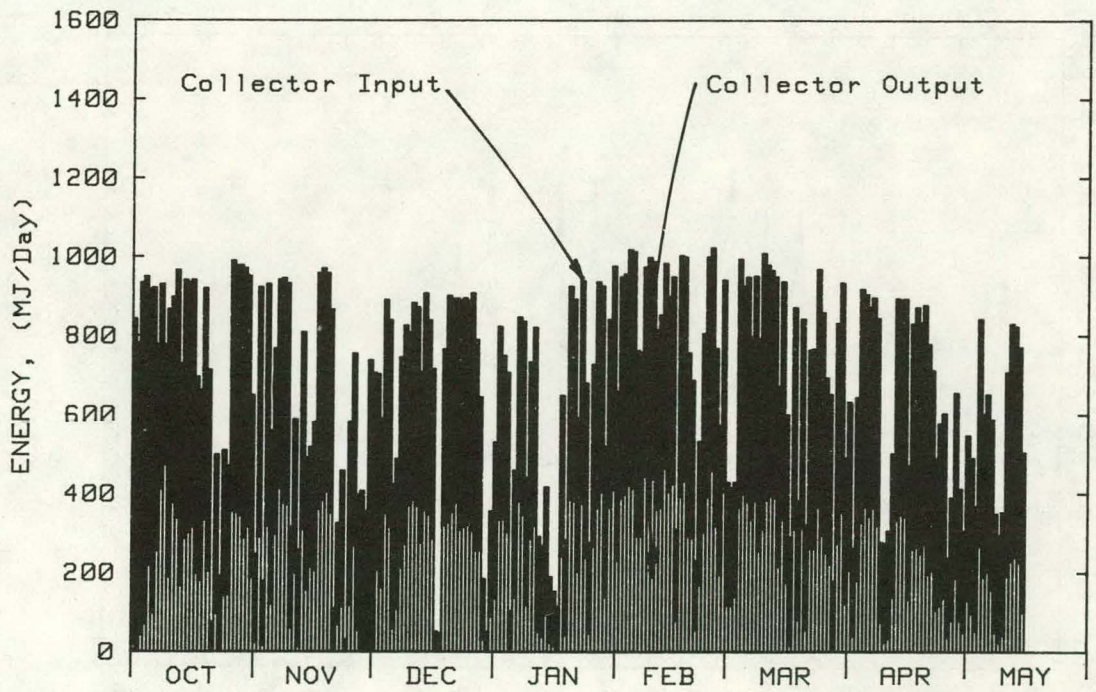


Fig. 2. Daily collector input and output, 1978-79 heating season.

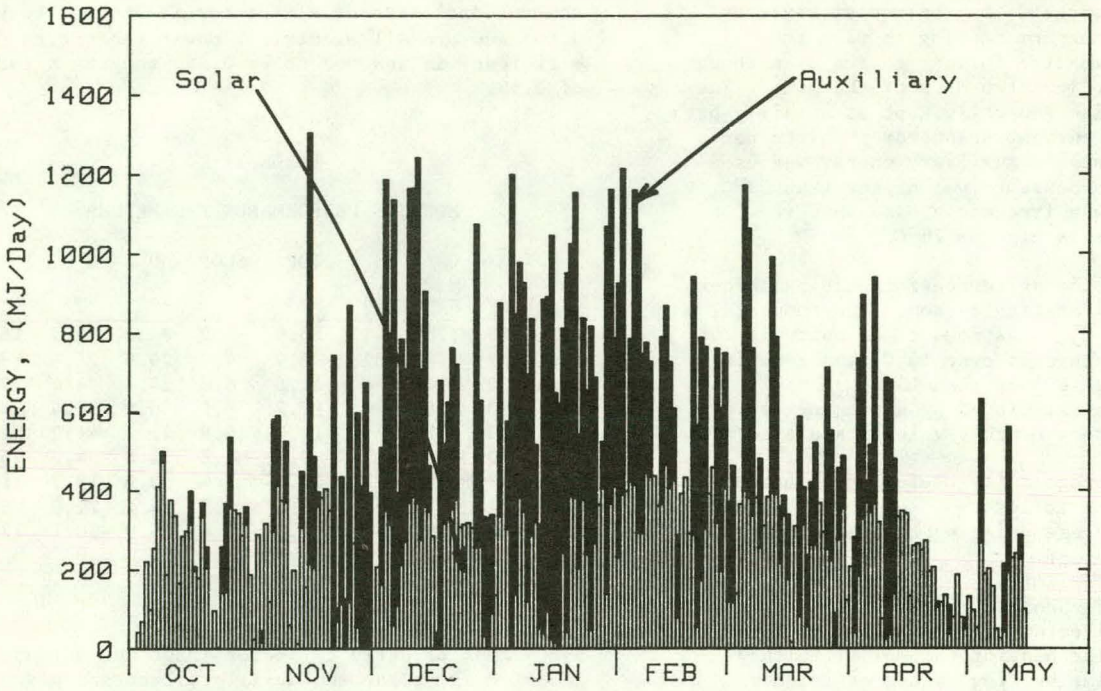


Fig. 3. Daily total load, solar and auxiliary, 1978-79 heating season.

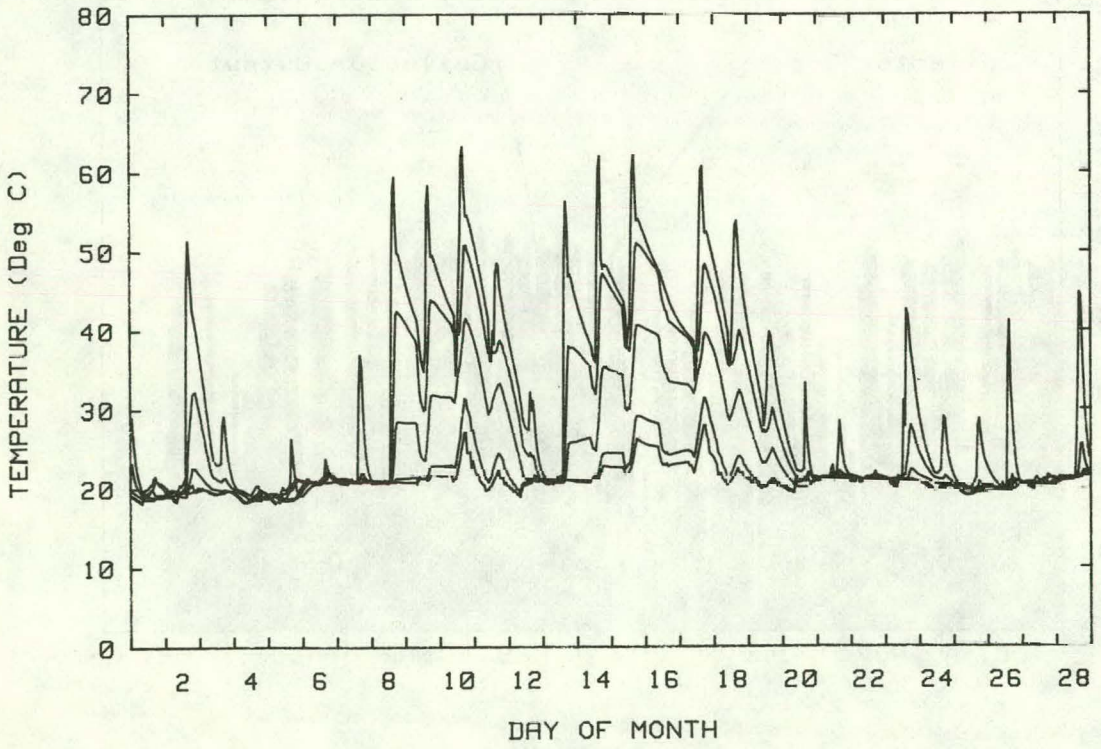


Fig. 4. Storage temperatures, February, 1979.

ooling System Performance

The cooling system was operational from the second week in June, 1979, through mid-September. Normal operation of this system is to charge the rock bed for four hours at night and to run the room fan for nine hours during the day. If cold storage is exhausted during the day, the cooler would start, but this usually did not happen unless ambient temperatures were 32°C or above for several hours. However, the system was not operated consistently in the normal mode. In particular, the occupants tended to lower the thermostat setting to start the fan in the morning, and forget to raise the setting in the evening. This caused the fan to run all night, defeating the rock bed cold storage mode. It was noted that if the setting were left at 24°C and not moved, the system would operate normally.

Several periods of normal operation were observed, one of which is summarized in Table III. The system cooling COP for this was 3.3, which was typical for other such periods.

A daily plot of rock bed temperatures is shown in Fig. 5 for the same period. The charging and depletion of storage can be seen in this figure.

TABLE III

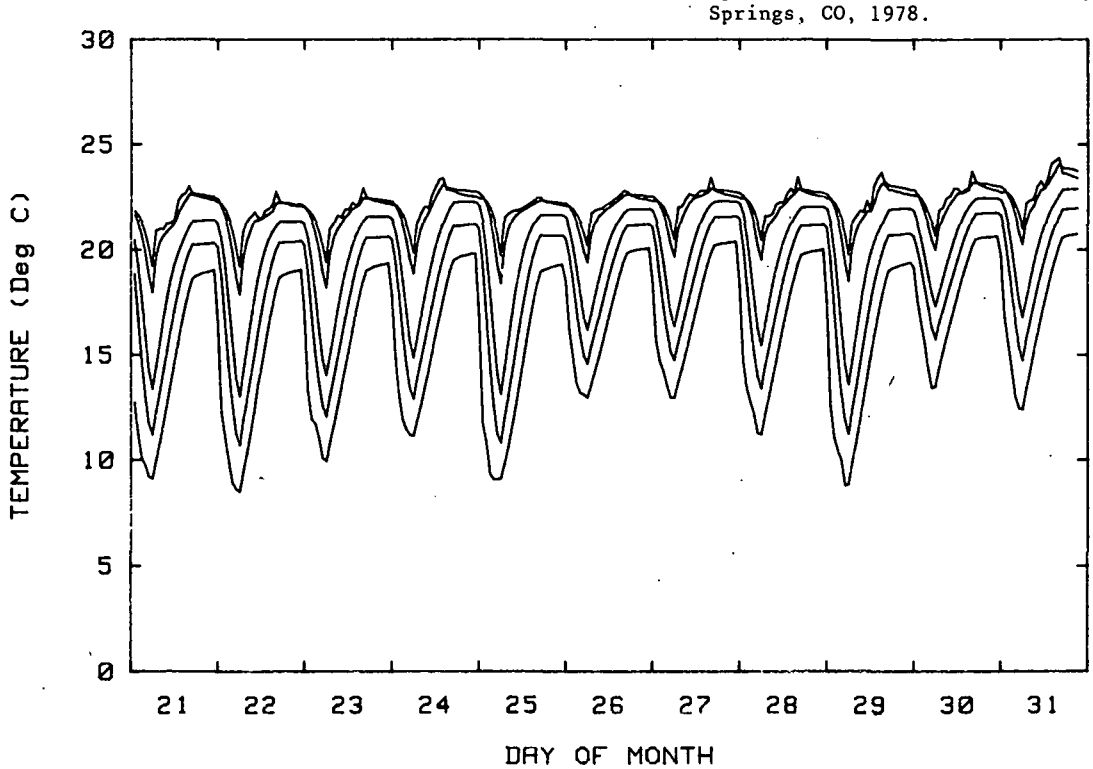
COOLING SYSTEM PERFORMANCE FOR NAMBE COMMUNITY CENTER
August, 1979

DAY	EVAP MJ	STOR MJ	DEL MJ	ELEC MJ	SCOP
21	- 178	- 173	- 106	28	3.8
22	- 187	- 176	- 106	28	3.8
23	- 162	- 162	- 93	28	3.3
24	- 153	- 162	- 96	28	3.4
25	- 193	- 175	- 109	28	3.9
26	- 122	- 179	- 72	28	2.6
27	- 128	- 135	- 78	28	2.8
28	- 153	- 144	- 81	28	2.9
29	- 188	- 188	- 117	28	4.2
30	- 109	- 127	- 64	28	2.3
31	- 137	- 149	- 86	28	3.1
Total	-1710	-1770	-1008	308	3.3

REFERENCES

1. J. D. Balcomb, et al, "Nambe Pueblo Community Building Solar System," Proceedings of Solar Heating and Cooling Demonstration Program Contractors' Review Meeting, New Orleans, LA, December 5-7, 1977.

2. H. S. Murray, et al, "Solar Heating Results for the Nambe Community Center," Proceedings of the Solar Heating and Cooling Systems Operational Results Conference, Colorado Springs, CO, 1978.



g. 5. Storage temperatures, August, 1979.

NOTES

THERMAL CHARACTERISTICS OF A ROCK STORAGE UNIT FOR SOLAR ENERGY

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ABSTRACT

A solar collection, storage, and distribution system was designed and installed to supply heated air to an environmental test chamber at the University of Maryland Broiler Research Facility at Salisbury, Maryland.

The collector consists of eight 1.22 x 4.88 m panels providing approximately 41.81 m^2 effective collector area. The rock storage unit was constructed with two sheets of 1.59 cm plywood separated by 5.08 x 15.24 cm studs on 60 cm centers. The area between the studs was filled with 15.24 cm of fiberglass insulation. The unit was filled with approximately 20.8 x 10^3 kg of No. 6 washed gravel.

Tests were conducted to determine collector efficiency, electrical power usage, pressure drop, and thermal characteristics of the system at air flow rates of 23, 15, and 10 m^3/min.

Tests results indicate that for the range of air flow rates considered, collector efficiencies decrease as air flow rates decrease. The results also indicate that for a given air flow rate the pressure drop through the rock storage increases linearly. For the range of air flow rates considered, tests also indicate that electrical power usage increases linearly as air flow rates increase. In addition, the study showed that pressure drop across the rock storage (ΔP) varied as the square of the apparent velocity $(V)^2$. The resulting equation becomes

$$\Delta P = 2.3 \times 10^{-3} \ (V)^2$$

for No. 6 washed gravel with a range of diameters of 0.65 - 3.18 cm.

The results also indicate that thermal stratification along the centerline of the rock storage increases as air flow rates decrease. Finally, as air flow rates decrease tests show an increase in thermal stratification from the centerline of the rock storage perpendicular to the direction of air flow.

INTRODUCTION

ture acceptance of solar energy systems as a vile alternative to fossil fuels depends largely on increased system efficiencies and reduced costs. Researchers have made considerable advancements in the design of efficient collectors and are continuing to investigate materials which will hopefully make collector efficiencies even higher. However, even with increased collector efficiencies, inexpensive and efficient thermal storage devices are a prerequisite to wide acceptance of solar systems.

Basically a thermal storage device is a component in a solar system in which energy is accepted as heat from a system transfer fluid and is stored usually as the specific heat or the heat of fusion of the media. The heat is stored by the media until its use is required. At that time the transfer fluid is heated by the stored energy and pumped to the desired application.

The desirable characteristics of a thermal storage system can be summarized as follows:

- The system should be able to receive and discharge energy at the maximum rate without excessive driving forces.
- The system should have low self-discharge characteristics.
- The system should be capable of a large number of charge-discharge cycles without deterioration.
- The system should have the capability of storing a large amount of energy per unit volume.
- The system should be inexpensive.

An extensive literature search indicates several storage mediums have one or more of the above characteristics. The three most widely used media are water, rock, and eutectic salts.

Recently there has been increased interest in utilizing air collectors in conjunction with rock storage since rock storage systems meet the desirable characteristics of a thermal storage system mentioned above.

The increased interest in utilizing rock as a ther-

mal storage system has created the need for design of efficient rock storage systems. A design which would maximize storage capabilities and concurrently minimize electrical power usage and storage space is needed. Such information would aid architects and engineers to more accurately design rock storage systems in relation to such parameters as heating and cooling loads, optimum storage shape and size, rock size (gradation), blower requirements, air flow rates, etc.

EXPERIMENTAL DESIGN

To consider the optimum design parameters of a rock storage unit, an air system utilizing rock storage was designed and installed at the University of Maryland Environmental Broiler Research Facility at Salisbury, Maryland.

The collector contains eight 1.22 x 4.88 m panels providing approximately 41.81 m^2 effective collector area. The collector absorber plate was fabricated from corrugated aluminum sheets, painted flat black and covered with two layers of double-strength window glass separated by a 1.27 cm air space. Located beneath the absorber plate is a 5.08 cm plenum through which air is passed to gather heat.

The rock storage unit was constructed with two sheets of 1.59 cm plywood separated by 5.08 x 15.24 cm studs on 60 cm centers. The area between the studs was filled with 15.24 cm fiberglass insulation. Outside dimensions of 2.13 x 2.13 x 5.18 m with a 1.82 x 1.82 x 0.305 m air plenum located at the inlet and outlet ends provide for approximately 14.27 m^3 of storage. The storage unit was filled with approximately 20.8 x 10^3 kg of No. 6 washed gravel. Table 1 and Fig. 1 show the physical characteristics of the gravel and a schematic of the system respectively.

TABLE 1. PHYSICAL CHARACTERISTICS OF THE ROCK STORAGE UNIT

Volume of Storage Unit	14.17 m^3
Rock Density	1457.00 kgs/m^3
Specific Heat	0.20 cal/gram-^{0}C
% Voids	45.00
Volume of Voids	6.42 m^3
Volume of Rocks	7.85 m^3
Range of Rock Diameters	0.65 - 3.18 cm
Av. Rock Spherical Dia.	1.60 cm
Total Weight of Rocks	20.80 x 10^3 kgs

DISCUSSION AND RESULTS

Collector Efficiency

Incoming solar radiation data was monitored with an Eppley black-and-white pyranometer installed in the same plane as the collector. The millivolt output from the pyranometer was integrated at one-hour intervals using an AGM Electronics Inc. integrator module, Model EA 40011-3M, with the output monitored on a Doric 220 data acquisition system and recorded on magnetic tape using a Kennedy Model 1600/S incremental magnetic tape recorder.

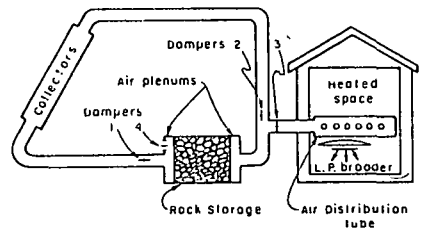


Fig. 1 Schematic of the System

The performance of the collector was determined using Equation (1).

$$\eta = F'(\tau\alpha) - F' U_L \left[\frac{\frac{T_{f,i} + T_{f,o}}{2} - Ta}{I} \right] \quad (1)$$

By plotting collector efficiencies against the appropriate $\Delta T/I$, equation (1) will result in a linear relationship where the slope of the line will be some function of U_L and the Y intercept some function of $\tau\alpha$. Figure 2 shows the collector performance for November 1978.

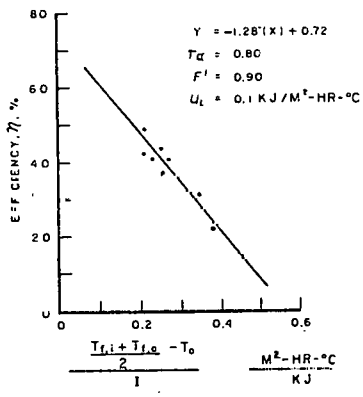


Fig. 2 Collector Efficiency for November 1978

Pressure Drop in the Rock Storage Unit

To store energy in the rock, heated air from the collector is passed laterally through the rock storage unit. Since the rock is at a lower temperature, a transfer of heat from the air to the rock results in energy stored as sensible heat. Two concerns with such a unit are electrical power usage and thermal stratification. To determine electrical power usage a kilowatt-hour meter was installed to monitor electrical energy requirements of the blower and pressure ports were installed through the centerline of the rock storage unit at 0.0, 0.3, 0.6, 1.2, 2.1, 3.4, and 4.0 m distances to monitor the pressure drop across the rock storage. Tests were conducted at air flow rates of 23, 15, and 10 m^3/min. Each test was conducted

m 9 a.m. to 4 p.m. on three consecutive days of approximately equal solar intensity. For a given air flow rate Fig. 3 indicates a linear relationship between pressure drop and distance into rock storage. For the range of air flow rates considered, results also indicate a linear relationship between air flow rates and electrical power usage (Fig. 4).

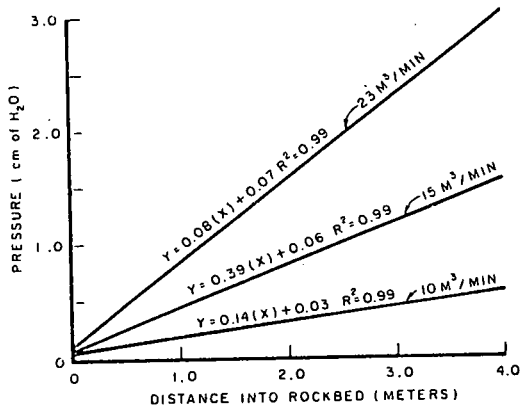


Fig. 3. Pressure Drop Versus Distance into Rock Storage for the Three Air Flow Rates

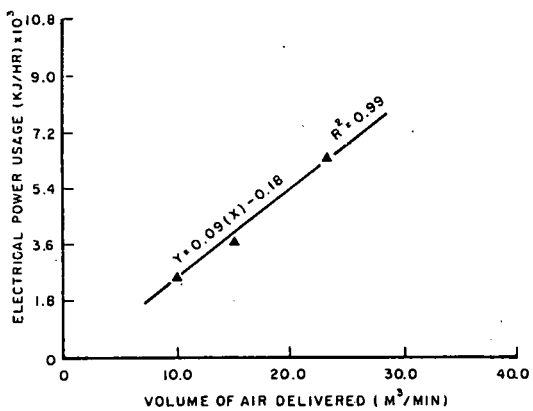


Fig. 4. Electrical Power Usage as a Function of Volume of Air Delivered

Additional tests were conducted to determine the pressure drop across the rock bed versus the apparent velocity. To determine this relationship, the pressure drop across the rock bed was monitored for the three air flow rates mentioned earlier. The equation of the resulting plot on log-log paper indicates the pressure drop across the rock varies as the square of the apparent velocity. The equation has the following form:

$$\Delta P = C(V)^2 \qquad (2)$$

Using a least squares regression analysis, the constant (C) was determined to be 2.3×10^{-3}, with $R^2 = 0.98$.

Thermal Stratification in the Rock Storage Unit

To determine the thermal characteristics of the rock storage unit, 13 thermocouples were installed at 0.3, 0.6, 1.2, 2.1, and 3.4 meter intervals within the rock storage, (Fig. 5). In addition, the air temperatures entering and leaving the rock storage were recorded. Tests were conducted at air flow rates of 23, 15, and 10 m^3/min to obtain temperature history curves for the charging cycle. Each test was conducted from 9 a.m. to 4 p.m. on three consecutive days at approximately equal solar intensity.

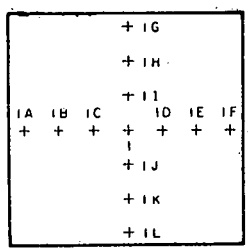


Fig. 5. Cross Section of the Rock Storage Showing Typical Thermocouple Arrangement

The centerline temperature profiles for the three air flow rates considered indicate thermal stratification increases as air flow rate decreases (Fig. 6, 7 and 8).

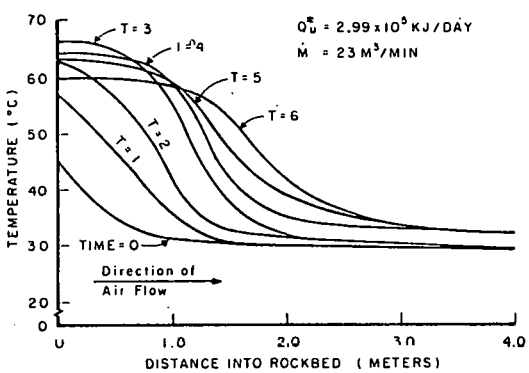


Fig. 6. Thermal Stratification as a Function of Time Through the Centerline of the Rock Storage for M = 23 m^3/min.

Temperature profiles perpendicular to the direction of air flow at the 0.6 meter interval were recorded at the end of each test. From Fig. 9 it can be concluded that for a given air flow rate the temperature difference for the horizontal and vertical profile at the four edges of the rock storage was insignificant. But Fig. 9 also indicates thermal stratification from the centerline outward increases significantly in all directions as the air flow rate decreases.

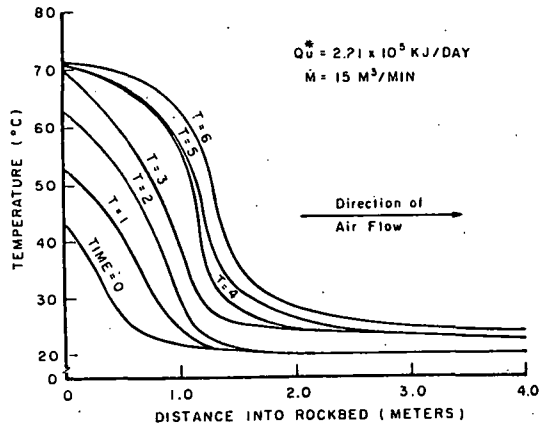


Fig. 7. Thermal Stratification as a Function of Time Through the Centerline of the Rock Storage for $\dot{M}$ = 15 m^3/min.

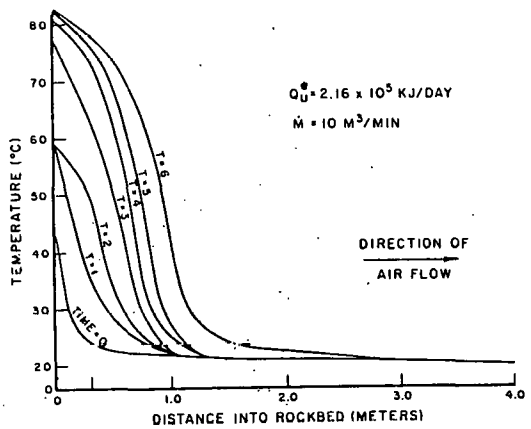


Fig. 8. Thermal Stratification as a Function of Time Through the Centerline of the Rock Storage for M = 10 m^3/min.

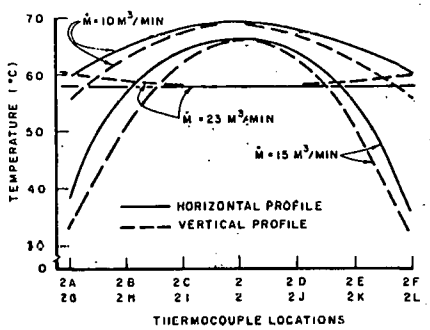


Fig. 9. Thermal Stratification in the Vertical and Horizontal Profiles for Air Flow Rates of 23, 15, and 10 m^3/min Respectively

To determine the self-discharge characteristics, the rock storage was charged and temperatures were recorded for 48 hours with no input or output. Results indicate that significant losses occurred only at the inlet and outlet plenums where insulation was minimum (Fig. 10).

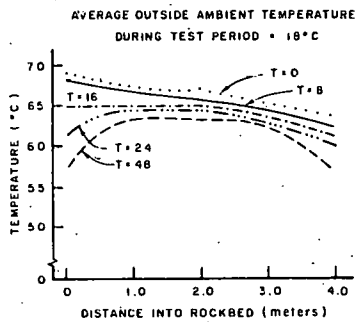


Fig. 10. Thermal Decay of the Rock Storage as a Function of Time for no Input or Output

CONCLUSIONS

For the rock storage unit used in this study, results indicate that for a given air flow rate the pressure drop per foot of bed length increases linearly. Results also indicate that for the range of air flow rates considered, electrical power usage increases linearly with increased air flow rates. In addition the study shows that pressure drop across the rock storage varies as the square of the apparent velocity, i.e.:

$$\Delta P = 2.3 \times 10^{-3} (V)^2 \qquad (3)$$

Test results also indicate that thermal stratification along the centerline of the rock storage increases as air flow rates decrease. As air flow rates decrease, tests show increase in thermal stratification from the centerline perpendicular to the direction of air flow.

ACKNOWLEDGMENTS

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NOMENCLATURE

F' = Solar collector efficiency factor

I = Integrated solar radiation incident to the collector per unit time per unit area, (kj/m^2-hr-^{0}C)

$\dot{M}$ = Volume of air delivered per unit time, (m^3/min)

ΔP = Pressure drop across the rock storage, (cm of H_2O)

$\dot{Q}_u^*$ = Average daily solar heat delivered to storage, (kj/day)

T_a = Ambient outside air temperature, (^{0}C)

$T_{f,i}$ = Fluid inlet temperature to the collector, (^{0}C)

$T_{f,o}$ = Fluid outlet temperature from the collector, (^{0}C)

U_L = Solar collector loss coefficient, (kj/m^2-hr-^{0}C)

V = Apparent air flow per square meter of cross-sectional area, (m/min)

η = Solar collector efficiency

$\tau\alpha$ = Effective collector transmissivity-absorptivity factor

NOTES

PERFORMANCE OF THE SUNEARTH HOUSE

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INTRODUCTION

Minimal energy use in buildings is successfully demonstrated in the SunEarth House. This house is an earth-covered passive solar heated and naturally cooled residence located in Longmont, Colorado. It was designed, built, and marketed by Colorado Sunworks in 1977-78. Performance is monitored by IBM computerized tracking of 90 energy sensors placed in the building during construction. Performance reports are published by the Department of Energy under the National Solar Data Program.(1) During the winter of 1978-79 the house was considered to be 100% solar heated because the occupants turned off the furnace pilot light and left it off

DESIGN

Minimal energy use in buildings is a goal that can be reached at moderate cost by mixing load reduction techniques, passive solar heating, and good thermal/architectural design. The design of the SunEarth House begins with concrete wall construction insulated on the exterior. Earth is bermed up over the walls and piled one foot deep on the roof to create an earth contact shelter.

The south side exposes 300 sq. ft. of glass windows to the winter sun. This directly warms large barrels of water about 10 F. on a clear day. Atrium windows admit sunshine directly to light-colored floor tiles which then reflect the light to interior concrete walls, storing more energy for night time use. Typical daily temperature rise of the massive concrete walls is about 3°F. On winter nights moveable insulation, (Beadwall®), converts large window areas to an R 20 heat loss barrier.

Other energy conserving features of the SunEarth House include a garage buffer to the northwest, air-lock entry, vertical skylights located over the north wall, airtight construction, thermal envelope ceiling plenum for natural heating and cooling, fireplace piped under the floor slab to outdoor air, summer shading devices, and passive tank type water heaters.

DETAILS

The house is a three bedroom residence comprising about 1800 sq. ft. Complete construction details have been published elsewhere. (2) (3)

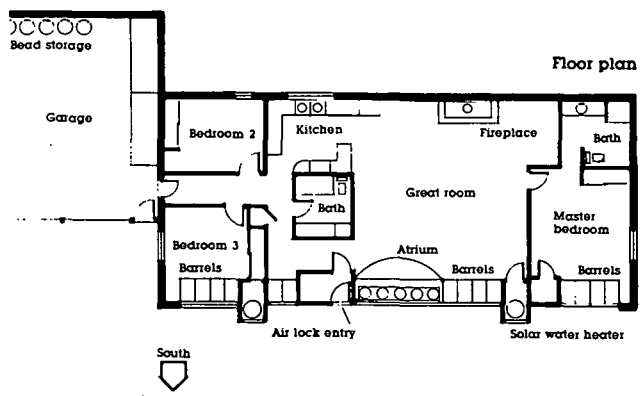


Fig. 1. Floor Plan

The floor plan shown above emphasizes elongation of the south side of the building for good solar exposure. Rooms are laid out to aid natural convection and radiation heat flow. Note that the south collector storage wall can "see" much of the north wall for effective distribution of heat. House interiors are pleasant and attractive. Rooms are arranged in a spacious manner around the great room.

In winter stored solar heat is released from the water containers as needed. Comfort zones are well balanced throughout the living spaces.

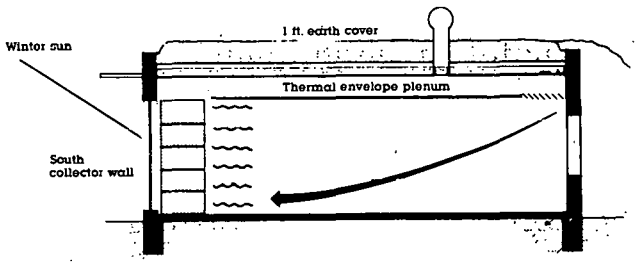


Fig. 2. Winter Mode Schematic

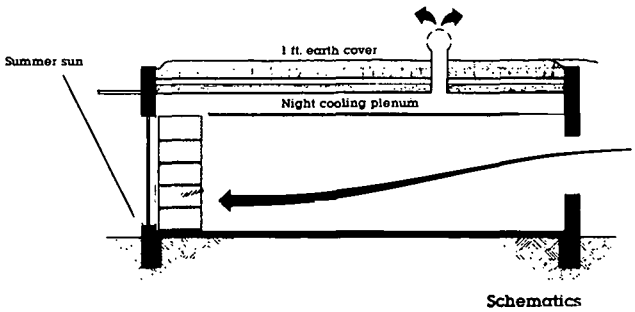


Fig. 3. Summer Mode Schematic

In the summer interior thermal mass is cooled down during the night by providing natural air flow paths. Daily heat gains are rejected by this method through turbine roof ventilators. The cooled massive house is then closed up on hot summer days for comfortable living.

Thermal mass in the water wall is 22,680 BTU/F contained in fifty four 55 gallon drums placed behind the south glass. Additional secondary thermal mass, (most of it out of the direct sunlight), in the concrete walls and floors is 50,000 BTU/F. Most of this is in the walls. About 2/3 of the floor slab is covered with thick carpet and has a low thermal admittance.

The extremely large quantity of thermal mass has contributed greatly to experienced comfort levels in the house by stabilizing temperature excursions regardless of outdoor conditions. Average temperature swings measured throughout the winter have been in the range of 4°F-6°F. A balanced thermal environment inside the house, due largely to good heat transfer from south side to north side, also contributes to experienced comfort levels. In this house the walls are almost always warmer than the air creating an environmental situation which is the reverse of the conventional space heating environments. The higher mean radiant temperature of surrounding walls brings further energy savings by allowing the air temperature to fall to the low 60's F. without causing apparent discomfort.

PERFORMANCE

The circle diagram below represents the total heat required in December 1978, to maintain comfort levels in the house. All of the space heating demand, after internal gains are counted, was supplied by the passive solar space heating system. This pattern is typical of all winter months.

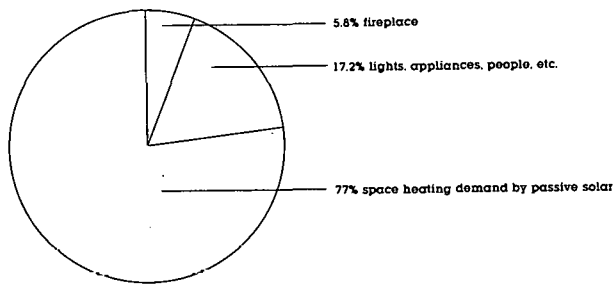


Fig. 4. House Heating Load December 1978
10 Million BTU

Weather conditions measured at the site are presented in the table below.

Month	Daily Solar Energy Incident Per Unit Area (BTU/Ft^2-Day) Measured	Long-Term Average(1)
Nov 78	1,257	1,712
Dec 78	1,601	1,690
Jan 79	1,493	1,856
Feb 79	1,488	1,769
Mar 79	1,126	1,496
Apr 79	1,019	1,106
May 79	584	890
Average	1,226	1,502

Month	Ambient Temperature (F) Measured	Long-Term Average
Nov 78	35	39
Dec 78	20	32
Jan 79	16	30
Feb 79	31	32
Mar 79	39	37
Apr 79	49	48
May 79	53	57
Average	35	39

ıe collection efficiency graph below is measured when the Beadwall is open, ("operational incident solar energy"), and with respect to total incident solar energy that fell on the glass. It should be noted that the values of the collection efficiency do not include thermal losses through the south facing glass. These losses are included as a part of the building heating load, since glazed areas are a portion of the exterior of the building.

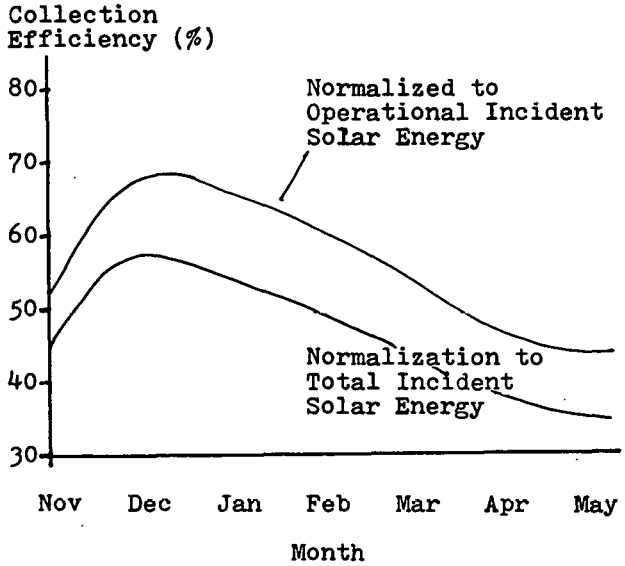


Fig. 6. Average Monthly Collection Efficiency

As illustrated by the monthly average building and storage temperature data presented in Fig. 7, the average temperature of the storage mass was consistantly above the average building temperature thus maintaining the capability to transfer energy from the storage mass to the conditioned space.

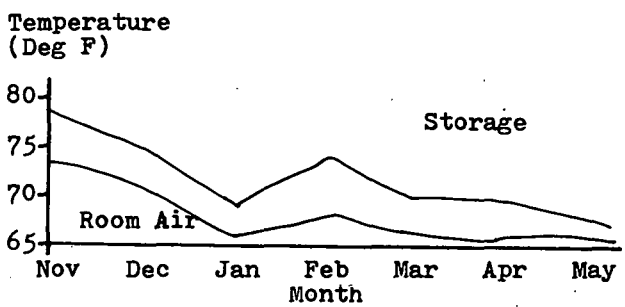


Fig. 7. Monthly Average Building and Storage Temperatures

The next table presents a summary of the thermal performance of the heating system at the SunEarth House for the winter of 1978-79.

Month	Building Load	Fireplace Energy	Space Heat Demand
	all in million BTU		
Nov 78	6.9	.4	4.99
Dec 78	10.15	.14	8.25
Jan 79	9.55	.55	7.35
Feb 79	7.69	.04	6.23
Mar 79	6.03	.23	4.37
Apr 79	4.44	.02	3.03
May 79	3.36	.18	1.71
Season	48.12	1.56	35.93

Month	Internal Heat Gain	Solar Fraction of Load %	Solar Fraction of Demand %	Ave. Bldg. Temp. (Deg F)
Nov 78	1.51	72	100	73
Dec 78	1.76	81	100	71
Jan 79	1.65	81	100	66
Feb 79	1.42	81	100	71
Mar 79	1.43	72	100	70
Apr 79	1.39	68	100	70
May 79	1.47	51	100	70
Season	10.63	74	100	70

Fig. 8. Heating System Thermal Performance Summary

Energy savings and the Beadwall energy of operation are summarized in the following table.

Month	Fireplace Energy		Building Energy Savings	
	Million BTU	Cu.Ft. Gas	Million BTU	Cu.Ft. Gas
Nov 78	.4	476	8.29	9,869
Dec 78	.14	167	13.75	16,369
Jan 79	.55	655	12.26	14.595
Feb 79	.04	47	10.39	12,369
Mar 79	.23	273	7.28	8,667
Apr 79	.02	23	5.05	6,011
May 79	.18	214	2.74	3,262
Season Total	1.56	1855	59.76	71,142

FIG 9. (cont. next page)

Month	Beadwall Operating Energy	
	Million BTU	kwh
Nov 78	.14	41
Dec 78	.18	53
Jan 79	.18	53
Feb 79	.15	44
Mar 79	.12	35
Apr 79	.14	41
May 79	.10	29
*gas assumes 840 BTU per cubic foot		
Season Total	1.01	296

Fig. 9. Energy Savings

A summary of average daily minimum and maximum building temperatures and storage temperatures, appears in the following table.

Month	Building Temperature		Storage Temperature	
	Average Daily		Average Daily	
	Min.	Max.	Min.	Max.
Nov 78	70.5	75.8	74.9	79.2
Dec 78	67.9	73.9	73.1	77.6
Jan 79	62.4	68.8	67.5	72.2
Feb 79	67.9	73.8	73.6	77.7
Mar 79	67.7	72.5	71.8	75.4
Apr 79	68.3	72.4	71.6	74.0
May 79	68.1	71.7	69.8	71.6

Fig. 10. Building Comfort Levels

The data presented in Fig. 11 illustrates the building temperature response on a cloudy day when the outside air temperature varied considerably. Average building temperature varied by only 3 degrees while the outside ambient temperatures varied from a low of 27° in the morning to a high of almost 65° in the afternoon. This illustrates the capability of the storage masses to moderate the building temperature when outside temperature conditions vary considerably.

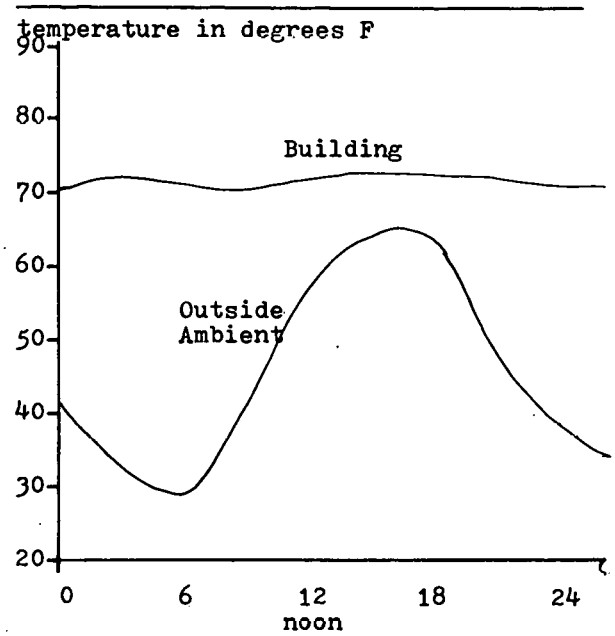


Fig. 11. Building and Outside Ambient Temperatures - May 8, 1979

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HAIL RESISTANCE OF SOLAR COLLECTORS WITH TEMPERED GLASS COVERS

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ABSTRACT

On 30 July 1979, an intense hailstorm passed through Fort Collins, Colorado, leaving estimated property damage of $50 million. Most of the hailstones were of golfball size, but many were as large as a softball. Among 11 solar heated buildings in the main hailpath, 10 were provided with collectors in which tempered glass covers were used. Six of these collectors were not damaged, two suffered one broken cover glass each, and a 700-panel collector on a new county building had seven broken covers. Two covers in a 26-panel collector glazed with ordinary 1/8 inch untempered glass were punctured.

Steel automobile bodies were heavily dented, roofs were penetrated by large hailstones, and greenhouses with glass and fiberglass roofs were demolished, but there was no significant damage to solar collectors in this intense hailstorm.

The paper contains information on the solar installations affected by the storm, photographs of damage, and conclusions on the insigificant risk of hail damage to well-designed solar collectors.

STORM CHARACTERISTICS

During the afternoon of 30 July 1979, severe thunderstorms developed over the Rocky Mountains in southern Wyoming and northern Colorado. These storms moved in a southeasterly direction, one of the cells passing through Fort Collins, Colorado between 4:30 and 5:30 pm.

In the storm cell which passed through Fort Collins, wind and rain were augmented by intense hail in a path two to four miles in width. Duration of the hail at any one location was five to ten minutes. Hail averaging about one inch diameter fell during the first few minutes, followed by a short lull, and then by one or two minutes of two inch to four inch hail. A remarkable feature of the storm was the large average size of the hailstones and, more particularly, the exceptional size of the largest stones. Massive hail damage to structures such as

enhouses had not occurred in Fort Collins since

2. A photograph of representative hailstones is shown in Fig. 1. The weight of the largest stones collected was nearly one pound.

SOLAR INSTALLATIONS IN HAILPATH

Directly in the hail path were 11 solar heating systems, 10 of which are used for space heating and one for commercial laundry drying. Table 1 contains the major features of the solar collectors. All but one collector are glazed with 1/8 inch tempered glass, the exception being glazed with 1/8 inch non-tempered glass. Most of the installations were provided with 3.0 by 6.5 foot panels, but a large system consists of 700 two foot by four foot panels and, on one house, 3.5 by 6.6 foot modules were used. The smallest collector contained 20 panels totaling 390 square feet and the largest contained 700 panels totaling 5600 square feet. All collectors face south at slopes of 32 to 56 degrees, most of the installations and over 80 percent of the total collector surface being at a slope of 45 degrees. Five of the collectors are associated with full-size residential solar heating systems at the Solar Energy Applications Laboratory in the Solar Village at Colorado State University. Figure 2 is a representative sketch map of the portion of the city of Fort Collins through which the storm passed. The map shows the main hailpath and the locations of the 11 solar heating systems.

HAIL EFFECTS ON SOLAR COLLECTORS

Also shown in Table 1 is the observed effect of the hailstorm on the 11 solar collectors. It is seen that among the 11 collectors, seven were undamaged, two lost one collector panel each, one collector with non-tempered glass lost two panels, and seven panels were broken in the large system containing 700 panels. Breakage among the 10 tempered glass installations totaled nine panels out of 956, equivalent to about one percent.

Figures 3 through 7 are photographs of the four installations in which the above minor damage occurred, and of the structures at the CSU Solar Village where there was no damage. The nine broken tempered glass panels showed characteristic complete disintegration, with no large shards or potentially dangerous fragments. Particularly interesting was the clean perforation through two sheets of glass in the non-tempered cover shown in Fig. 5. The extremely high velocity of what apparently was a very large (possibly four inch) hailstone produced a hole characteristic of glass perforations produced

by bullets and other types of projectiles.

Also shown in the table are total costs of repair and replacement in the eight damaged collectors. The figures include the cost of glass panels and commercial installation labor. The total damage was about $800, all covered by insurance. These costs are estimated to represent less than one percent of the value of the solar collector systems.

OTHER HAIL DAMAGE IN FORT COLLINS

Newspaper accounts quote insurance officials estimating $50 million damage caused by the hailstorm. The largest losses were associated with house roofs and automobile tops and hoods. A company underwriting 19% of Fort Collins automobile and home insurnace has reported 9300 claims totaling $7.6 million. Most of the shingled and composition roofs in the hailpath require repair or replacement, and nearly every automobile suffered dented hoods and tops. Estimates of typical house roof repairs fall in the $1000 to $2000 range, whereas damage to large solar collectors, 400 to 600 square feet in area, ranged from zero to $100.

Figure 8 shows nearly complete destruction of glass and "plastic" glazing on greenhouses on the Colorado State University main campus, designated location 12 on the map, Fig. 2. Approximately 50 percent of the glass roof was destroyed and, although visually acceptable, the fiberglass reinforced polyester (FRP) glazing must be replaced because of small fractures and perforations at points of hail impact.

Crop losses in areas near Fort Collins have been estimated at $1.8 million, affecting about 50,000 acres of sugar beets, corn, beans, and small grains, mainly in a four-mile strip extending southeast of the city into agricultural land.

A small child suffered a fatal head injury and an elderly woman sustained a double arm fracture from large hailstones. Twenty other people required emergency treatment, mainly for head lacerations resulting from hail impacts.

CONCLUSIONS

From the evidence obtained from the 30 July hailstorm in Fort Collins, it is concluded that:

(1) The risk of hail damage to commercial solar collectors glazed with 1/8 inch tempered glass of 2.0 to 3.5 feet width, mounted at slopes greater than 30 degrees, is negligibly small.

(2) Hailstones of three to four inches diameter may be expected to fracture no more than five percent of 1/8 inch tempered glass covers of 2.0 to 3.5 foot width in a commercial solar collector array.

(3) The risk of hail damage to roofs of buildings and automobiles greatly exceeds the risk of damage to solar collectors provided with 1/8 inch tempered glass covers.

(4) Appreciable financial loss from hail damage to solar collectors provided with 1/8 inch tempered glass covers appears to require hailstones larger than four inches in diameter.

(5) Fiberglass reinforced polyester (FRP) sheets used as collector glazings appear to be less resistant to hail damage than 1/8 inch tempered glass.

TABLE 1

Building Number (1)	Building Identification	Collector						Hail Damage	
		Manufacturer	Glazings	Slope	Panel Size, ft	# of Panels	Total Area, ft^2	Broken Panels	Repair Cost, $
1	CSU Solar House I	American Heliothermal	1 - Tempered	45	3.5 x 6.6	26	600	0	0
2	CSU Solar House I	Philips Research Lab	1 - Tempered	45	3.5 x 6.2	24	520	0	0
3	CSU Solar House II	Solaron	1 - Tempered	45	3.0 x 6.5	32	624	0	0
4	CSU Solar House III	Chamberlain	1 - Tempered	45	3.0 x 7.0	30	630	0	0
5	CSU Solar House IV	Solaron	2 - Tempered	45	3.0 x 6.5	26 (2)	570 (2)	0	0
6	Pelley Residence	Solaron	2 - Tempered	56	3.0 x 6.5	30	585	1	52
7	Smith Residence	Site-built	2 - Non-temp		2.0 x 4.0	54	432	2	150 (3)
8	City Hall Municipal Bldg	Rocky Mtn	2 - Tempered	45	2.0 x 4.0	700	5600	7	402
9	Poudre Valley Hospital (4)	Rocky Mtn	1 - Tempered	44	2.0 x 5.0	44	440	0	0
10	Splittgerber Residence (5)	Solaron	2 - Tempered	32	2.0 x 6.5	24	468	1	102
11	Buttonwood St. Residence	Solaron	2 - Tempered	32	3.0 x 6.5	20	390	0	0

Notes: (1) Shown in map, Figure 2
(2) Also 26 double tempered sheets over greenhouse area
(3) Estimate
(4) Laundry dryer
(5) Occupant of house measured sample hailstone which fell on ground adjacent to house at 3.25 to 4.25 inch diameter. Another hailstone penetrated a cedar shingle, two layers of #30 roofing felt, and one-half inch plywood, leaving hole in roof. Damage to car parked adjacent to house appraised at $500.

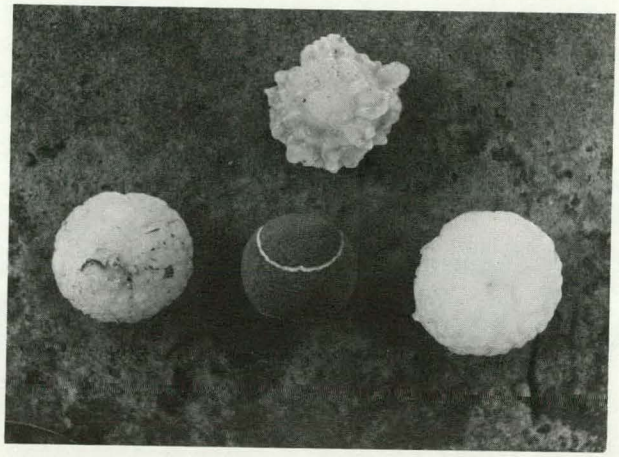


Fig. 1. Hailstones collected at 2700 Trenton Way, Fort Collins, 30 July 1979

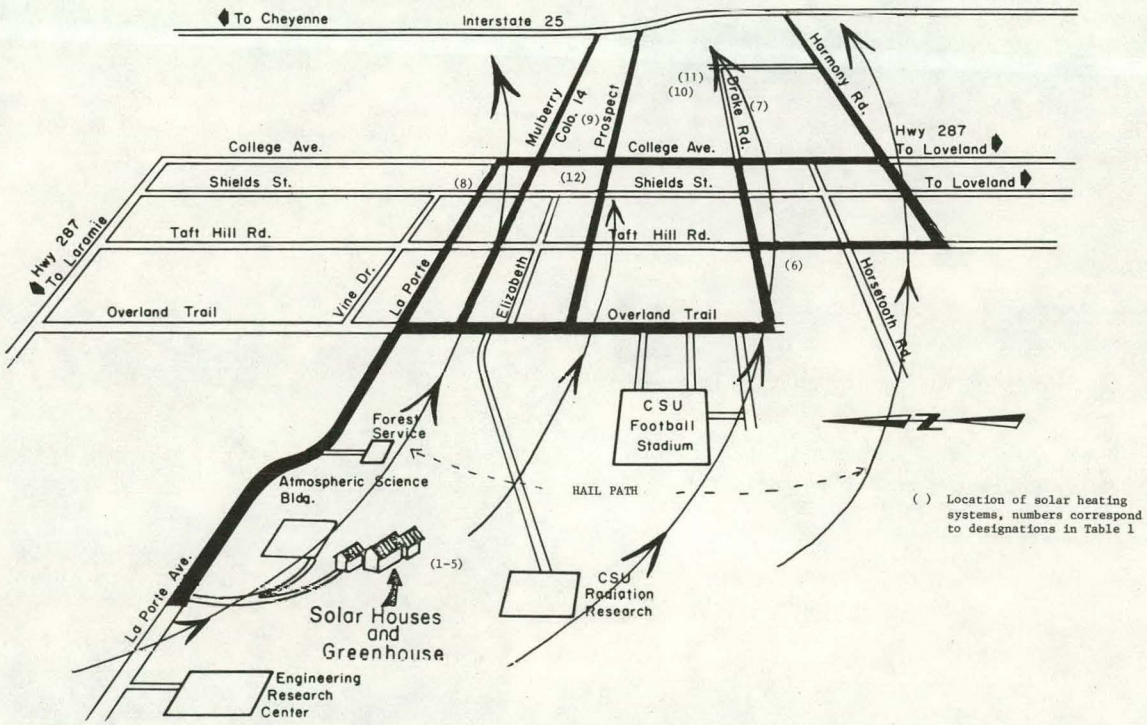


Fig. 2. Map of 30 July 1979 hail path and location of solar heating systems, Fort Collins, Colorado. Numbers correspond to designations in Table 1.

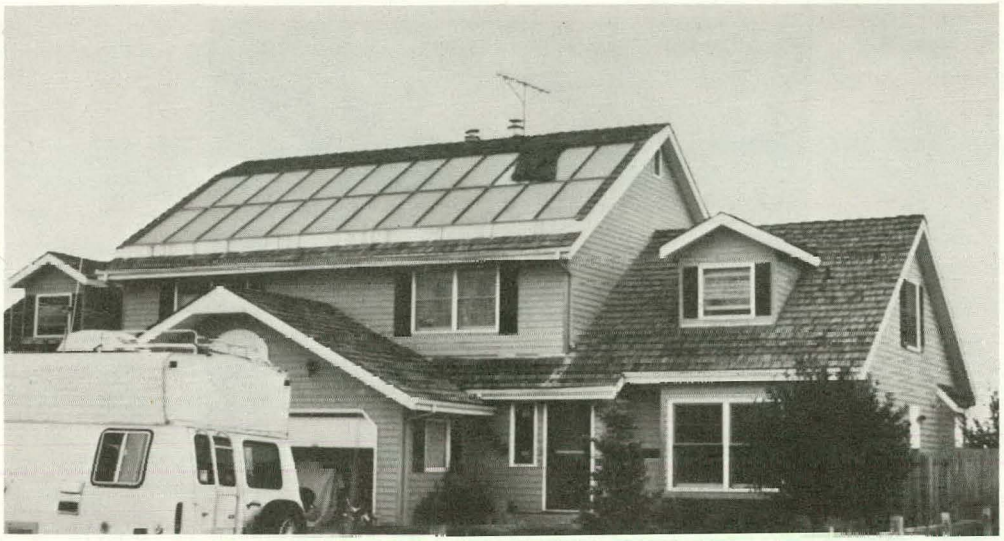


Fig. 3. Solar collector on Splittgerber residence, Fort Collins, after July 30, 1979 hailstorm (one broken panel)



Fig. 4. Solar collector on Pelley residence, Fort Collins, after July 30, 1979 hailstorm (one broken panel)

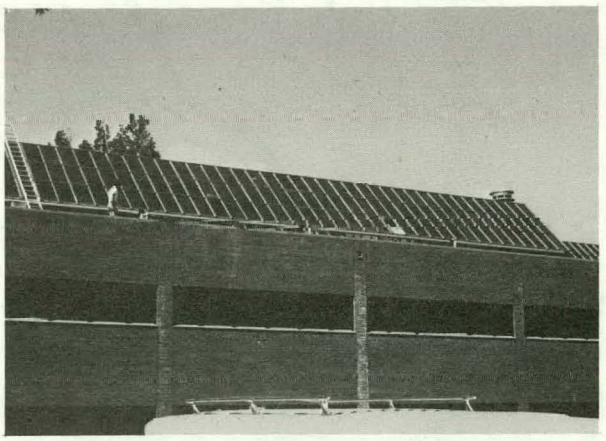
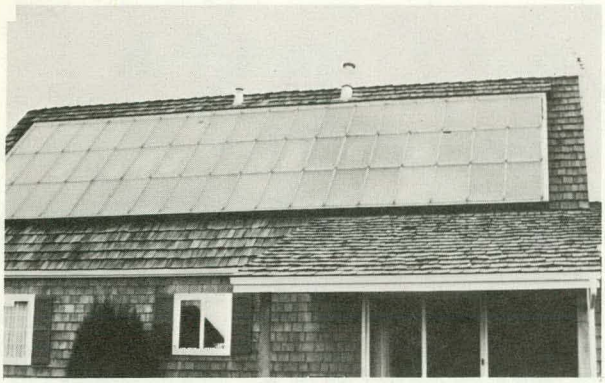


Fig. 5. Solar collector on City Hall Municipal Building, Fort Collins, after July 30, 1979 hailstorm (seven broken panels)



← Fig. 6. Solar collector on Smith residence, Fort Collins, after July 30, 1979 hailstom (two perforated non-tempered panels)

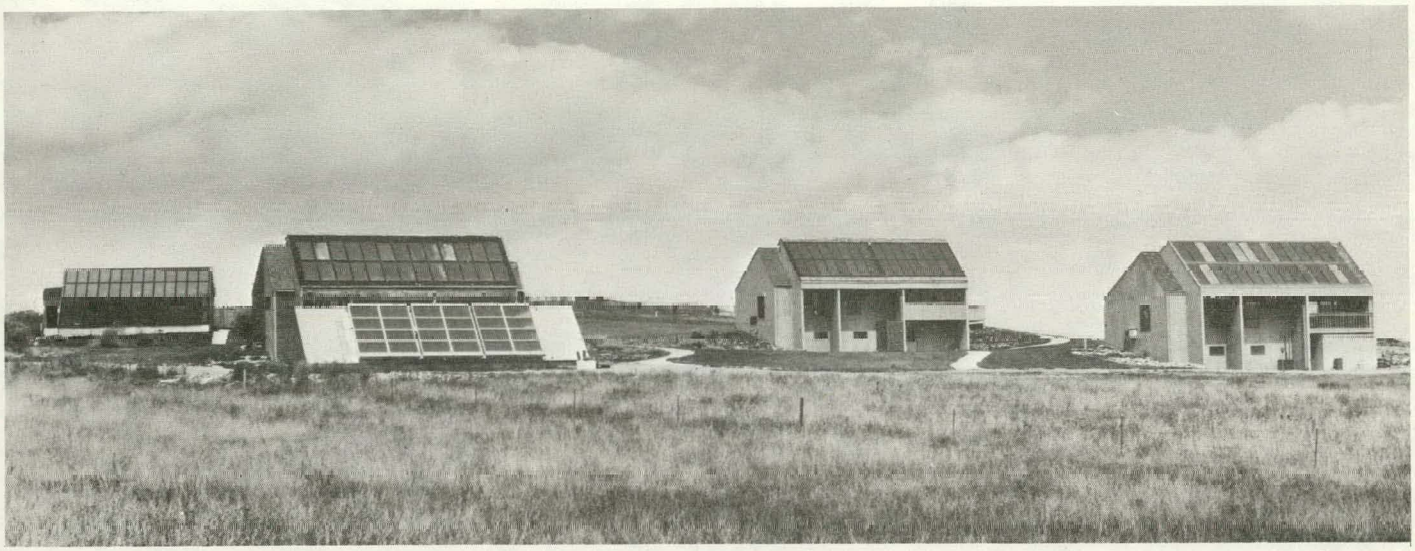
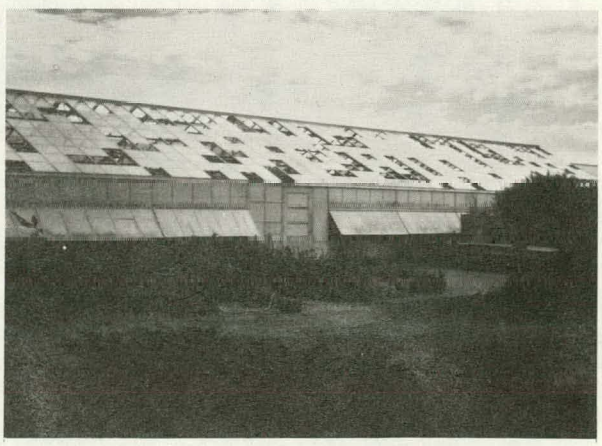


Fig. 7. Five undamaged solar collectors after July 30, 1979 hailstorm at CSU Solar Village

Fig. 8. Hail damage to glass roof → on CSU main campus following storm of July 30, 1979



NOTES

ENERGY USAGE OF AN ENERGY EFFICIENT, SOLAR HEATED HOUSE

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ABSTRACT

An energy efficient, solar assisted house was built in a small midwestern rural community where the sun shines slightly less than half the time in the winter. The premise was an energy efficient, solar assisted house would attract buyers, even in a conservative community.

The house is designed for minimum air conditioning and best use of passive solar heating in a low thermal mass structure. The basic construction and thermal characteristics of the house are presented.

The unoccupied house has been monitored for temperature variations, energy consumption and hours of equipment operation. Building thermal characteristics from these data are reported.

The building heating and cooling requirements are calculated, along with solar contribution. These values are compared with monitored data.

The heating season is reduced from 8 to 5 months; the energy required is calculated to be 32 percent of a HUD-MPS (1977) house. The air conditioner was only operated 40 hours this summer.

INTRODUCTION

Will an energy efficient, solar assisted small house have appeal and acceptance in a conservative, rural community? This is the question which was to be answered by the residence in this paper.

The house is built in a small community 90 miles south-southeast of St. Louis, Missouri. The nature of potential buyers required conventional architecture. The construction used advance methods compared to local construction practice and is designed to produce a year-around energy efficient house. Placement of windows, overhangs, porches, and the attached garage were carefully considered.

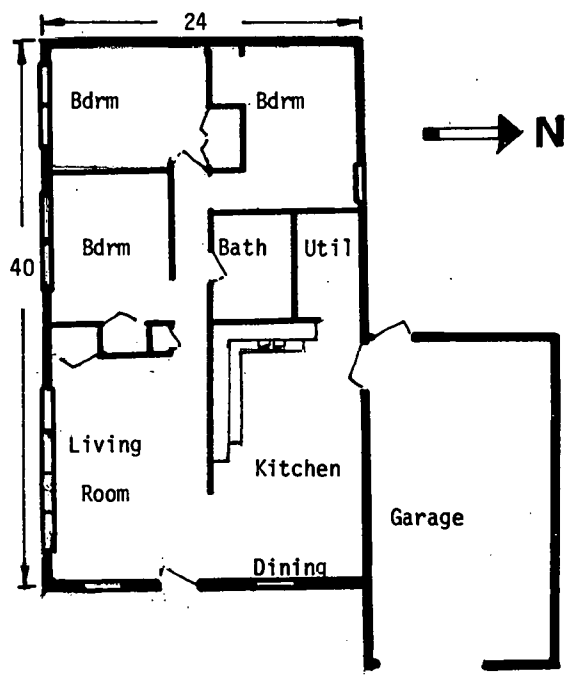


Figure 1. House plan and orientation

CLIMATIC CHARACTERISTICS

The climatic characteristics of the location are:

Winter

[1] Average degree days 2400 °C-day (4320 °F day)

Average percent sunshine 49%

[2] Design temperature -10°C (7°F)

Summer

[2] Design dry bulb temperature 33.8°C (93F)

Design wet bulb temperature 23.9°C (75F)

warm, humid, hazy atmosphere

BUILDING/CHARACTERISTICS

The house is a small (960 ft^2) 89.2 m^2 structure oriented as shown in Fig. 1. The long axis is on an east-west line.

The windows are oriented with maximum exposure on the south wall (69.7% of the total glass) and minimum exposure on east and west walls. To reduce summer air conditioning load, there is no glass on the west wall and one east window is beneath a porch. The north glass is 8 percent of the total glass area. The south-facing glass is 0.496 m (16 inches) below the 0.762 m (30 inch) overhang. These windows are fully shaded at solar noon from April 21 to August 21, and fully sunlit from November through January. The windows occupy 13.5 percent of the total wall area.

The walls are frame construction with 4cm by 20 cm (1.5 x 3.5 inch) studs on 61 cm (24 inch) centers. Electrical outlets and construction joints were sealed to reduce infiltration. Plastic film was used beneath all plaster board.

The ceiling trusses were constructed to permit R 33 insulation in the ceiling, to provide adequate ventilation of the attic and to provide proper shading of south windows by the overhang.

The crawl space is insulated on the inside of the walls with 5.1 cm (2 inches) of styrene foam board insulation. The floor was R 19 insulation between joists, the ductwork has R 11 insulation and the floor of the crawl space is covered with plastic film.

The doors are steel with insulation cores and magnetic weather stripping. The front door also has a storm door.

With this construction, the building has the thermal characteristics shown in Table 1.

TABLE 1

BUILDING THERMAL CHARACTERISTICS

Component	R Value $\frac{m^2\ C}{W}$	R Value $\frac{hr\ ft^2\ F}{Btu}$	UA Value $\frac{W}{°C}$	UA Value $\frac{Btu}{F\ hr}$
Walls	7.21	22.7	121.4	38.5
Windows	1.08	3.4	106.5	33.8
Floor	6.90	21.7	139.3	44.2
Ceiling	8.13	25.6	117.9	37.4
Door (1)	2.05	6.4	11.7	3.7
Door (2)	1.24	3.9	14.5	4.6

These values exceed values necessary to meet two nationally recognized building thermal standards (HUD-MPS and ASHRAE 90-75).

Comparison of some thermal characteristics with recommended values may lead to a better understanding of this house's performance. These comparisons are made in Table 2.

TABLE 2

Factor	House Under Study	Values Recommended by Ref. 3	Values Recommended by Ref. 4.
Solar Window Factor	0.110	0.15-0.24	----------
Overall[a] value	6.0	4.25-5.5	4.5-5.5

a. Value based on floor area

The unoccupied building was monitored for operating time and energy usage of the electric furnace, running time of the blower, temperatures in each room, crawl space temperature, and outside temperature. Incident solar radiation was also monitored. Fig. 2 (located at the end of the text) shows temperatures for a sunny February day.

A careful review of this figure reveals several important features of this building. The low thermal mass results in a response time lag of about three hours to a sudden change in temperature (see change after 8 a.m.).

Measurements of a pyranometer and subsequent calculations indicate that approximately 26.6 kW were available at the windows. Using the measured inside and outside air temperatures, the energy requirement between 8 a.m. and 12 a.m. was 18.1 kWh. This is a solar utilization efficiency of 68 percent.

The solar energy utilization efficiency of this house will be lower than desired because of its low mass construction (5000 Btu/F). The additional furnishings of occupants will slightly raise this value. However, the well insulated structure maintains internal temperature well, as seen in Fig.2.

A study of the temperature distribution within the house for sunny and cloudy days over a month showed the air temperature variation within the house did not exceed 1.7 C (3 F). The need to use the house circulation air fan to equalize the air temperature in the house did not arise.

SOLAR ENERGY AVAILABLE FOR HEATING

The energy in incident solar radiation was measured by an Eppley pyranometer. The energy available at the house windows was calculated using the shade line values of Reference 2. Finally, the energy entering the house was calculated. The values are shown in Table 3.

TABLE 3

SOLAR ENERGY AVAILABLE AT SOUTH WINDOWS

Month	Solar Energy Unshaded Window		Solar Energy South Window		Energy to House	
	$\frac{kWh}{m^2}$	Btu/ft^2	$\frac{kWh}{m^2}$	Btu/ft^2	$\frac{kW}{m^2}$	Btu/ft^2
Feb	6.55	22340	5.24	17870	3.89	13260
March	5.17	17645	2.69	9175	1.96	6700
April	3.98	13580	2.29	7820	1.59	5440
May	4.59	15679	0.92	3140	0.70	2390

HEATING REQUIREMENTS AND ENERGY CONSUMPTION

Calculated Energy Requirements

As constructed, this house has low heating requirements (5778 Btu/°F day, 1693 W/°C day). The energy requirements at design conditions are shown and compared to a HUD-MPS house in Table 4.

TABLE 4

COMPARISON OF CALCULATED ENERGY REQUIREMENTS

	HUD-MPS* HOUSE		HOUSE UNDER STUDY	
	KW	Btu/hr	KW	Btu/hr
HEATING[a] LOAD	10.16	34675	4.59	15650
COOLING[b] LOAD	4.06	13850	3.36	11485

a. 22.2 C (72 F) inside; -13.9C (7 F) outside 15 mph wind,
b. 23.9 C (75 F) inside; 33.9 C (93 F) dbt 23.9 C (75 F) wbt outside; 7½ mph wind.
*Department of Housing and Urban Development-Minimum Property Standards (HUD-MPS) 4900.1 for One and Two Family Dwelling with Revision to May 1977.

The heating load values do not include internal heat generation or solar contributions whereas cooling loads do. Assuming the heating load is proportional to the mean outside temperature (degree-day method), the calculated load for this house is shown in Table 5 (located at the end of the text). Some judgement must be used in estimating the internal heat generation. Reference 2 provides some guidance. In practice the values may vary widely depending upon how occupants live.

The estimated heating requirements for this house indicate significant auxiliary heating requirements only five months of the year instead of the customary eight months for conventional houses. ssive solar heating reduces calculated auxiliary heating demand 21 percent. This solar assisted house requires only 32 percent of the heating energy of the HUD-MPS (77) house.

Measured Heating Energy Requirements

Useful Heating data on this house are only available for April and May of 1979. These data are summarized below.

Month	Solar Energy Available for Heating		Auxiliary Heating Required	
	kWh	MBtu	kWh	MBtu
April	70	0.24	0	0
May	123	0.42	0	0

These limited data compare favorably with auxiliary energy use projections, Table 5, This April was milder than average and resulted in no auxiliary heating requirement.

COOLING REQUIREMENTS AND ENERGY USAGE

The cooling energy requirements of this house were expected to be quite low, based on the design air conditioning load of Table 4. The occupants are a young couple who work every day and who are very frugal with energy. They turned the air conditioner on only in the evening. During June the air conditioner operated only 6.6 hours; July, no hours (air conditioner inoperative 3 weeks); August, 26.7 hours; September, 6 hours.

The effect of the frugal habits of the occupants and the thermal response of the house is shown in Fig. 3. The maximum outside air temperatures are average daily temperatures, with a 12 C (22 F) swing typical. Note that the inside temperature never exceeded 30 C (86 F). This occured when the daily maximum temperature was over 32 C (90 F). More importantly, the inside temperature remained below the daily maximum temperature. This reduced the auxiliary cooling energy when the air conditioner was used.

The frugal life style of the occupants has limited the quantitive information available for analysis, but they have proven the house can be lived in comfortably while using very small amounts of electricity.

SUMMARY

Analysis of monitoring data has shown that this small house can save fossil fuel energy. The structure effectively stores energy for heating. The heating season has been been reduced from just over 8 months to 5 months.

The building is fuel efficient for summer cooling. As operated by frugal occupants, the air conditioner was operated only 40 hours all this summer. This is unusual in this climate.

Calculations indicate this house will use 32% of the heating energy a HUD-MPS (1977) house will use under the same conditions and that solar energy can provide about 30 percent of the annual heating requirement. Further monitoring is planned to verify this and other calculations.

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3. Edward Mazria, Passive Solar Energy Book, Rodale Press, Emmaus, PA., 1979.
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TABLE 5

BUILDING HEATING REQUIREMENTS

Month	Gross Energy Requirement		Internal Heat Generated		Net Heating Requirement		Solar Heat Input[b]		Auxiliary[a] Heat		Solar Load Ratio	Solar Heating Fraction
	kWh	MBtu	kWh	MBtu	kWh	MBtu	kWh	MBtu	kWh	MBtu		
Jan	1627	5.55	185	0.63	1442	4.92	167	1.03	1140	3.89	0.305	0.208
Feb	1304	4.45	167	0.57	1137	3.88	188	0.87	882	3.01	0.450	0.223
Mar	1034	3.53	185	0.63	850	2.90	202	0.69	648	2.21	0.350	0.238
Apr	394	1.35	179	0.61	217	0.74	173	0.59	44	0.15	1.172	0.797
May	132	0.45	185	0.63	0	0	---	----	0	0	-----	-----
Sep	36	0.12	179	0.61	0	0	---	----	0	0	-----	-----
Oct	356	1.21	185	0.63	170	0.58	261	0.89	0	0	-----	-----
Nov	955	3.26	179	0.61	776	2.65	193	1.06	466	1.59	0.588	0.400
Dec	1480	5.05	185	0.63	1295	4.42	158	0.89	1034	3.53	0.296	0.201
Annual	7318		1629		5887							

a. Estimated 68% solar energy utilization and monthly percent sunshine the same as St. Louis.

b. Measured incident solar radiation values Jan. through August. August to Dec. values, Ref. 2 incident solar radiation.

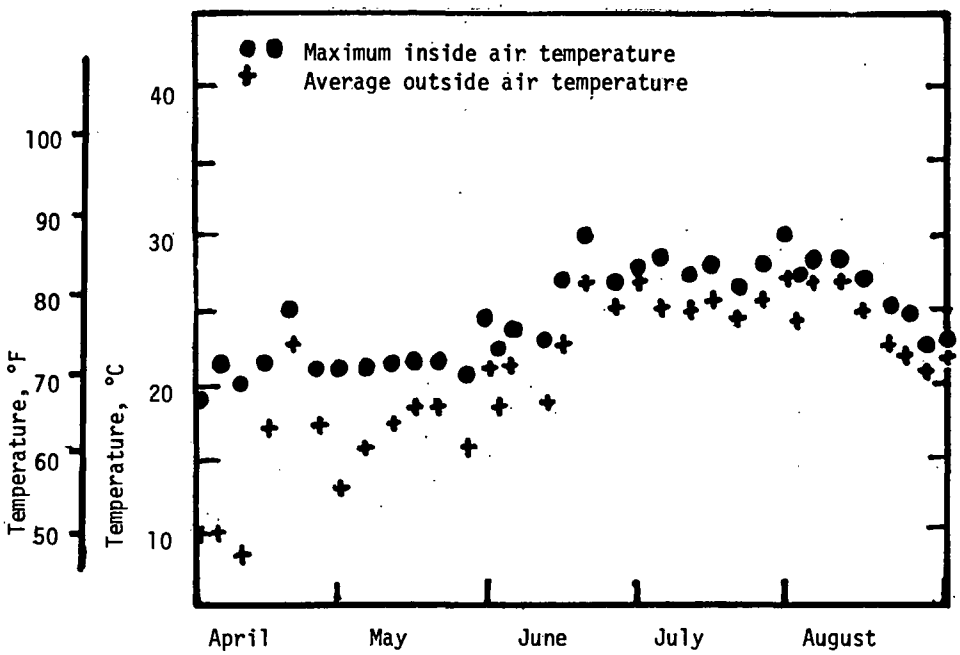


Figure 2. Effect of outside temperature and occupant's life style on the miximum inside air temperature.

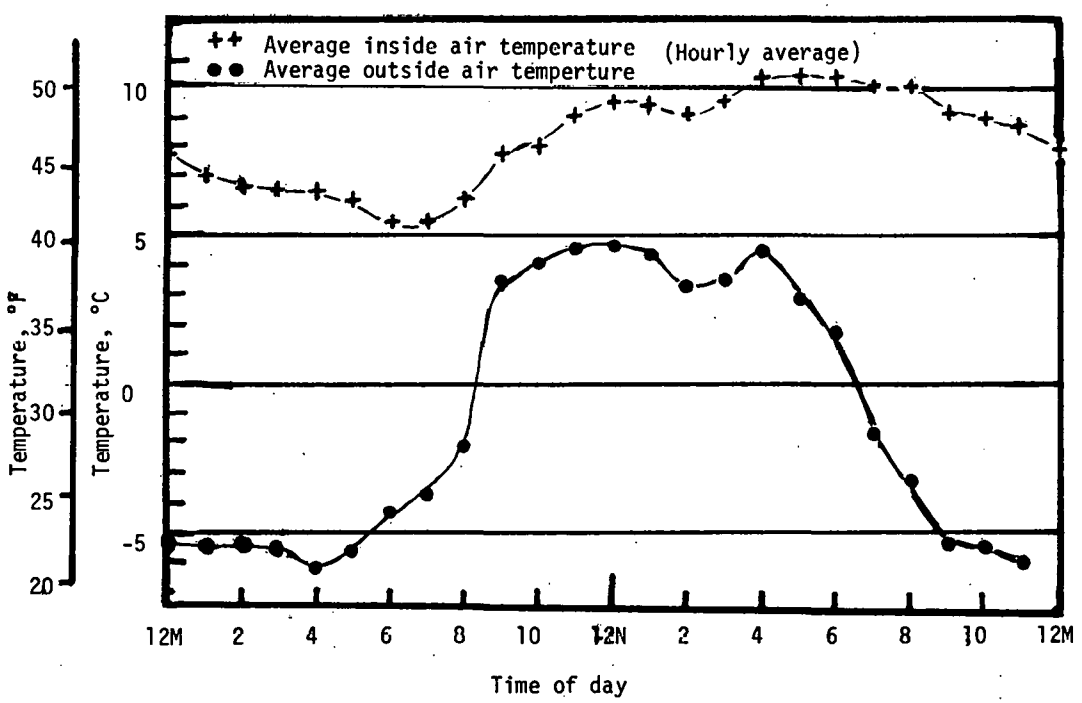


Figure 3. Response of unheated house to outside temperature; sunny day, February

NOTES

COMBINED SOLAR THERMAL/PHOTOVOLTAIC SYSTEM - COOLING SEASON

Dr. Ghazi Darkazalli, Dr. T. J. Lawley, and Mr. Stephen Kugle
The University of Texas at Arlington

ABSTRACT

The University of Texas at Arlington's Solar Energy Research Facility (UTA/SERF), under contract with the Massachusetts Institute of Technology-Lincoln Laboratory, has added a 110 square meter 8 kw_{pk} photovoltaic system to its solar thermal experiment. Electricity from the 8 kw_{pk} photovoltaic system is used to supply the electric demands of the UTA/SERF 145 m^2 (1550 ft^2) test residence. A series water-to-water heat pump is used to chill a cold water storage tank in accordance with the time of day metering rates. Excess photovoltaic energy is fed back to the utility during peak demand hours. A 55 square meter (590 ft^2) flat-plate solar thermal collector system is used for domestic hot water preheat and space heating. Preliminary results from the first year operation of the combined system is presented in this paper.

INTRODUCTION

The southwest region of the United States has a great potential for using solar energy heating and cooling systems for single family residential applications. The investigation of such systems has been under study at The University of Texas at Arlington's Solar Energy Research Facility (UTA/SERF) for the last three years, Fig. 1.

The solar thermal heating and cooling experiment, started in 1976, compares the operational performance and economics of: solar heating to conventional electric heating, solar assisted heat pump cooling to absorption cooling, solar assisted heat pump heating to electric heating of air or water as a back-up for solar heating, and solar domestic hot water preheating to conventional electric systems.

In 1978, a flat-plate photovoltaic system, Fig. 2, was added to SERF's ongoing experiment. Operation of the flat-plate photovoltaic collectors and the flat-plate thermal collectors in conjunction with a heat pump for residential heating and cooling will provide experimental data on the performance of hybrid solar thermal and solar photovoltaic systems.

SYSTEM DESCRIPTION

The UTA Solar Energy Research Facility consists of the 145 square-meter (1550 ft^2) single family residence, concentrating solar collectors; flat-plate solar collectors, flat-plate photovoltaic collectors, thermal storage (water), absorption cooling unit, heat pump, and a heat delivery system. Operation of the absorption unit and the concentrating collectors has been suspended since the fall of 1978.

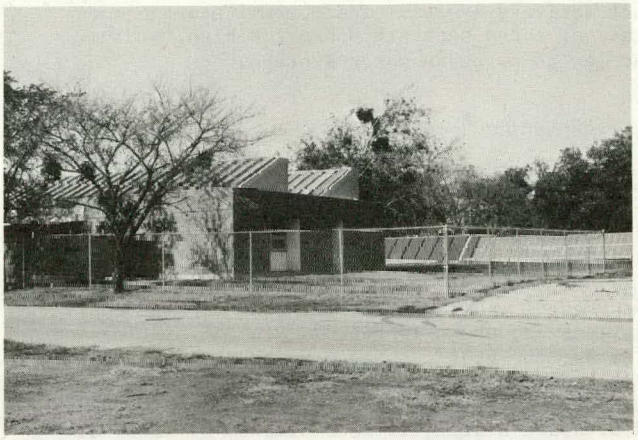


Fig. 1 Solar Residence at the University of Texas at Arlington

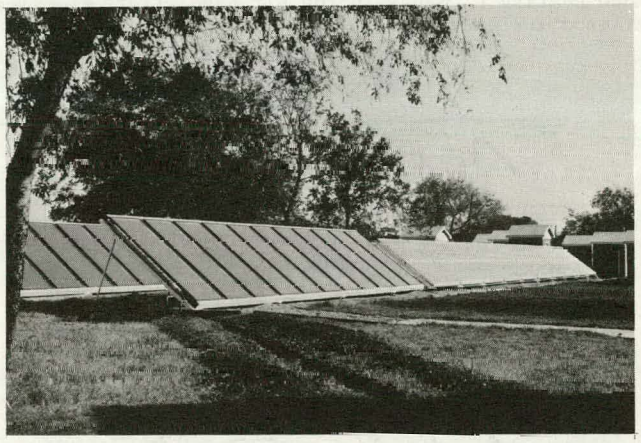


Fig. 2 Separate Photovoltaic and Thermal Collectors

Photovoltaic Subsystem

The 110 m^2, 8 kw DC_{pk} PV array is composed of 240 solar cell modules mounted on 20 adjustable tilt, array support frames. The tilt angle of the collectors is adjusted seasonally between 25 and 38 degrees from the horizontal. The PV Array DC power is converted to 240 VAC by 8 kw, line commutated inverters. The inverter-array ground is isolated from the utility ground by a 15 KVA isolation transformer between the inverter output and the utility [1].

Solar Thermal Subsystem

A flat-plate liquid-solar-collector subsystem is used with the PV subsystem to evaluate direct solar heating and solar-assisted heat-pump heating of the SERF residence. The flat-plate collector subsystem consists of 55 square meters (590 ft^2) of flat-plate collectors, liquid thermal storage, two-speed, 3-ton capacity, heat pump, and a fan/coil duct delivery system. The thermal storage consists of three separate banks for high, medium, and low temperature water storage.

SYSTEM CONFIGURATIONS

In order to determine the optimum combination of solar thermal/photovoltaic and conventional heating and cooling systems, the following procedure is followed.

The Phase I experiment heating and cooling configurations are shown in Fig. 3. In the heating mode, Fig. 3a, the heat pump evaporator is supplied with heat from the 4500 liter thermal storage tank [2]. When the temperature of the stored water exceeds 27°C, space heating is accomplished by water circulation through the air-handling unit. Should the storage water temperature fall below 7°C, resistance heating elements in the air handler are utilized for backup.

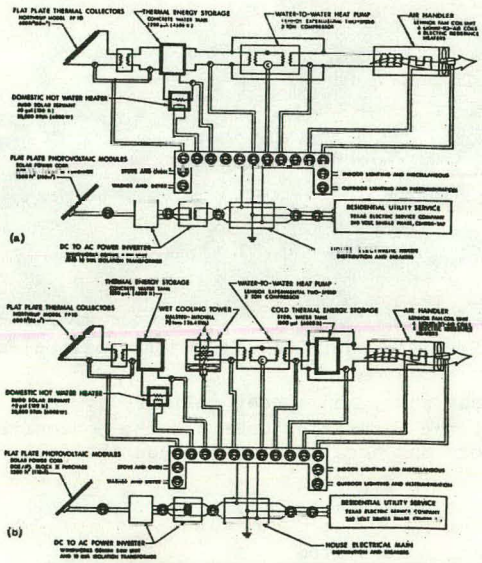


Fig. 3. Residential solar (a) heating (b) cooling system at the University of Texas' solar energy research facility.

The cooling mode configuration is shown in Fig. 3b. A wet cooling tower and a cold thermal-energy storage tank are utilized in addition to the heating components. The cooling tower rejects heat from the heat-pump condenser to ambient air as in any cooling system. The additional storage tank is used to store cold water for cooling purposes.

Cold water storage permits shifting of electric power demands associated with compressor operation to periods when either solar photovoltaic electric power is available or inexpensive off-peak utility power can be purchased.

The system is thoroughly instrumented and all thermal and electrical energy transfer are monitored. Two meters are placed between the house main and the utility so that two-way flow of electricity can be monitored. A detailed description of the system and the system metering is given in reference [1].

SYSTEM OPERATION AND PERFORMANCE

During the first phase of this experiment the system is operated according to the following two modes:

Winter Operation

The winter mode of operation extends from 1 November 1978 to 31 March 1979. During this portion of the experiment, space heating is either by circulating solar heated water from the water storage through the air handler or by supplying electricity to the resistance elements placed in the air handler. The electric resistance heating will be used only if the temperature of the thermal storage is less than 7°C. If the storage temperature is between 7°C and 27°C the heat pump will operate in series with the thermal storage and the air handler.

Results shown in Fig. 4 reflect the winter electrical demands of the heating system. The solar thermal participation is shown in the figure for both the concentrating and flat-plate collectors, the concentrating collector operation ceased when the flat-plate collectors became fully operational in January 1979. The figure shows the heat pump and the solar system operation in the form of the number of days in the month these functions were operational.

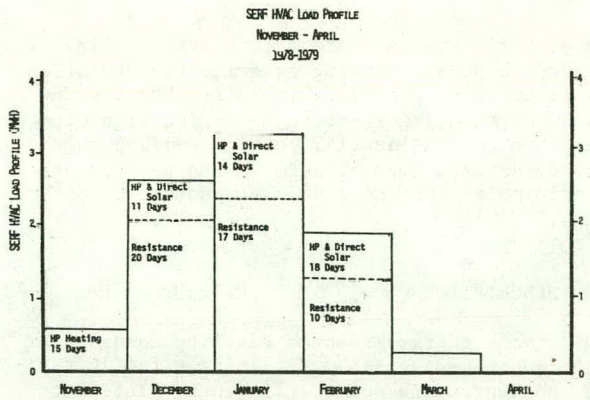


Figure 4. Heat Pump and Solar Participation During Winter Season

ıer Operation

Summer operation of the system extends from 1 April 1979 to 31 October 1979. During this period of the experiment cooling is supplied by circulating water from the chilled-water-storage tank through coils in the air-handler. The circulation of water from chilled storage is done when the temperature of the tank is between 4°C and 16°C. If photovoltaic power is available and the temperature of the storage is greater than 4°C, then the heat pump compressor will be operated to remove heat from the tank. If the chilled water temperature is greater than 16°C, then the compressor will be operated independently of the photovoltaic power availability. When the chilled storage temperature is less than 4°C, photovoltaic power will be fed to the utility.

During summer operation the compressor will run during off-peak hours to chill the cold thermal-storage which in turn reduces the summer peaking of utilities, and supplement the solar system operation [2]. Also, off-peak operation will take advantage of the time-of-use rates proposed by the utility companies, Fig. 5.

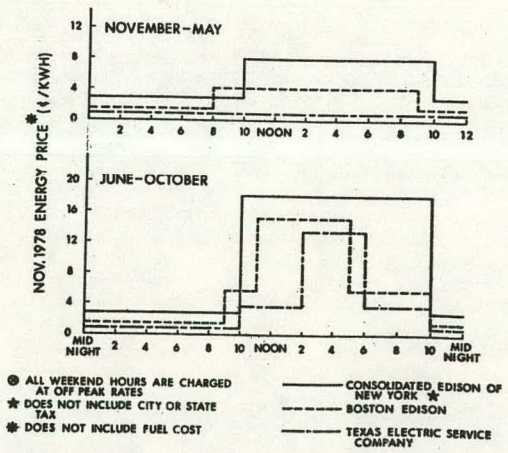


Figure 5. Time-of-use rate schedule for weekdays.

System Performance [3]

The photovoltaic winter and summer participation is shown in Fig. 6 as a percent of the monthly total electrical requirement of the residence. These percentage values were obtained by dividing the net AC photovoltaic power available by the sum of the electric power from utility to house and the photovoltaic to house minus the photovoltaic from house to utility. The figure shows an average of 16 percent photovoltaic participation during the winter months (November-March) and an average of 33 percent during the summer months (April-August), September and October data will be available at the conference.

m in Fig. 7 is the DC and converted AC photo-
taic power generated by the solar system. The shaded area of the figure is the AC photovoltaic power fed to the house main from the isolation transformer. The efficiency of converting the DC power to AC, using the inverter and the transformer, is shown in Fig. 8. The average DC/AC conversion efficiency of the system is 79.5 percent. Also, shown in Fig. 8 is the flat-plate photovoltaic collector efficiency. The drop in the collector efficiency during the summer months is due to the high ambient temperature surrounding the collector.

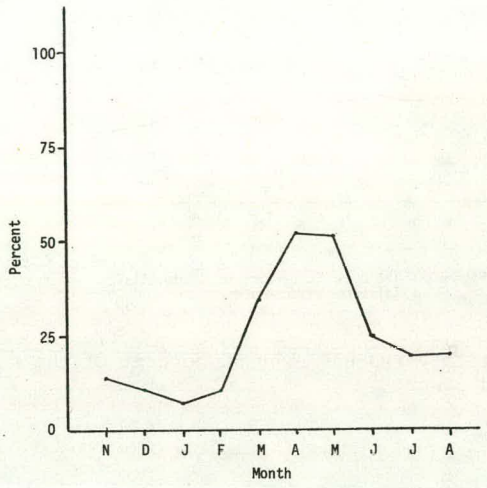


Figure 6. Photovoltaic Winter and Summer Participation

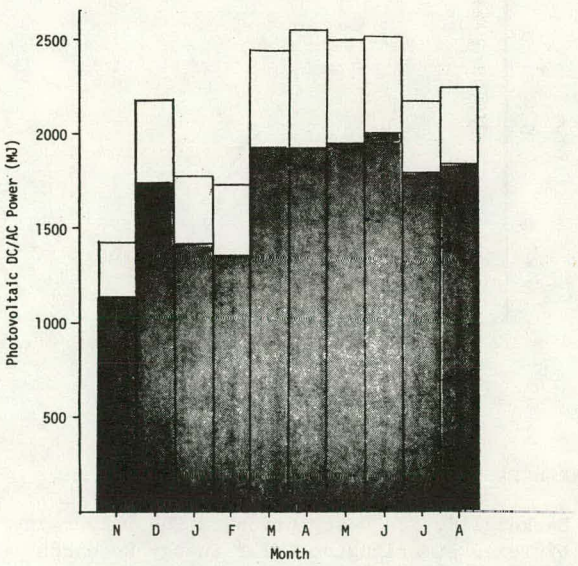


Figure 7. DC/AC Photovoltaic Power Generated (Shaded is AC)

The amount of the AC photovoltaic power fed back to the utility company is given as a percent of the net AC-photovoltaic power generated, Fig. 9. It is noticed that during the months with minimum heating and cooling requirements the photovoltaic feedback

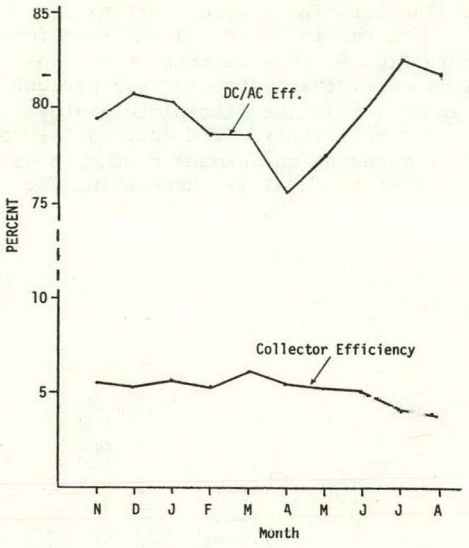


Figure 8. The DC/AC Conversion and Photovoltaic Collector Efficiencies

to the utility reaches over 50 percent of the generated power.

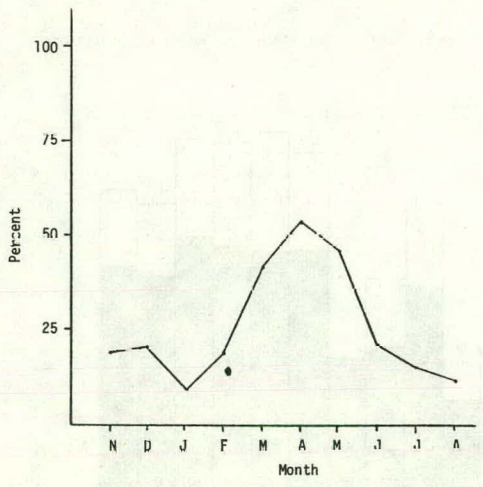


Figure 9. Percent Photovoltaic Feedback

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[2] Kern, E.C., Jr., "Phase-one Experiment Test Plan Solar Photovoltaic/Thermal Residential Experiment at the Solar Energy Research Facility - University of Texas at Arlington," COO-4577-6, MIT/Lincoln Laboratory, March 1979.

[3] Darkazalli, G., and Kugle, S.T., "Progress Report: Solar Photovoltaic/Thermal Residential Experiment at the Solar Energy Research Facility at UTA," Mechanical Engineering Department, University of Texas at Arlington, July 1979.

SOLAR ECONOMICS

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Vice President Sales
Research Products Corporation

ABSTRACT

There is a great need to simplify the economics of solar systems for heating space and service hot water.

Just using life cycle costing information is adding to the confusion. We feel that utilizing the economics from a life cycle costing standpoint and then arriving at a fixed cost for the energy would be a more meaningful method. The energy from an active solar system could then be compared to the cost of the fuels that would be displaced through the use of a solar system.

The fixed cost of solar then can be compared to the fuels available in various parts of our country and at the prices applicable in that particular area. These current fuel costs for home heating range from less than $5.00 per million Btu's to over $20.00 per million Btu's.

These kinds of comparison will be more meaningful and more effective to the consumer.

SOLAR ECONOMICS

The major question in solar is whether or not the system will "pay back." We are not sure just why it was decided that this was an important consideration but seems to be part of most individuals' decision-making process.

If we use this method in deciding on how we would buy other things from a personal standpoint, we would have a very difficult time with our economy. For example, if you were deciding whether you would buy a Volkswagen Rabbit or a Fleetwood Cadillac and used the life cycle cost method, you wouldn't buy either model. A personal automobile just does not pay back ... ever. Also, the same would apply to buying other conveniences (TV, stereo, microwave oven, etc.) and even in deciding on "necessities" for a home such as an extra bath, an extra bedroom, a den, etc. These kinds of things also do not pay back, while being used.

A close look has to be given to a number of variables when deciding on life cycle cost methods
ıd payback periods. Most important is what fuel
; available and what is its cost in addition to
...ıat is its future cost.

Recently, Research Products has taken another look at the solar economics compared to current day costs of fuel, using certain accelerated rates to determine payback periods and also fixed costs for fuels.

ECONOMIC & THERMAL FACTORS

Heat Loss: 40,200 BTU/Hr...Temperature Differential 95°

Collector Area: 372 sq ft...58° Collector Slope

Water Usage: 80 Gal/Day...140° Set Temperature

Cost for Solar: $30/sq ft...$11,160 (less $2,200 Fed. Tax Credit)

Effective Fed. & State Income Tax Rate: 25%

Earning Return on Savings and/or Investments: 8% (after taxes 6%)

Property Tax Rate: 1.5% (of original cost)

For this purpose, we're using a 13% accelerating fuel cost rate increase. We think this is realistic and possibly even conservative based on the rate in the past 8 to 10 years. The chart reflects increasing rates of natural gas in Wisconsin and predictions are the rate will increase more rapidly in the near future.

ACCELERATING FUEL COST RATES

For 7 Years as of January 1, 1979 in Madison, Wisconsin

Natural Gas:	
Residential	14.26%
Commercial	15.93%
Gasoline	13.51%
Heating Oil (Sept. '79 for 5 Years)	19.00%

The economic analysis will be conducted for a period of 20 years. At the end of the 20 years, the solar system is assumed to have a value of 50% of its new value (or $5,580) in today's dollars. This means that if the home were sold at the end of 20 years, the added value to the home would be approximately 160% of the first cost of the system (or $17,890) which is a conservative inflation rate of 6%.

By purchasing the solar heating equipment described above, it is possible to guarantee the price of that portion of the energy consumed. Using the solar cost covered above, energy supplied by the solar system over the next 20 years is supplied at a fixed cost of $10.42 per million Btu's.

USABLE HEATING COSTS per Million Btu's

7¢/KWH Electricity	$20.51 Cost/MMBTU's
6¢/KWH Electricity	$17.58 Cost/MMBTU's
5¢/KWH Electricity	$14.65 Cost/MMBTU's
4¢/KWH Electricity	$11.72 Cost/MMBTU's
3¢/KWH Electricity	$ 8.79 Cost/MMBTU's
70¢/Gal. Fuel Oil	$ 8.33 Cost/MMBTU's
65¢/Gal. Fuel Oil	$ 7.74 Cost/MMBTU's
60¢/Gal. Fuel Oil	$ 7.14 Cost/MMBTU's
36¢/therm Natural Gas	$ 6.00 Cost/MMBTU's
33¢/therm Natural Gas	$ 5.50 Cost/MMBTU's
30¢/therm Natural Gas	$ 5.00 Cost/MMBTU's

Fuel Oil and Natural Gas based on 60% efficiency.

Fuel used over the next 20 years, when calculated for a uniformed heating bill over that time span, would result in $16.15 per million Btu's for natural gas at a current price of 30¢ per therm, $23.05 per million Btu's for oil with the current price of 64¢ per gallon and $40.63 per million Btu's for electricity at its current price of 4.3¢ per kilowatt hour.

On a payback period, solar used with the backup fuel being natural gas would have a discounted rate of return of 10% and a payback period of 20 years. With the backup being fuel oil, this solar system would have a discounted rate of return of 15% and a payback period of 16 years. When electricity is used, the solar unit has a discounted rate of return of 37% with a payback period of nine years.

The next chart will compare the fixed cost of the energy supplied by the solar system with a range of fuel costs (low of 24 cents for natural gas to a high of 7.5 cents for electricity). Each of these is convereted to a cost per million Btu (MMBTU) and then compared to a uniform cost per MMBTU for the next twenty years.

20 YEAR ANALYSIS

Solar Uniform Cost at $30/Sq Ft Per MMBTU	Conventional Fuel Initial Cost Per MMBTU	Conventional Fuel Uniform Cost Per MMBTU
$10.42	$22.00	$70.95
$10.42	$20.00	$64.50
$10.42	$18.00	$58.05
$10.42	$16.00	$51.60
$10.42	$14.00	$45.15
$10.42	$12.00	$38.75
$10.42	$10.00	$32.29
$10.42	$ 8.00	$25.82
$10.42	$ 6.00	$19.37
$10.42	$ 4.00	$12.92

The following chart covers the same cost of fuel range and gives the discounted rate of return and discount payback period.

SOLAR INVESTMENT

Conventional Fuel Uniform Cost Per MMBTU	Discounted Rate of Return	Payback
$70.95	89%	1 yr
$64.50	78%	2½ yrs
$58.05	67%	4 yrs
$51.60	56%	6 yrs
$45.15	46%	7 yrs
$38.75	35%	10 yrs
$32.29	24%	13 yrs
$25.82	17%	17 yrs
$19.37	12%	20 yrs
$12.92	8%	20 yrs

These comparisons were based on the solar system installed at $30 per square foot of collector complete (but does not include the auxiliary heating system). At $35 per square foot, the energy from the solar system would have a uniform cost of $12.70 per MMBTU.

As covered earlier, the figures used included the deduction for the Federal Tax credit. But, it included the cost of a property tax on the complete installed price of the solar system and many states (over 30) have a property tax exemption for that cost. Other states have a tax credit (such as California, Michigan, Wisconsin, etc.) which would reduce the cost for solar further and make the solar investment even more economical.

For additional details, please contact Vern Hellenbrand, Research Products Corporation, P.O. Box 1467, Madison, Wisconsin 53701.

HEAT EXTRACTION FROM A LARGE, SALT-GRADIENT SOLAR POND

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ABSTRACT

The Miamisburg Solar Pond is the largest functional, salt-gradient solar pond in the U.S. It provides low-cost solar energy collection and low temperature heat storage. The heat in the pond is removed by use of a copper-tube heat exchanger which has a surface area of 138 m^2 (1500 ft^2). During the summer of 1979 the initial heat, 39,970 kW·hr (136 million BTU), was removed for heating an outdoor community swimming pool. Based upon the performance thus far, the pond could yield 381,000 kW·hr/yr (1300 million BTU/yr) of useful heat. An economic analysis based upon a 10-yr amortization of the installation costs of the pond indicates that the cost of the heat, 1.8¢/kW·hr ($5.40/million BTU), is already below the current price for heating with fuel oil.

INTRODUCTION

The construction of the largest salt gradient solar pond in the U.S., located in Miamisburg, Ohio, has provided the city of Miamisburg with an alternate energy source for the summer heating of its community outdoor swimming pool [1]. The solar pond concept provides low-cost solar energy collection as well as low temperature heat storage. Several research ponds have demonstrated the potential usefulness of such ponds in the U.S. [2,3]. A large pond, 7000 m^2, near the Dead Sea area in Israel has been used for commercial heating purposes and also was used to demonstrate the generation of electrical power when adapted to a low temperature turbine [4].

The solar pond in the city of Miamisburg is 55.4 m x 36.9 m (180 ft x 120 ft), nearly 2000 m^2 surface area, with sides tapered at an angle of 45° to a depth of approximately 3 m (10 ft). The pond was stabilized against normal convective flow when a salt concentration gradient was established which varied from 0% NaCl for the top convective layer to 18.5% NaCl for the bottom convective layer. Between the two convective layers was formed a nonconvective layer approximately 1 m thick, whose salt concentration increased with depth. The three-layer structure of the pond is clearly seen in the temperature-depth profile of the pond, Fig. 1. The top convective layer, approximately 0.4 m thick, is caused by the action of sunlight and wind. It nearly disappeared when the pond was covered with ice during February. The middle layer of the pond has a temperature, and salt, gradient which increases with depth. The bottom layer of the pond, nearly 1.5 m thick, contains the high temperature water. The seasonal changes in the temperature of the pond are noted in the figure.

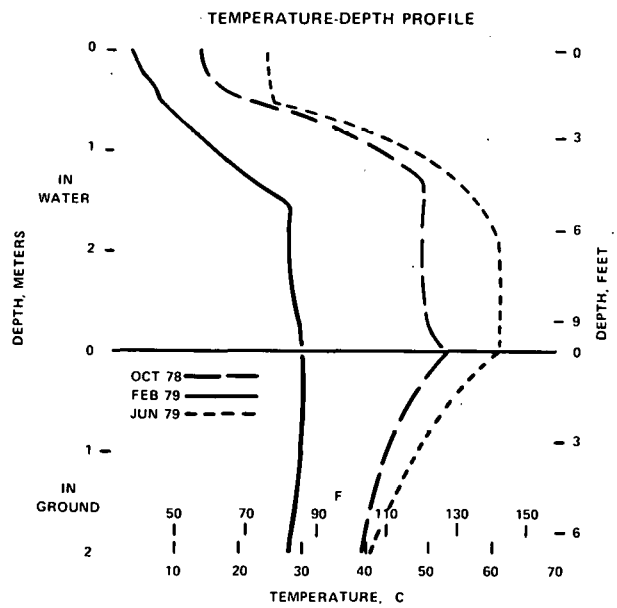


Fig. 1 - Temperature profile as a function of depth in the solar pond water and in the ground beneath the pond for three seasons of the year.

Some of the heat in the pond diffuses into the ground below the pond, as shown in Fig. 1. Part of the heat in the ground does return to the pond during the winter as determined from the shape of the thermal gradient in the ground during February.

*Mound Facility is operated by Monsanto Research Corporation for the U.S. Department of Energy under Contract No. DE-AC04-76-DP00053.

A heat exchanger lies near the interface between the gradient layer and the bottom convective layer for removal of heat from the high temperature storage layer. The heat exchange is made of 25.4 mm (1 in. O.D.) copper tubing having a total effective surface area of 138 m^2 (1500 ft^2), although only half has been used thus far. Water is circulated from the swimming pool through the heat exchanger and returned to the pool whenever the pool temperature drops below 26.7°C (80°F). Only gravity circulation causes water to flow across the heat exchanger.

The performance of the pond is recorded by the use of several instrumental techniques. Numerous temperature sensors measure the temperature of the pond water, the ground below the pond, and the performance of the heat exchanger. One pyranometer measures the total horizontal insolation at the site while another pyranometer scans the pond to determine the solar radiation penetration into the water. An additional meter records the heat delivered by the heat exchanger. All of this information is read and recorded periodically throughout the day by use of a small process computer.

POND PERFORMANCE

The pond was completed in August, 1978 and reached operational temperatures in June, 1979. The first heat extraction from the solar pond occurred during the summer months of 1979, as shown in Table 1. The total heat extracted for use in the swimming pool between June 26 and September 5 was 39,970 kW·hr (136 million BTU). The average heat extraction rate from the pond was 23.06 kW, or 340 W/m^2 of heat exchanger surface. On a particular day of high demand in July, the extraction rate approached 3600 W/m^2 by the heat exhanger for a temperature difference of 12°C between the intake and return water.

The thermal performance of the pond has to be based on an annual cycle because of the long-term heat storage in the pond; consequently, daily variations in solar insolation or heat use have little effect upon total pond performance. The thermal energy balance of the pond is, therefore, the net difference between the annual energy accumulation in the storage layer, which occurs by solar radiation absorbed in the storage water, and the amount of thermal energy removed annually as useful heat or lost by thermal diffusion to the environment.

Table I - HEAT EXTRACTION FROM THE SOLAR POND DURING THE 1979 SUMMER SEASON

Period	Heat Extraction kW·hr	Heat Extraction BTU(million)
Jun 26-30	7,030	24
Jul 01-31	19,340	66
Aug 01-31	12,720	43.4
Sep 01-05	880	3
Total	39,970	136.4

The principal loss mechanism is thermal diffusion ward through the gradient zone caused by the air temperature being lower than the pond temperature. A minor heat-loss path is by thermal diffusion downward into the ground below the pond. Thermal losses do occur at the edges of the pond; however, such losses should be of minor consideration in such a large pond. The net thermal energy balance which results in a temperature change in the storage water is expressed by the relationship [5,6],

$$\frac{C_s\,(T_2-T_1)}{t} = \tau\,(e^{-\mu\lambda z_c})\,I_o - K_w\left(\frac{dT_w}{dz}\right)_{z=z_c} - K_G\left(\frac{dT_G}{dz}\right)_{z=d}\frac{-U}{t} \qquad (1)$$

where C_s = heat capacity of the storage layer ($°C^{-1}$)
T_1 and T_2 = temperature of the storage water at the beginning and end of time period, t.
I_o = incident solar radiation
τ = transmission corrected for reflection
μ_λ = absorption coefficient for wavelength λ (m^{-1})
z = depth of pond
z_c = depth of the storage layer boundary below the pond's surface
K_w and K_G = thermal conductivity of water and ground, respectively
T_w and T_G = temperature of water and ground, respectively
d = total depth of the pond
U = heat removed

Based upon the local solar insolation, and ambient air and ground temperatures, the efficiency of the pond is theoretically predicted [7] to be nearly 20%, which would produce 381,000 kW·hr (1300 million BTU) of useful heat.

A projected utilization of the pond has been determined [1] based upon month-to-month actual performance data accumulated for one year at the pond and evaluated by use of Equation 1. In order to evaluate the solar energy collection in the pond, the term, τ, was corrected for reflection and μ_λ was corrected for refraction during each month of the year. The values of $(e^{-\mu\lambda z_c})$ were evaluated after the solar radiation spectrum was divided into four wavelength groups. In addition, only 90% of τ was used to correct for diffuse sunlight, which is incident upon the pond at such a steep angle that it is not absorbed. When the pond was covered with up to 0.2 m of ice and snow during January and February, a value of τ equal to 0.33 was used.

The evaluations of the heat loss terms (second and third terms, right-hand side of Equation 1) necessitated the determination of the temperature differentials in the water near the top of the storage layer zone, i.e., $(dT_w/dz)_{z=z_c}$, and in the ground immediately below the pond, i.e., $(dT_G/dz)_{z=d}$. Examination of the temperature differential in the water revealed that this term over a period of a month approached the value $(T_p-T_a)/\ell$ where T_a and T_p = temperatures of the air and pond, respective and ℓ = the thickness of the gradient zone. The

n for the heat loss to the ground was evaluated ed upon temperature measurements made at 0 and 0.5 m below the pond.

In order to complete the evaluation of Equation 1, reasonable heat use values, U, for each month were assumed, Table II, based principally upon the required heating demands. The predicted heat extraction for the summer months is 234,000 kW·hr (800 million BTU). The heat extracted in May will be used for the initial heating of the swimming pool which opens in late May. In addition to heating the swimming pool in the summer months, heat extracted from the solar pond can be used to heat the adjacent bathhouse from October through December, without the use of a heatpump. The total heat use predicted for the solar pond indicates that approximately 15% of the incident solar radiation is utilized. The performance of the pond could approach the theoretical value, if, for instance, the pond was used to pre-heat domestic hot water during certain months of the year.

Table II - PROJECTED ANNUAL HEAT USE FROM THE SOLAR POND BASED UPON MEASURED PERFORMANCE DATA

Month	Air Temp. (°F)	Air Temp. (°C)	Insolation (W/m^2)	Heat Use ($kW \cdot hr \times 10^3$)
Jan	22.0	-5.6	61.6	0
Feb	30.9	-0.6	121.6	0
Mar	38.9	3.8	121.3	0
Apr	50.7	10.4	154.3	0
May	61.6	16.4	239.9	88
Jun	70.3	21.3	248.1	88
Jul	73.0	22.8	215.4	0
Aug	71.6	22.0	187.3	32
Sep	64.7	18.2	177.6	26
Oct	53.0	11.7	101.6	3
Nov	46.0	7.8	49.2	25
Dec	35.0	1.7	48.1	19

COST EVALUATION

The results from the Miamisburg Solar Pond indicate that it can be a viable source for low-temperature heat in most areas of the U.S. With proper match between the heat use and the pond parameters, the theoretical efficiency of the pond appears to be obtainable.

The entire pond system was installed by the city for approximately $70,000 and has a projected lifetime greater than 10 years. Based on the predicted full heat extraction from the pond (381,000 kW·hr/yr), the cost of heat is approximately 1.8¢/kW·hr (5.40/MBTU), which is very competitive with other conventional heating systems. This cost is already below the current price for heating with fuel oil. The operational and maintenance expenses appear to be small and consist chiefly of operations to maintain the optical clarity of the water. In this respect, it is similar to the maintenance of large swimming pools.

ACKNOWLEDGEMENTS

The authors gratefully acknowledge the cooperation of Richard S. Bryant, Parks and Recreation Director, City of Miamisburg, who has permitted us to place our instrumentaion at the pond site. Valuable consultations (and engineering support during the construction) with Dr. C. E. Nielsen, Ohio State University, R. P. Bowser, B. E. Pack and W. E. Kesling are also acknowledged. This evaluation study is sponsored by the U. S. Department of Energy, Office of Solar Applications.

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NOTES

LUMSDAINE SOLAR HOUSE

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ABSTRACT

This paper summarizes summer and winter data for the Lumsdaine solar house in Las Cruces, New Mexico. The house has a modified building heat loss coefficient of 0.11 MJ/m^2-D.D.(°C) or 5.4 Btu/D.D.-ft^2 for a total heat loss of 52,800 MJ (50.2 MBtu) per year. At worst, the house has an auxiliary heat requirement equivalent to 1100 kWh/year. Heating is about 75 percent by passive solar (direct gain and atrium) and 20 percent by a hybrid solar roof system. Thermocouple data for the solar roof system in the heating and cooling mode and air and surface temperature data in graphical and tabulated form are presented. Operational experience is also discussed as well as construction problems and solutions. Total energy consumption for the house is around 10,000 kWh/year (refrigerator/freezer = 30%, cooking, lights, appliances = 40%, solar roof system fans and solar water heater = 15%, backup water heater = 15%). Power demand is less than 2 kW except for a peak of about 4 kW during late evening hours; thus this house does not contribute to the summer peak load problems of the utility.

1. DESCRIPTION OF HOUSE AND CLIMATE

The Lumsdaine solar house, with 254 m^2 (2730 ft^2) heated floor area, is located in Las Cruces, New Mexico, on a slightly south-sloping lot on mesa land. December daily minimum mean temperatures are -3.3°C (26°F), June daily maximum mean temperatures 34.2°C (94°F), with the yearly extremes ranging from -23°C (-10°F) to 43°C (109°F). Total average heating degree days equal 1870 (°C) or 3400 (°F); winter winds are from the north at 10 km/hr (6 mph), summer breezes from the southeast at 8 km/hr (5 mph). Average yearly precipitation is 20 cm (8 in.); relative humidity is less than 30 percent in the late afternoon.

The primary solar heating method is passive by direct gain through 19.4 m^2 (182 ft^2) of south-facing windows and 11.7 m^2 (126 ft^2) of sloped skylights. This heat gain is stored in 31 m^3 (1100 ft^3) of interior masonry walls, tile and parquet floors, and a large planter in the atrium. Figure 1 shows a schematic of the solar roof system which contributes about 20 percent of the heating load and can also be used for cooling in the summer. A heat-circulating fireplace is the backup heater, supplying at most 5 percent of the heating load. The house is also equipped with two active solar water heaters, only one of which has a backup (electric). This
ectric water heater is hooked up to a separate
wer meter.

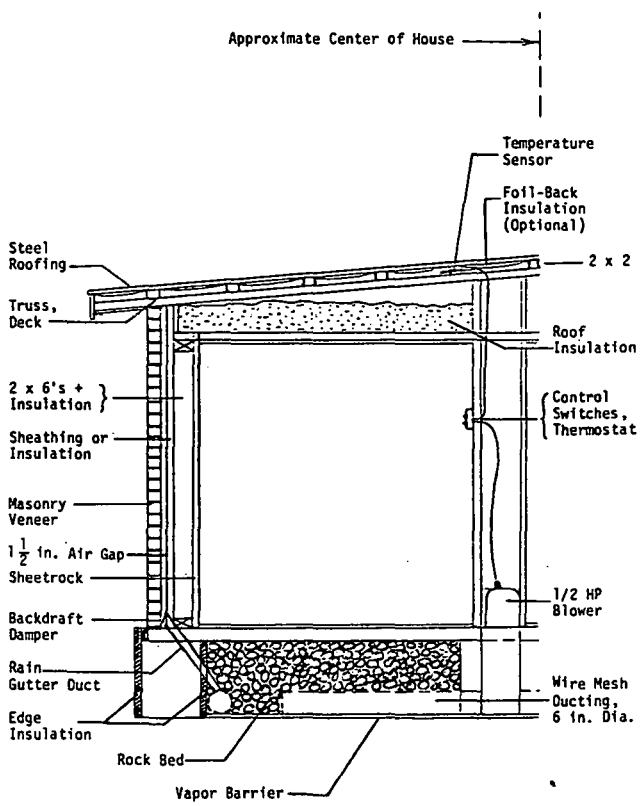


Fig. 1 Schematics of Solar Roof System

2. HEAT LOAD CALCULATIONS

Table 1 summarizes the heat load calculations for the Lumsdaine house, using a simplified calculation method [1]. The total auxiliary load for an average heating season amounts to 3900 MJ (3.7 MBtu). This value compares favorably with computer calculations and operational experience. The New Mexico Energy Extension Service, through the New Mexico Energy Institute at the University of New Mexico, has analyzed the Lumsdaine house with its modified home energy audit computer program (using an area-averaged value of 2947 heating degree days). The computer has calculated an auxiliary heat requirement of 3.55 MBtu per season and a useful solar heat gain of 35 MBtu (compared with a total of 30 MBtu solar heating contribution for the hand calculations). According to the computer program, if

Table 1 Heat Load Calculations

BUILDING SKIN HEAT LOSS: U x A

		Btu/hr-°F	W/°C	%
Ext. Walls,	2075x0.04 =	83	44	20
Doors	60x0.5 =	30		
Windows	323x0.4 =	129	216 114	51
Skylights	126x0.4 =	50		
Skylights	11x0.6 =	7		
Roof	2656x0.02 =	53	28	13
Slab Edge (R = 7.1)	=	67	36	16
TOTAL		419	222	100

INFILTRATION: V x C_a x ACH
24,570 ft^3 x 0.016 Btu/ft^3-°F x 0.5/hr
= 197 Btu/hr-°F or 104 W/°C

MODIFIED BUILDING HEAT LOSS COEFFICIENT C_B:
24(419 + 197) = 14,800 Btu/D.D. = 28.2 MJ/D.D.(°C)
C_B/A = 5.4 Btu/D.D.-ft^2 or 0.11 MJ/m^2-D.D.(°C)

AUXILIARY HEAT LOAD PROFILE:

Month	D.D.	Gross Load	Int. Heat	Net Load	Solar Gain	SHF	Aux. Load
O	150	2.2	2.4	-	-	1.00	0
N	490	7.2	2.5	4.7	11.4	0.98	0.1
D	750	11.1	2.5	8.6	10.8	0.82	1.5
J	780	11.5	2.4	9.1	11.3	0.81	1.7
F	570	8.4	2.5	5.9	10.9	0.93	0.4
M	460	6.8	2.4	4.4	11.6	0.99	0
A	200	3.0	2.4	0.6	-	1.00	0
TOTAL, MBtu		50.2					3.7
TOTAL, MJ		52,800					3900

the house temperature is allowed to float to 63°F before auxiliary heat is supplied, only 1.1 MBtu of energy will be required from the backup heater in January, instead of 1.7 MBtu for 65°F day, 55°F night input [2]. Actual auxiliary requirements were found to be lower because of the heat storage and time lag properties of the internal masonry and the zoned plan of the house (the bedroom wing remains unheated when the fireplace is in use). Last winter, air temperatures never fell below 15.5°C (60°F) anywhere in the house. Thus this house has an auxiliary heat requirement of at most 2.3 Wh/m^2-D.D. (°C) or 0.4 Btu/ft^2-D.D. Actually, no fossil fuels will be used to provide this heat, since only scrap wood, newspaper logs and desert brush will be burned in the fireplace.

3. FUEL SAVINGS CALCULATIONS

Table 2 gives the energy savings in kWh when the performance of the Lumsdaine house is compared with an equivalent reference building [1]. By definition, the reference building has a square floor plan of equal size as the solar design, with 10 percent window area equally distributed; U-values are New Mexico Energy Conservation Code minimum specifications. Only heat savings are given; the Lumsdaine house also saves about 2 MBtu/month in domestic water heating and about 1000 kWh for summer cooling; more importantly, the power used for cooling (for the blowers of the solar roof system) is an off-peak load.

Table 2 Heat Savings Calculations

Month	Ref.Bldg. Heat Load MBtu	Pass.Bldg. Aux.Load MBtu	Energy Saved MBtu	Solar System Power Use kWh	Net kWh	Saved kWh
N	9	0.1	8.9	2610	40	2570
D	15	1.5	13.5	3960	60	3900
J	17	1.7	15.3	4480	60	4420
F	11.5	0.4	11.1	3260	60	3200
M	9	0	9.0	2640	60	2580
A	2.5	0	2.5	730	0	730
TOTAL/YEAR						17,400
Aux. Load of Passive Bldg. (Waste Products)						1,100
TOTAL SAVINGS PER YEAR FOR HEATING						18,500

With present energy costs here in Las Cruces, the annual savings amount to

DHW: 7000 kWh @ 2.7 ¢/kWh =	$189.00
Heating: 18,500 kWh @ 5.5 ¢/kWh =	$1017.50
Cooling (Net): 1000 kWh @ 5.5 ¢/kWh =	55.00
TOTAL	$1261.50

4. INSTRUMENTATION AND OPERATIONAL DATA

4.1. Thermocouple Data (Solar Roof System)

In order to evaluate the effectiveness of the solar roof system, the house has been wired with a number of thermocouples in the rockbed under the slab (Fig. 2). Additional thermocouples are located at the inlet to the blower, in the return air space just below the roof (south wall), above the insulation under the garage roof, and at the center of the roof insulation over the living room. A strip chart recorder was available in August 1978 (during the start-up period) from the Mechanical Engineering Department at New Mexico State University. Figure 3 shows a typical daily thermocouple profile together with ambient and floor-level air temperature. Outside the solar roof system operating during cool night hours, there is no other cooling system in the house except natural ventilation.

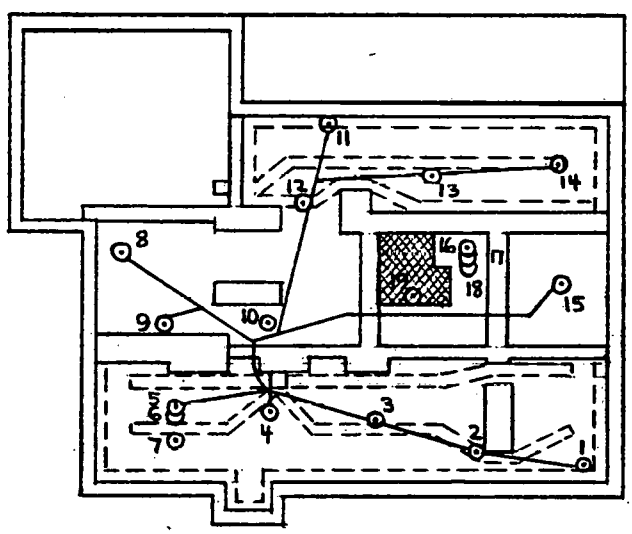


Fig. 2 Thermocouple Network Under the Slab (dott lines indicate wiremesh ducts).

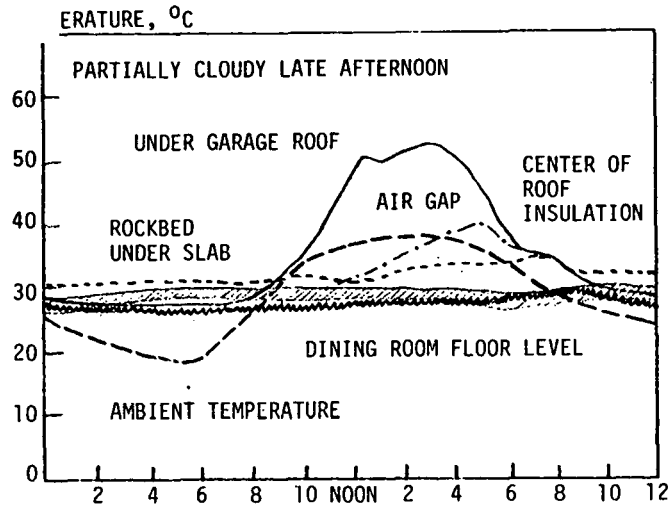


Fig. 3 Thermocouple Data, 18 August 1978

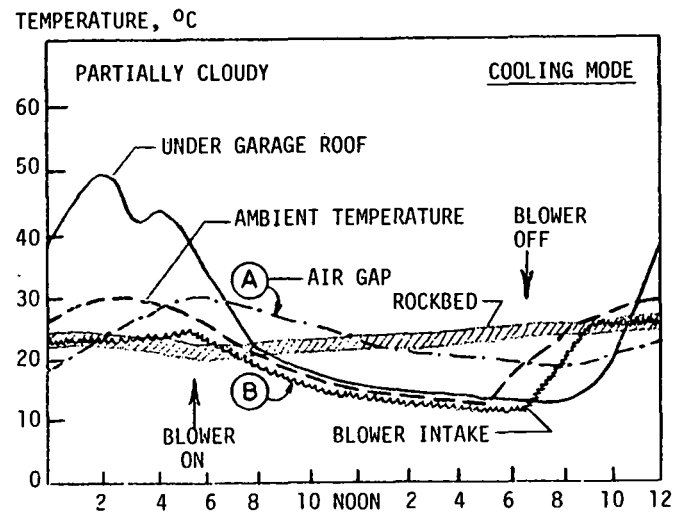


Fig. 4 Thermocouple Data, 19/20 September 1979

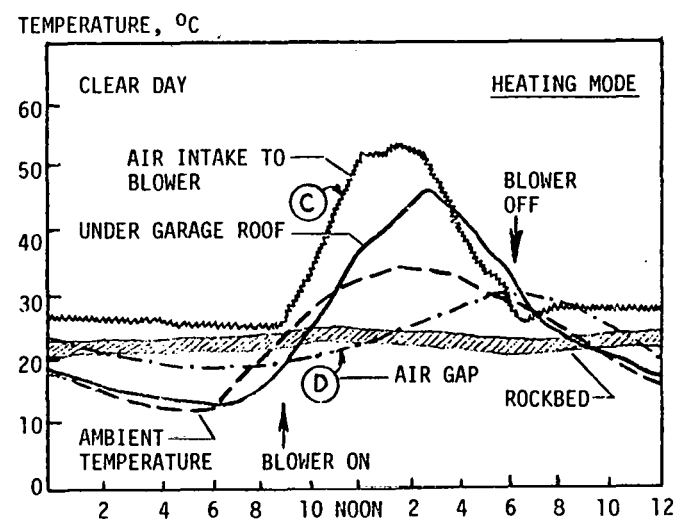


Fig. 5 Thermocouple Data, 21 September 1979

Figure 4 shows a thermocouple temperature profile in the cooling mode. The difference between curves A and B while the blower is operating represents heat removed from the rockbed and dissipated through the roof. Conversely, Fig. 5 illustrates operation in the heating mode, where the difference between curves C and D indicates heat added to the rockbed. The air gap temperature curve increases in the late afternoon due to heat being received through the south-facing wall. Temperatures under the dark steel roof are considerably higher than those under the grey built-up garage roof; this difference is especially noticeable in winter when the garage roof system will frequently not cycle on at all during very cold, sunny days. The fall 1979 data was recorded with a strip chart recorder on loan from the New Mexico Solar Energy Institute.

The thermocouple data and operational experience have shown that there is a time lag of several days in increasing or decreasing slab temperature when the roof system is in operation. On the other hand, the rockbed has a storage capacity of about three weeks; thus the system is turned off in spring and fall and activated a couple of weeks before cold or hot weather is normally expected.

4.2. Air and Surface Temperatures (Passive System)

Since a strip chart recorder was not available during the winter of 1978/79 and through early summer 1979, temperature measurements inside the Lumsdaine house were made with mercury thermometers and a portable infrared thermometer (to measure surface temperatures); these instruments were borrowed from the Physical Science Laboratory at New Mexico State University. The accuracy of these measurements is estimated at ±1°F. Table 3 shows air temperatures inside the house in December 1978 under a variety of weather conditions. As shown in Fig. 6, maximum daily air temperature swing in December 1978 was 7°C (12°F); the largest temperature drop (lows) over a 24-hour span was 2°C (4°F). Table 4 gives surface temperatures inside the house on three days during that same period.

Table 3 Winter Air Temperatures in Lumsdaine House

Location	Temperature, °F					
	A	B	C	D	E	F
Study Table	80	66	--	73	67	68
Dining Floor	69	68	63	71	68	68
Dining Table	72	63	67	71	65	67
Kitchen N. Sink	67	60	65	69	64	65
Hall Shelf	66	64	64	71	68	68
Atrium Planter	71	65	64	68	64	66
North Beds	65	60	65	66	62	66
MBR Chest	--	64	66	71	64	65
Baths Floor	64	62	64	--	61	63
LR Sofa	--	64*	69**	71	68	67
South Ambient	42	30	--	40	46	45

A = 12 Dec. 1978, 12-1 p.m., hazy
B = 13 Dec. 1978, 7-8 a.m., overcast
C = 13 Dec. 1978, 8-9 p.m., overcast
D = 14 Dec. 1978, 6-7 p.m., clear all day
E = 18 Dec. 1978, 10-11 a.m., 3rd day rainy
F = 31 Dec. 1978, 2-3 p.m., 4th day cloudy

*3°F rise (to 67°F) after fireplace used one hour.
**Reading after fireplace in use two hours.

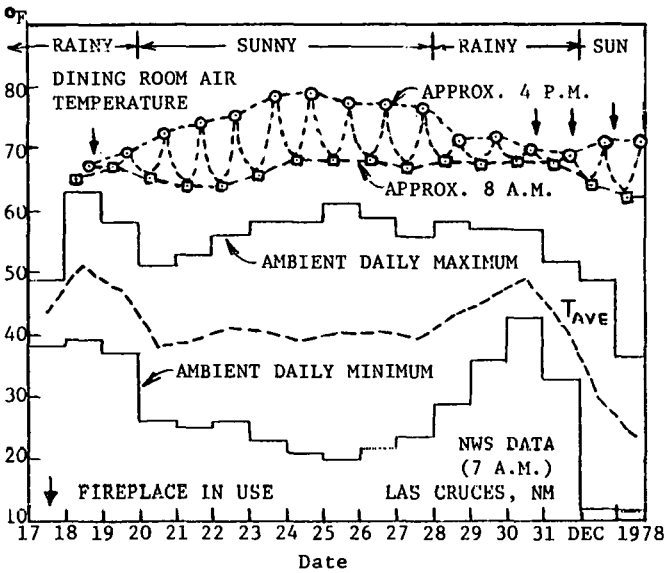


Fig. 6 Daily Air Temperature Fluctuation in Dining Room (maxima and minima), December 1978

Table 4 Typical Winter Surface Temperatures, °F

13 Dec. 1978, 8 p.m.
(Overcast day, fireplace on since 6 p.m.)
Dining: 64-66 Walls, 64 Skylid, 58 Clearstory, 68 Ceiling Top Beam
Kitchen: 61-63 Walls, 61 Shutter
Atrium: 62 Exterior Wall, 63-64 Interior Walls, 62 Planter, 63 Floor, 58 Clearstory with Storm Glass, 48 Clearstory without Storm Glass
E Bedroom: 65 Masonry, 64 other Surfaces
N Bedroom: 63 Floor, N Wall, 64 other Surfaces
Bathrooms: 62 Floor, 64 Walls
Living Room: 65 Floor, 67-71 Masonry, 67 other Walls
Laundry: 62 all Surfaces

20 Dec. 1970, 9 a.m.
(Sunny, Skylids opened at 8:30 a.m.)
Atrium: 62 Shade, 66 Sun, 61 Floor
Living Room: 65 Shade, 70-80 Sheetrock, 65-68 Masonry
5 p.m., Skylids Close
Dining Room: 71 Sheetrock, 74 Masonry, 68 Floor
Living Room: 76 Sheetrock, 74-78 Masonry, 74 Floor
Study: 76 Sheetrock, 74 Masonry, 74 Floor
MBR: 76 Sheetrock, 75 Masonry (73 Bathroom-side), 74 Floor
Atrium: 73 Sheetrock, 69-76 Masonry, 69 Floor
N Bedrooms: 67 Sheetrock, 68 Masonry, 65 Floor
Hall: 70 Sheetrock

21 Dec. 1978, 7:30 a.m. (Clear, 26°F Ambient)
Kitchen: 57 Exterior Wall, 62 Masonry, 63 Floor
Dining Room: 62 Sheetrock, 65 Masonry, 64 Floor
Living Room: 60 Exterior Walls, 64 Interior Walls, 62-64 Masonry, 64 Floor, 68 Fireplace Wall
Study: 64 all Interior Surfaces
MBR: 63 all Surfaces (including Bath)
Hall and Atrium: 64 all Surfaces

Table 5 Surface and Air Temperatures in Lumsdaine House, 14-15 June 1979, in °F

Location	7:40 a.m.	9:40 a.m.	4:00 p.m.	10:50 a.m.
AIR TEMPERATURES				
Ambient	63	79	92	83
Atrium	73	75	76	77
MBR	71	73	75	76
Dining Room	73	74	81	79
Hall	73	74	78	77
SURFACE TEMPERATURES				
Dining Floor	72	73	75	75
Hall Floor	74	74	77	75
Atrium Floor	73	73	74	75
Study Floor	71	71	72	74
NE BR Floor	74	74	74	76
Dining Wall*	75/80	71/76	76/86	75/81
Atrium Wall*	75/80	71/76	72/82	76/80

*Stratification between 2 ft/11 ft levels.

Table 6 Surface and Air Temperatures in Lumsdaine House, 14 July 1979, in °F

Location	7:00 a.m.	9:00 a.m.	1:30 p.m.	6:30 p.m.
AIR TEMPERATURES				
Ambient	75	86	102	96
MBR	77	79	81	82
Atrium	80	80	82	83
Dining Room	80	80	84	86
SURFACE TEMPERATURES				
Study Floor	79	79	80	83
Dining Floor	80	80	81	84
Kitchen Floor	80	80	80	84
Atrium Floor	80	80	78	83
Hall Floor	82	82	80	84
Dining Wall*	80/84	80/84	80/85	85/93
Atrium Wall*	81/85	82/85	80/88	86/94
12 ft Ceiling	80	81	95-100 in spots	

*Stratification between 2 ft/11 ft levels.

Tables 5 and 6 give typical summer temperature data (in degrees Fahrenheit) as measured with mercury thermometers and the portable infrared thermometer. July 14, 1979 turned out to be the record high for the year, with 105°F or 40.5°C.

4.3. Discussion of Data and Operational Experience

Because wall and floor temperatures were very even, with very little stratification occuring, the house never actually felt "cold"; even when air temperatures fell to the lower 60's (°F), the house seemed to be several degrees warmer than that. Humidity levels in the house during the winter ranged from 40 to 50 percent on most days. Since there is no air moving through the living space, and because of good insulation and tight windows, no drafts are experienced. The fireplace was completely adequate in providing all necessary auxiliary heat. It was used once before dinner, twice in the early morning; otherwise normal operation was 1-2 hours after dinner perhaps a total of twenty times for the entire winter (which was colder than average).

The temperature data provided some interesting information. There was a surprising difference in temperature rise behind screened and unscreened windows (thus window screens were removed from mid-December to mid-February). A low reading on a clearstory window showed that the builder had left out a storm sash. Cold spots were found in the kitchen along the north wall baseboard, around the sink drainpipe and especially around the back door, even though it is equipped with a storm door. Since the covered patio along the north side of the house will be glazed-in soon in order to gain extra storage space, the heat losses through the kitchen wall will be reduced considerably. In the summer, stratification is quite pronounced, especially in the center part of the house (dining room and atrium) which has a 12 ft (4 m) ceiling. The operable clearstory windows proved adequate to ventilate this heat buildup in the late afternoon. The insulated Skylids work well to keep heat from entering through the skylights. The temperature on a sunny summer day on the room-side of the Skylid was 80°F (27°C); on the glazing side it was 125°C (52°C). The hot spots along some sections of the high ceiling indicate that there must be gaps in the insulation. The thermocouple data indicated that a lower fan speed would increase the efficiency of the solar roof system in the cooling mode. (Additional data will have to be gathered in the coming winter to determine the optimum fan speed for the heating mode.)

In the summer, even air temperatures in the low 80's are not perceived as uncomfortable due to the low humidity in this desert climate. However, because the dining/kitchen area was warmer than the remainder of the house due to large internal heat sources, a small (2800 cfm or 100 m^3/min) evaporative cooler was temporarily installed in the clearstory window above the dining area on 15 July 1979 and subsequently run an average of 2 to 3 hours a day for three weeks. Its contribution to comfort was minimal since the cooling effect was counterbalanced by an increase in humidity (and perhaps even worse, a terrible increase in noise level). It is planned to use ceiling or oscillating fans next summer if necessary to provide some air movement where needed during very hot days.

Night ventilation was not used to its fullest potential because of allergy problems. Strangely enough, the feeling of comfort in the house also depended on what conditions were like elsewhere. While the office was at the recommended 78°F, the house felt fine; when temperatures at work were lowered to a chilly 55°F, the house began to feel hotter than it did before. Overall, the house is delightful to live in; window shutters and operable windows provide a sufficient amount of temperature and lighting control with very little time required for operation.

5. CONSTRUCTION PROBLEMS AND SOLUTIONS

The house was built by a contractor without previous solar experience. Actual construction time was nine months, more than twice the original estimate. One of the major problems that occured was in scheduling the different subcontractors. This house was unusual because it needed masonry work during several stages. Particularly the framing crew found it very difficult to work while the masons were in their way. Unfortunately, the masonry work (especially on the interior walls) left quite a bit to be desired as to quality. Doorways ended up being too low, courses on adjoining walls did not match, light switches and electrical outlets were left out, etc. One entire wall had to be taken down and rebuilt because it was not lined up correctly. The building permit was held up for one week because the unfamiliar casement wood-frame windows had to be checked for fire code compliance. The masons were very reluctant to install the fresh-air duct to the fireplace. As specified on the plans, the chimney was to be 60 cm by 60 cm (2 ft by 2 ft); it was built 2 m (6 ft) wide and now shades the solar collectors of the water heating systems during winter afternoons. The perimeter insulation at the slab edge was almost left out (it was left out between the laundry room and the garage). There were also some problems with roof leaks. However, none of these problems are inherent to passive solar construction.

The same design principles have been incorporated into a second house (174 m^2 or 1865 ft^2) recently completed; to increase efficiency, simplify construction and raise buyer acceptability, a number of minor changes were made in the new design, such as poured concrete instead of slump block for the interior storage walls, a simpler roof line, brick floors instead of parquet, and job-built sliding window insulated shutters. Taped and sealed gyp-lap board was substituted for the more easily damaged styrofoam board exterior to the 2x6 stud walls to reduce winter infiltration (and summer heat gain). This spec house (a HUD design and construction award winner) sold within a few days after being put on the market.

In general, a well-thought-out, practical floor plan with a room layout that fits the requirements of the people living in the house is of prime importance to marketing a solar house, even outside of the energy savings. The solar system must be well-designed and construction must be of the highest quality. In the Lumsdaine house, desirable features have turned out to be the following: the masonry walls (no upkeep, noise barrier), planter in the atrium (winter humidity), air lock entry (privacy), a well-designed kitchen with two sinks (on separate walls) and an island range with a pop-top skylight instead of a vent, and a combined laundry room/shower for extra winter heat [3].

6. ENERGY CONSUMPTION

Figure 7 shows total monthly energy consumption for the Lumsdaine house which is equipped with the usual appliances: 24 cu. ft refrigerator-freezer with ice maker/dispenser, dishwasher, washer, dryer, electric range, oven, microwave oven, and trash compactor. In addition to these, there are two blowers (1/2, 1/3 HP) for the solar roof system and the pumps and blowers of the two solar water heaters, together with a conventional water heater.

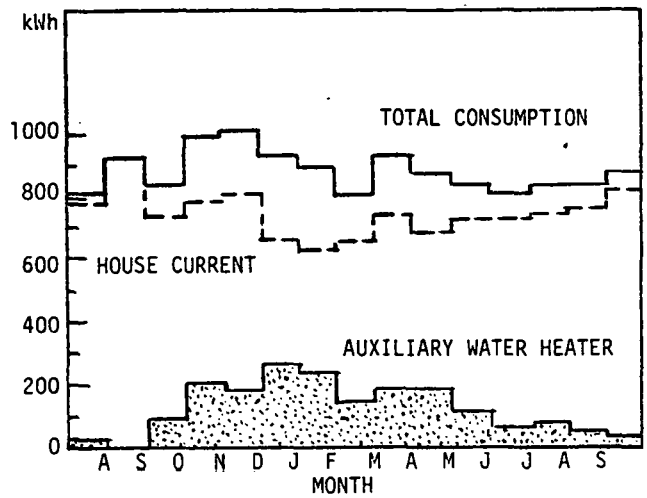


Fig. 7 Lumsdaine House Energy Consumption/Month (2730 sq. ft, 6 people)

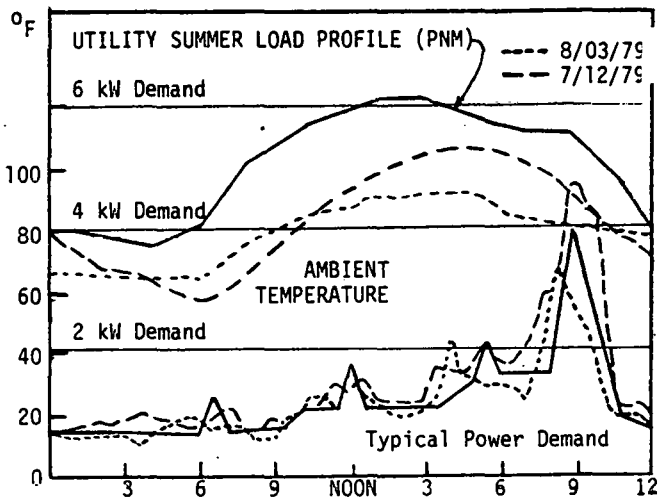


Fig. 8 Typical Summer Power Demand Profile for the Lumsdaine House

There is also a collection of hair dryers, stereo sets, clocks, crock pots, radios, TV, etc. The energy consumption of these appliances has been measured separately; it is estimated that yearly total energy consumption will remain close to 10,000 kWh. The refrigerator/freezer will consume about 30 percent of this, with cooking, baking, lights and appliances 40 percent, the solar water heaters and the fans of the solar roof system 15 percent, and the electric backup water heater also 15 percent. In comparison, typical annual residential energy consumption for a house and family of equal size in Southern New Mexico is approximately 30,000 kWh with evaporative cooling and almost 40,000 kWh with refrigerated air conditioning.

In July 1979, the utility (El Paso Electric Company) installed a time-of-day demand meter in the house as part of their loads research program. Figure 8 shows the energy demand profile for two typical summer days, together with ambient temperature. During school days, the demand between 8 a.m. and 3 p.m. would be a base load without the noon peak; otherwise, the winter profile is expected to follow pretty much the same pattern. Figure 8 does not include power consumption for the auxiliary water heater. Its load was found to be somewhat more concentrated in the morning hours; daytime peaks do occur only during cloudy weather. To make this water heater an off-peak load, it should be coupled to a timer, so it will be charged up only at night. The water heater demand seems to be strongly influenced by dishwasher use.

Figure 8 clearly shows that this passive solar house does not contribute to the mid-afternoon summer peak load of the utility. The peak demand in the late evening of over 4 kW is due to living and housework habits and could easily be changed if necessary (i.e. if a 2 kW photovoltaic system were to be installed in the house). This high peak indicates simultaneous operation of several electric appliances such as oven, clothes dryer, range and dishwasher.

7. REFERENCES

[1] Monika Lumsdaine, "Design Calculation Procedure for Passive Solar Houses," Report #22-SA, New Mexico Energy Institute, New Mexico State University, Las Cruces, NM 88003, 22 April 1979.

[2] Edward Lumsdaine, Monika Lumsdaine, "Lumsdaine Solar House: Summer and Winter Data and Experience," Proceedings, 3rd National Passive Solar Conference (ISES), San Jose, California, 11-13 January 1979, Vol. 3, pp. 741-745.

[3] Figure 9 shows the floor plan of the Lumsdaine house for reference, where 1 = air lock entry, 2 = living room, 3 = laundry/bath, 4 = dining room with skylights, 5 = kitchen, 6 = bedrooms, 7 = atrium with skylights, 8 = study, 9 = patio, and 10 = garage with workshop.

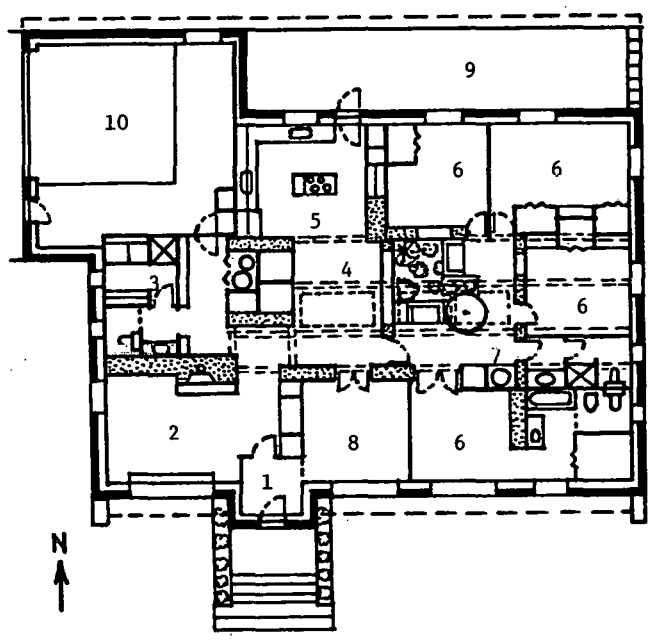


Fig. 9 Floor Plan

A LOW COST RETROFIT SOLAR HOME FOR DEMONSTRATION AND CONSUMER ACCEPTANCE EXPERIMENTS

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ABSTRACT

Recognizing the need for a retrofit solar system for demonstration in the Piedmont Region of North Carolina and the need for studies on consumer attitudes towards acceptance of retrofit solar use, a project was undertaken to retrofit the University's existing two-story Home Management House. The solar system consists of active solar water heating and passive space heating. Two collectors (55 sq. feet area) for the water heating system were fabricated on site, using 2"x6" pine joists, EPDM absorber extrusion, Urethane insulation boards, polycarbonate double wall extruded Tufak-Twinwall glazing and copper headers. The collecors are easy to construct and low-cost ($5.50-6.00/sq. ft.). The thermostat in the storage tank is set at 100°F so that it seldom comes on. If needed, an in-line demand heater raises the temperature of the water to the required level only when the water is being used. A large solar greenhouse (22'x10'x12') attached to the southern side of the house provides the passive heat gain which is supplemented by a wood stove and the existing gas heating system of the house. The system is economical and ideally suited for retrofitting.

INTRODUCTION

There is a great need, all over the United States, to demonstrate the use of solar energy on existing homes. In order to fulfill this need in the Piedmont Region of North Carolina, a project was undertaken to retrofit the University's existing two story Home Management House for demonstration of solar heating and some other energy-conservation ideas. In order to motivate the average homeowner to install a solar system, it is very important to demonstrate:

(a) The economic advantage of using solar energy and energy conservation techniques.

(b) That the aesthetics of the house do not have to be sacrificed in order to achieve the above.

The home management house is a two story brick Georgian style house with 2700 sq. ft. of heated area. A low-cost system was designed and installed his house, which is suitable for many existing es.

The solar system consists of active solar water heating and passive space heating. A unique feature of the water heating system is the design of the collectors using Solaroll® EPDM Extruded absorber material. The design is such that the collectors can be fabricated by do-it-yourselfers at a materials cost of $5.50-$6.00 per square foot. This represents significant savings in collector cost when compared to commercially available collectors ranging in cost from $10.00 to $20.00 per square foot. In addition the collectors have proven to be very efficient. For space heating a solar greenhouse was constructed to replace an existing covered porch on the buildings south side. In addition to the solar system certain other energy conservation ideas are also demonstrated in this house. To cut down the storage losses and operate the collectors at higher efficiency the thermostat in the storage tank is set at the minimum level (110°F), so that the back-up heating element in the tank is seldom energized. An In-Line Demand Heater with a proportional controller raises the temperature of the water to the pre-set level of 140°F only when hot water is being used.

A pyranometer measures the insolation at the collectors while a BTU Meter measures the amount of heat delivered to the storage tank. So far, the performance of the water heating system has been very satisfactory. It is hoped that the low cost and the good performance of the system will help motivate consumers in this region to install such solar systems on their own homes.

THE SYSTEM

A. Water Heating System

The water heating system consists of a closed loop solar water heater, an 80 gallons stone-lined storage tank with heat-exchanger and an In-Line Demand Electric Heater with a proportional controller for power input. Fig. 1 shows a schematic of the water heating system. The total cost of the system excluding labor is less than $1400. Most of the components used in the system are readily available and therefore don't need much explanation. However, two components of the system, namely, the collector and the demand-heater with

proportional controller, need some explanation because of the uniqueness of their design.

The design of the collector is such that it can be fabricated on site by anybody with some mechanical aptitude. Two collectors of 55 square feet area used in the system were fabricated on site, using 2"x6" wood joists, EPDM extrusion (Solaroll), insulation boards with foil backing, Tufac-Twinwall glazing and copper headers. The total cost of the materials is $5.50-$6.00 per square foot. Figures 2, 3 and 4 show different views and the details of the collector. The collectors were installed flat on the roof making their tilt angle approximately 30 degrees.

The "In-Line Demand Heater" is a 9 KW electrical resistance heater which heats the water as it passes through it. A heater of this kind, which is available in the market, cuts down the storage losses. However, it is still wasteful unless power input to it is controlled proportional to needed temperature rise. For this purpose a controller was designed to control the power input to the heater, to be proportional to the difference between the set temperature needed and the temperature of the water incoming from the storage.

B. Space Heating System

A large solar greenhouse measuring 22'x10'x12' is attached to the southern wall of the house, for passive heat gain. The size of the greenhouse was decided by the constraints on the house but the angles were chosen to maximize the heat gain in the winter. A variable speed controlled fan is used tc transfer the hot air from the greenhouse to the house. Double walled polycarbonate sheets (Tufac-Twinwall) are used as the glazing for the greenhouse. The heat from the greenhouse is supplemented by a wood-stove and the existing gas heating system of the house. There is no performance data on the space heating system yet, but it is expected that the solar greenhouse will provide about 25% to 30% of the heating needs of the house.

SYSTEM PERFORMANCE

The system is being monitored to measure the amount of solar insolation incident on the collectors and the amount of useful heat delivered to the storage. An Eppley Precision Pyranometer is used to measure the solar irradiance which is recorded on a strip-chart recorder. The pyranometer is attached to one of the collectors at the same angle as the collector (about 28^{o}). A BTU Meter (from Natural Power Inc.) is used to measure the amount of useful heat delivered to the storage. The BTU Meter measures temperatures in the solar loop at the entrance to heat exchanger and at the exit from the heat-exchaner and measures the flow rate in the solar loop (Fig. 1). The amount of water used from the storage tank is measured using a water flow meter in the water line.

The data was recorded in the month of September, 1979. Two separate days' data is being presented here. Out of these, one day was almost totally clear and the other day was totally cloudy. Although data over a long period of time is needed in order to draw any conclusions on the average performance of the system, but in the absence of such data, it is hoped that performance on these two distinct days will give some idea of the system's performance. Table 1 gives the experimental results of the performance of the system on a totally clear and a totally cloudy day in the month of September, 1979.

It is seen from the table that the overall efficiency of the system is quite good. One factor that probably contributed to the high efficiency of the system was the low operating temperature of the system. The operating temperature of the system was lower because this experiment was conducted with the electrical heating element of the storage tank, turned off. When an experiment is conducted with the electrical heating element turned on, the efficiency may be slightly lower than 37.7%.

It is hoped that a system like this one, which is low-cost and efficient will demonstrate the economic feasibility of solar water heating to the consumers in this region and encourage them to install such systems on their houses.

TABLE 1. SYSTEM PERFORMANCE PARAMETERS

Type of Day	Parameter	Experimental Performance
Totally Clear Day (Sept. 18)	Insolation (KJ/Day)	104,631
	Heat to Storage (KJ/Day)	39,455
	System Efficiency	37.7%
Totally Cloudy Day (Sept. 21)	Insolation (KJ/Day)	23,706
	Heat to Storage (KJ/Day)	6,224
	System Efficiency	26.26%

Acknowledgement: This project was funded by the North Carolina Energy Institute, Research Triangle Park, North Carolina.

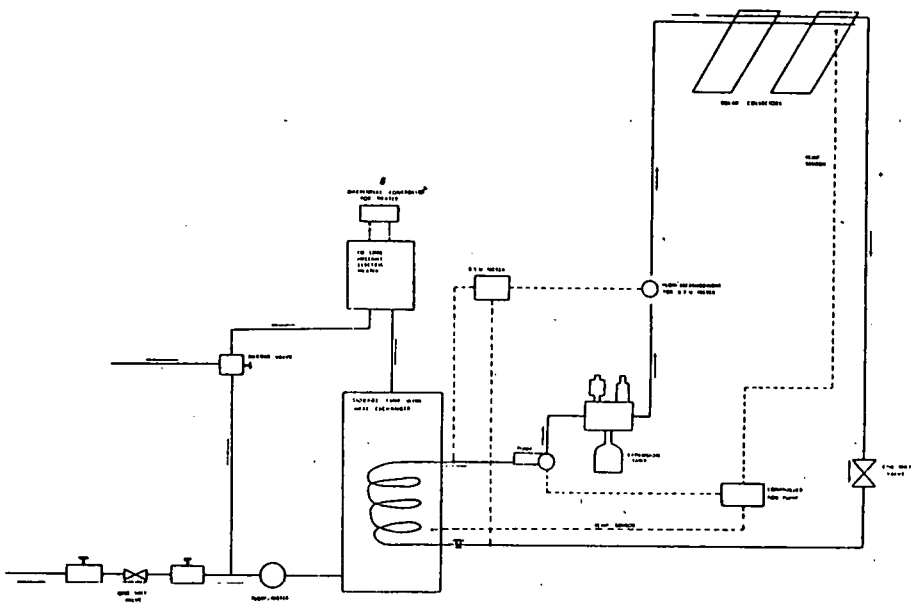


Fig. 1. Schematic of Water Heating System.

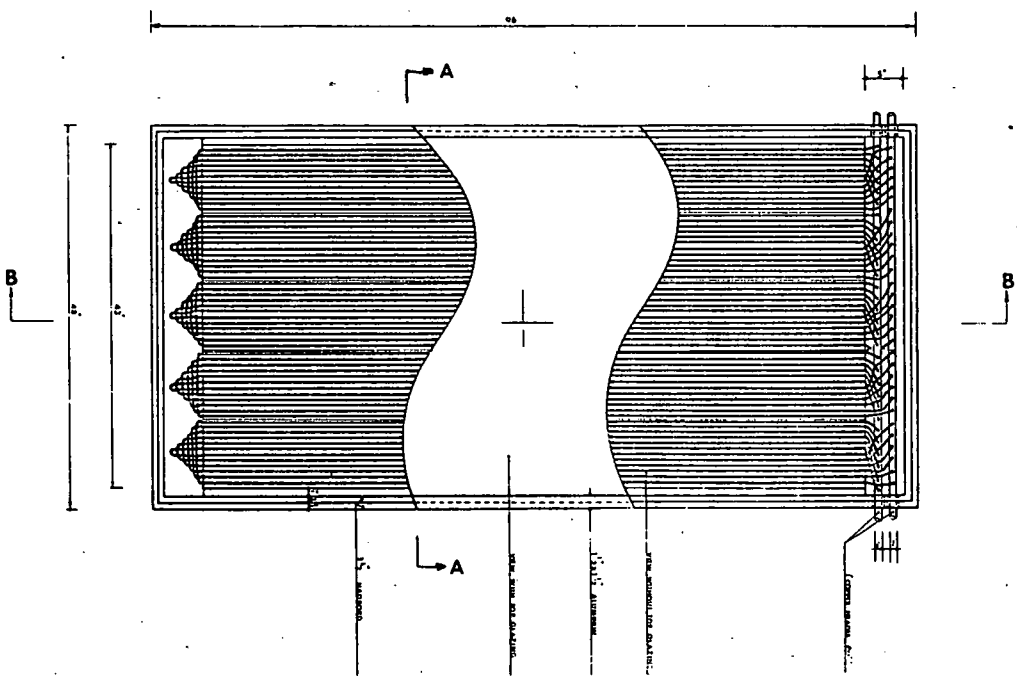


Fig. 2. Solar Collector - Plan with and without Glazing.

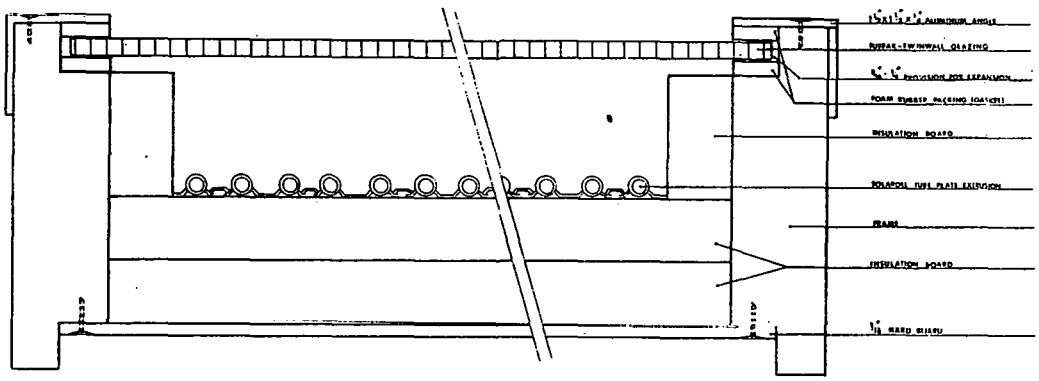


Fig. 3. Solar Collector - Section A-A

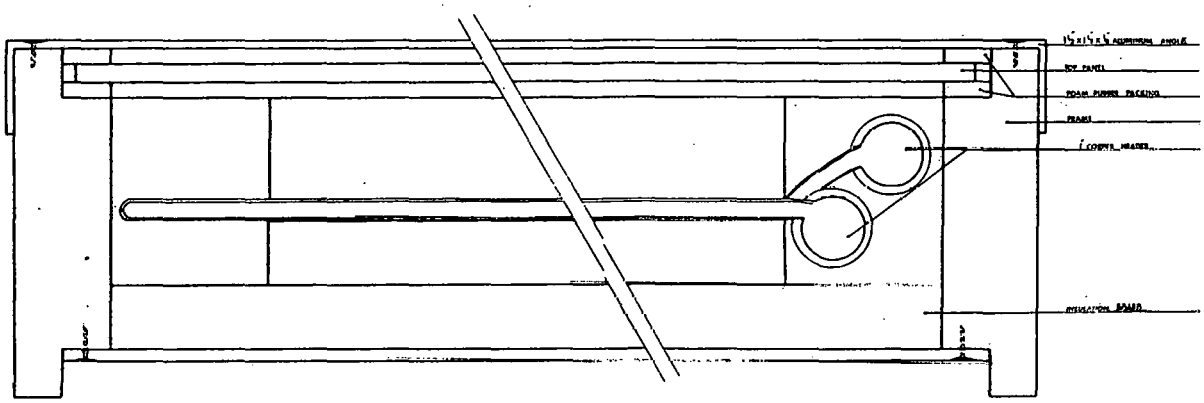


Fig. 4. Solar Collector - Section B-B.

ECONOMICS OF HEATING MARYLAND BROILER HOUSES WITH SOLAR ENERGY

by

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ABSTRACT

Heating costs for broilers were compared, during 1978, between a solar energy system and conventional (propane) heating at the Environmental Research Broiler Facilities of the University of Maryland's Poultry Research Farm in Salisbury, Maryland. Both series of test flocks utilized the limited area brooding concept. Heating cost per 1000 broilers for the conventional system was $13.54 and for the solar system was $54.43. Low utilization of relatively high fixed cost solar facilities due to current management practices caused the sizeable difference in cost. Finding alternative uses for the overflow energy from the solar system and reducing fixed cost of solar facilities are the keys to economic feasibility of solar energy in broiler production.

INTRODUCTION

A solar energy collection, storage and distribution system was designed, constructed and installed in the Environmental Research Broiler Facilities at the University of Maryland's Poultry Research Farm in Salisbury, Maryland during the Fall of 1977. The system is an air system with 41.81 m^2 effective collector area, and 14.27m^3 of rock storage designed to deliver a daily average total of 527,400R_j of useable heat.[1/]

OBJECTIVE AND PROCEDURE

This study was completed to identify and compare fixed, variable and total costs of this solar system with those of the conventional (propane) petroleum based heating system. Details of cost and fuel used were carefully recorded for the calendar year 1978. The data were analyzed and are presented in the following section.

DISCUSSION AND RESULTS

Fixed Costs

For the conventional (propane) system, fixed costs were a $60 brooder, amortized over 10 years at 10% and divided over the four test flocks per year. In this case the value was $2.44 (Table 2). This charge was made against both the conventional and the solar system. For the conventional system, the propane brooder was the sole source of heat (sensible heat was excluded from both calculations). For the solar system, the brooder was necessary as a supplementary heater in extremely cold weather and for emergency purposes.

For the solar system total fixed costs were determined to be $9835.46 or $235.89 effective collector area. This cost was amortized over 20 years at 10% to yield an annual fixed charge of $1158.66 to be distributed over 34 test chambers of 1000 birds each or $34.07.[2/]

Variable Costs

Fuel and maintenance were the two variable costs considered in the conventional system. Fuel was the single largest expense for the conventional heating system, $22.82 for the full house brooding and $10.35 for the limited area brooding system (Table 2). Maintenance cost for the conventional system was determined to be 5% of the purchase price of the heater as an annual charge divided by the number of flocks per year. In this case the cost was $0.75.

There was a small fuel cost ($2.67) for the solar system to cover supplementary heating needs of the flock. Maintenance costs for the solar system were determined to be 5% of the purchase price annually divided by the number of flocks per year. For the solar system this also included costs of operating the necessary fans. The maintenance charge of $0.75 for the conventional

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1/ "A Solar Collection, Storage and Distribution System for Broiler Production", R.B. Brinsfield and K.E. Felton, Scientific Article Number A2641, Contribution Number 5682, Maryland Agricultural Experiment Station, 1978.

2/ To facilitate research on the solar system itself, it was designed to provide much more heat than was necessary for the test chamber being evaluated here. In order to make a fair allocation of fixed costs, it was determined that 34 flocks of broilers could be theoretically heated by this system (Table 3). Therefore, fixed costs were spread over 34 flocks and not the four test flocks.

back-up heater was also included.

Table 1. Total Fixed Cost, University of University Maryland Solar Energy System Salisbury, Maryland, 1978

Collector - 41.81m^2		$2,709.17
Duct work		
Collector	$ 144.00	
Delivery	1,091.80	
Distribution	22.00	
Total		1,257.80
Rock Storage (14.27m^3)		1,573.00
Controls		510.00
Fans and motors		525.00
	Sub-total	6,574.97
Labor (50% of fixed cost)		3,287.49
	Total	$9,862.46
Total fixed cost per m^2 of collector		$ 235.89

COMPARISON OF TOTAL COST OF ENERGY SYSTEMS

Total heating cost per 1000 birds for the full house brooding system using conventional heat was $26.01. For limited area brooding, the cost was $13.54, a 48% decrease in heating cost over the full house brooding system.

Total heating costs for the solar energy system was $54.43 per 1000, four times the cost of the comparable conventional heating system during the test year 1978.

Table 2. Energy Cost per 1000 broilers University of Maryland System Salisbury, Maryland, 1978

	Conventional		Solar
	Full House Brooding	Limited Area Brooding	Limited Area Brooding
Fixed Cost			
Brooders	$ 2.44	$ 2.44	$ 2.44
Solar System	----	----	34.07
Total Fixed Cost	$ 2.44	$ 2.44	$36.51
Variable Cost			
Fuel(@ $0.16/L)	$22.82	$10.35	$ 2.67
Maintenance-			
A) Conventional	0.75	0.75	0.75
B) Solar (including electricity for fans)	----	----	14.50
Total Variable Cost	$23.57	$11.10	$17.92
Total Cost	$26.01	$13.54	$54.43

UTILIZATION OF SOLAR ENERGY GENERATED

It was determined that under optimum conditions, the University of Maryland solar energy system (assuming 60% efficiency) could have delivered an annual total of 196,222,756 K_j or an average of approximately 527,400 K_j/day. However, due to special heat needs of broilers (concentrated in the first weeks of the eight week growth cycle) and the four flock per year management system at the test facilities, only 21.2% or a potential of 41,599,224 K_j, was theoretically available to heat broilers.

The rest of the deliverable heat may have been utilized for other purposes. However, during the test year it was not.

This was the principle reason that the cost of the solar energy generated was so high compared to the conventional heating system. The solar system was used only 11 of the 52 weeks, and the high fixed costs had to be covered during a small fraction of the year. For example, if even 75% of the potential deliverable heat could be used in broiler production, then the heating cost for the solar system would drop to $19.62/1000 birds. Considering the rapidly rising prices of conventional fuels, this would make the solar alternative quite attractive.

CONCLUSIONS

During the 1978 tests, the cost of heating 1000 broilers in Salisbury, Maryland under limited area brooding regimes was $13.53 for conventional and $54.43 for the solar system. The principle reason for the high solar energy cost was the low rate of utilization of the fixed solar facilities.

RECOMMENDATIONS

Utilization of solar energy heating system in broiler production can be improved by:

1. finding alternate uses for the overflow energy.
2. adapting the management system to use the solar system a greater portion of the year.
3. modifying the solar system to reduce fixed cost.

Scientific Article Number A2660, Contribution Number 5701 of the Maryland Agricultural Experiment Station, Department of Agricultural and Resource Economics, Department of Agricultural Engineering. This research was also sponsored by the United States Department of Agriculture in cooperation with the Department of Energy and the Delmarva Broiler Industries Association.

e 3. Energy Utilized to Heat Test Broilers
Under Selected Management Systems
Salisbury, Maryland, 1978

(K_j/bird/week)

1978

A. Full House Brooding

Week	Jan.- Mar.	Apr.- June	July- Sept.	Oct.- Dec.	Total	Average
0-1	3762	1436	---	656	5854	1464
1-2	2797	938	---	516	4251	1063
2-3	2591	---	---	328	2919	730
3-4	891	---	---	257	1149	287
4-5	422	---	---	---	422	105
Total	10,464	2373	---	1757	14,594	3649

B. Limited Area Brooding

Week						
0-1	2333	665	---	410	3408	852
1-2	621	382	---	352	1355	389
2-3	352	---	---	375	727	181
3-4	398	---	---	340	737	186
4-5	387	---	---	---	387	97
Total	4092	1046	---	1477	6615	1654

C. Solar & Limited Brooding

Week						
0-1	1478	---	---	105	1583	395
1-2	117	---	---	---	117	30
2-3	---	---	---	---	---	---
3-4	---	---	---	---	---	---
4-5	---	---	---	---	---	---
Total	1595	---	---	105	1700	425

D. Solar Energy Utilized (K_j/bird/week)

Week	Limited Area	Solar + L.A.	Average Solar used
0-1	852	395	457
1-2	339	30	309
2-3	181	---	181
3-4	186	---	186
4-5	97	---	97
Total	1654	425	1229

Propane = 26,176 K_j/L

NOTES

THE USE OF OPERATIONAL RESULTS TO IDENTIFY POTENTIAL IMPROVEMENTS IN THE THERMAL PERFORMANCE OF AIR SOLAR HEATING SYSTEMS AND TO ESTABLISH PERFORMANCE CRITERIA

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ABSTRACT

The National Bureau of Standards is developing definitive performance criteria for solar heating, cooling, and hot water systems. This study was performed to quantify the effect of air leakage on the thermal performance of air type solar heating systems.

Field air flow measurements taken at seven instrumented residential air type solar heating systems show substantial air leakage. A TRNSYS computer model was developed based on the system at one instrumented site where air flow measurements were taken. Air leakage from this system had been reduced as much as possible. The effect of air leakage on system performance was examined analytically by including the measured air leakage in the system model.

The results of the computer simulations show that by eliminating all of the measured air leaks that remained after the modeled system was repaired, a reduction of as much as 19 percent in seasonal auxiliary energy can be realized. The relationship between air leakage and auxiliary energy use in this system is described.

INTRODUCTION

The National Bureau of Standards was directed by Public Law 93-409, the "Solar Heating and Cooling Demonstration Act of 1974," to assist the Department of Housing and Urban Development (HUD) to develop performance criteria for solar heating and cooling systems in buildings. Air leakage has been identified to have significant occurrence in the operation of air-type solar heating systems. Therefore, a study was undertaken to investigate the effect of air leakage on system thermal performance and to provide guidance in selection of recommended leakage levels for the performance criteria.

The purpose of this study was to:

- Determine the extent of air leaks in existing air-type solar heating systems,
- Determine the component leakage that has the most pronounced effect on thermal performance,
- Quantify the reduction in system thermal performance due to leaks relative to a no-leak condition,
- Determine the availability and cost of hardware for reducing leaks, and
- Provide information that will lead to the recommendation of reasonable leakage rates in the performance criteria.

Field measurements, manufacturers' literature, and computer simulation techniques were used to perform the study. Field measurement of air leakage on air heating systems in the National Solar Demonstration Program were reviewed. An analytical model was developed to permit simulation of both damper leaks and external leaks and their effect on system performance. The model was exercised to simulate the seasonal performance of a typical system located in a relatively cold climate for various combinations of component leakage rates. The simulation results are presented along with information on available low-leakage dampers as a partial basis for recommendation of allowable leakage rate.

FIELD MEASUREMENTS OF AIR LEAKAGE

Air flow surveys have been conducted in at least seven typical residential solar air heating systems in the National Solar Demonstration Program by a HUD support contractor to measure the extent of air leakage in these systems. Velocity profile measurements at key locations in each system indicated that substantial air leakage existed in each system. All of the systems exhibited external air leakage in the collector array and the rock bin storage container, as well as internal leakage through control, back-draft and shut-off dampers. Some of the systems also exhibited leakage along duct seams and localized leakage at duct-to-component joints. Air leakage and blower flow rates were found to vary from one operational mode to another in response to the varying system pressure. Some leak locations were found to infiltrate air in one mode and exfiltrate air in another.

One of the systems surveyed is the basis of the computer analysis performed for this study. This system, shown schematically in Fig. 1*, is described later.

* Figs. 1, 2 and 3 are included at the end of this paper.

The air leakage in this system, measured after extensive efforts by the HUD support contractor to reduce air leaks where practical is shown in Table 1.

There are localized air leaks in each of the systems surveyed. In the modeled system there is a localized air leak at the joint where the solar

TABLE 1. COMPONENT AIR LEAKS IN MODELED SOLAR AIR HEATING SYSTEM

LEAK LOCATION	AIR LEAK RATE PERCENT OF DESIGN FLOW RATE*		
	MODE		
	Storage-to-House	Collector-to-House	Collector-to-Storage
Collectors	-8	-42	-1
Storage	-17	-1	+11
Solar Blower	-17	-19	-11
Damper D2	5	3	14
Damper BD2	2	---	---
Damper MD1A	1	---	---
Damper MD2A	---	---	---
Damper MD2B	---	4	3

NOTE: Leak sign convention for external leaks - positive = leak out (exfiltration), negative = leak in (infiltration).

* Design flow rate = 5547 ℓ/s (1092 cfm).

The collector array design flow rate (DFR) is 5547 ℓ/s (1092 cfm), which corresponds to 10 ℓ/s·m^2 (2 cfm/ft^2) based on gross collector area. The collector air leak rate varies from one percent DFR in the collector-to-storage mode to 42 percent DFR in the collector-to-house mode. In the storage-to-house mode, when the collector loop is isolated from the system by dampers BD-2 and MD1A, the collectors leak eight percent DFR. This air leakage is assumed to be ambient air leaking into the array (infiltration) at internal manifolding joints. Measurements at other sites also having collector arrays operating under negative pressure (due to blower placement) indicated leakage rates varying between 4 and 52 percent DFR. At another site, where the collector array operates under positive pressure, the measured exfiltration leakage rate was 56 percent DFR.

The storage air leak in the modeled system is an infiltration air leak of 17 percent DFR in the storage-to-house mode and one percent DFR in the collector-to-house mode and an exfiltration air leak of 11 percent DFR in the collector-to-storage mode. Other systems exhibited storage air leaks of between 10 and 56 percent DFR (some infiltration and some exfiltration).

The individual damper air leaks in the modeled system and in other surveyed systems range from 1 to 24 percent DFR.

system ductwork connects to the solar blower. This air leak, which was inaccessible for repair in the modeled system but could be prevented in new systems, varies between 11 to 19 percent DFR depending on operating mode.

SIMULATION OF AIR LEAKAGE IN A SOLAR AIR HEATING SYSTEM

System Description

The system is illustrated in Fig. 1. The 28 double-glazed flat plate air-type collectors form a 50.7m^2 (546 ft^2) collector array that is oriented due south at a tilt angle of 57 degrees. The design flow rate (DFR) equals 10 ℓ/s·m^2 (2 cfm/ft^2) based on gross collector area. The mechanical room contains 9.77 m^3 (345 ft^3) of rock storage, a 29 kW (100,000 Btu/hr) gas furnace, two 0.56 kW (3/4 hp) blowers, dampers, ductwork, piping, controls, domestic hot water heat exchanger, storage tank, and heater.

The system is controlled by a two-stage thermostat in the conditioned space and temperature sensors at the top of the rock bin, top of the collector array and in the collector inlet duct. The control scheme is such that air is directed through the c lectors to either storage or to the space when there is a 22°C (40°F) differential between the

collector array inlet and outlet temperatures, and this continues until the differential drops to 14°C (25°F).

On a call for heating, air is directed from either the collectors or storage through the furnace to the conditioned space. When the room thermostat calls for heat, if the temperature at the top of storage is less than 32°C (90°F), and the collectors are not operating, the furnace operates to boost the temperature of the air to the conditioned space, while continuing to draw air from collectors or storage.

The system includes a domestic hot water (DHW) system with a 450 litre (120 gallon) preheat tank, auxiliary tank with heater, recirculation loop pump piping, and an air-to-water heat exchanger in the collector return ductwork. Water is preheated each time the system operates in an energy collection mode until the stored water reaches 60°C (140°F).

Computer Model Description

A TRNSYS [1] computer model of this typical solar air heating system was developed for the simulations performed in this study. A TRNSYS computer model is a group of subroutines, each of which mathematically represents a component in the physical system. These subroutines are connected by information flow paths in such a way that when driven by a set of hourly weather data they simulate the transient response and performance of the physical system. The short and long term effects of various system alterations on the performance of the physical system can be readily determined by altering the appropriate system component model parameters in the TRNSYS system model and performing a seasonal simulation.

The particular modeled system is actually located in a climate characterized by mild winters that are usually overcast. Therefore, the solar fraction during the heating season is normally quite small. To avoid drawing conclusions on system performance in this atypical climate, the simulation was actually performed using weather data for Madison, Wisconsin. Madison, at 43 degrees north latitude, 860 feet above sea level, can be characterized as having a relatively sunny and cold winter typical of the north central states with an annual total of approximately 4,400 degree-C days (7,900 degree-F days). The Climatic Atlas of the United States reports a 56 percent possible sunshine annually. The average annual temperature is approximately 7°C (45°F) while the 97.5 percent design dry bulb temperature for the heating season is approximately -22°C (-7°F).

The TRNSYS system model developed for this study contains all the components found in the physical system except the DHW subsystem (Fig. 1). It includes ducts, dampers, blowers, controls, collector array, thermal storage rock bin, auxiliary furnace, and air leaks. The air leaks are included in the model at the collector array, storage, solar blower inlet, and dampers. The collector array is modeled according to the relationship known as the Hottel, Whillier and Bliss equation as described by Beckman and Duffie [2] and used in TRNSYS. The equations of Close and Yusoff [3] were incorporated to account for the effect of collector array air leaks on the instantaneous array efficiency. These equations modify the collector heat removal factor F_R to account for air leakage (infiltration) equally distributed along the collectors. For purpose of simulation, the solar air heating system was located in Madison, Wisconsin, on a modular two-story townhouse of wood frame construction.

Energy loss, consisting of conductive heat loss and air leakage into and out of solar components located in the mechanical room, was found to be affected by the mechanical room temperature. For the results presented here, the mechanical room is isolated from the living space with a fixed air temperature equal to 20°C (68°F).

The magnitude of air leakage was varied for each simulation performed to determine the potential effect of that air leakage on the physical system seasonal auxiliary energy use, which includes auxiliary thermal energy output of the furnace and electrical operating energy for the blowers. Blower operating energy was also added as heat to the air stream at the blowers.

Simulations

The modeled system was simulated with various leakage rates. Approximately 50 simulations were performed. The system simulated with the measured air leaks used 41.9 GJ (40 mmBtu) of auxiliary energy (including operating energy) to meet a seasonal heating load of 56.8 GJ (54 mmBtu). As shown in Fig. 2, by eliminating just the measured damper leaks, the seasonal auxiliary energy use is reduced six percent and by eliminating all the measured air leaks, the seasonal auxiliary energy use is reduced 19 percent from the measured-leak condition. The corresponding solar fractions of the load for these cases are 26.2%, 30.6%, and 40.1%. Solar fraction is defined here as solar energy input to the house (load minus auxiliary thermal minus operating energy) divided by the fixed load of 56.8 GJ (54 mmBtu).

The savings resulting from reducing damper leaks to one percent of the collector design flow rate (equivalent to replacing all the dampers with low-leakage type) was essentially equal to the six percent savings from totally eliminating these leaks. In the physical system this savings would amount to a $15-$35 seasonal decrease in the auxiliary energy bill (depending on fuel type). The application of commercially available low-leakage dampers appears to be economically feasible. The seasonal economic benefit of eliminating all air leaks in the modeled system is between $50 and $112.

Simulations were also performed with the modeled system where damper leaks were set at one percent of the collector design flow rate (low leakage dampers) and collector and storage leaks were varied. The results indicate that there is one-to-one correspondence, in this system, between the collector and storage leak rates and the increase in seasonal auxiliary energy use. See Fig. 3. For

example, with a collector and storage leak rate of 10 percent of the collector design flow rate, the system auxiliary energy used increased by 10 percent over the no-air-leak case.

CONCLUSIONS & RECOMMENDATIONS

It appears that many air-type solar heating systems in residences leak considerably. In the federal Demonstration Program the air leakage in these systems may decrease performance and interfere with the evaluation of performance.

External leaks in collectors and storage and ducts (if in unconditioned spaces), and past dampers that allow outside air into the system have the most pronounced effect on the thermal performance of the system.

Computer simulation of a system with various external and damper air leaks and an annual solar fraction of 40 percent, has shown that great savings can be realized by eliminating air leaks.

- The elimination of all air leaks results in a 19 percent reduction in the seasonal auxiliary energy use.
- Installation of low-leakage dampers (one percent leakage) results in a six percent reduction in the seasonal auxiliary energy use.

It is expected that the potential savings by eliminating the air leaks in other air-type solar heating systems will be even greater since the system studied here leaks the least of all the systems measured.

To reduce possible thermal performance degradation due to air leaks, the system should be designed to avoid large positive or negative gage pressure at any point. Sealing of ducts and storage unit should be done using long-life sealants. Low-leakage dampers that limit leakage to about 1 percent are probably cost-effective and should be used. It is important to test the system for leakage after construction by air pressurization or smoke tests, and to provide ample space and inspection ports for routine visual inspection and maintenance.

Prevention of air leakage in solar heating systems can significantly increase the performance of these systems. Computer simulation results indicate that thermal performance can be significantly improved if air leakage is controlled through proper design, construction, maintenance, and field testing prior to operation of the system.

It is recommended that the TRNSYS model be revised to include the capability of various degrees of heat transfer and mass transfer exchanges among the following four components: outdoor environment, solar system (collectors, ducts, storage), mechanical room, and house. In addition, it is recommended that the load model be revised to separate conduction losses and infiltration losses to enable investigation of displacement of natural infiltration by air leakage into the house via the solar collectors.

REFERENCES

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[2] John A. Duffie and William A. Beckman, Solar Energy Thermal Processes. John Wiley & Sons, New York: 1974.

[3] D.J. Close and M.B. Yusoff, "The Effects of Air Leaks on Solar Air Collector Behavior," Solar Energy Journal, Vol. 20, No. 5, pp. 459-463.

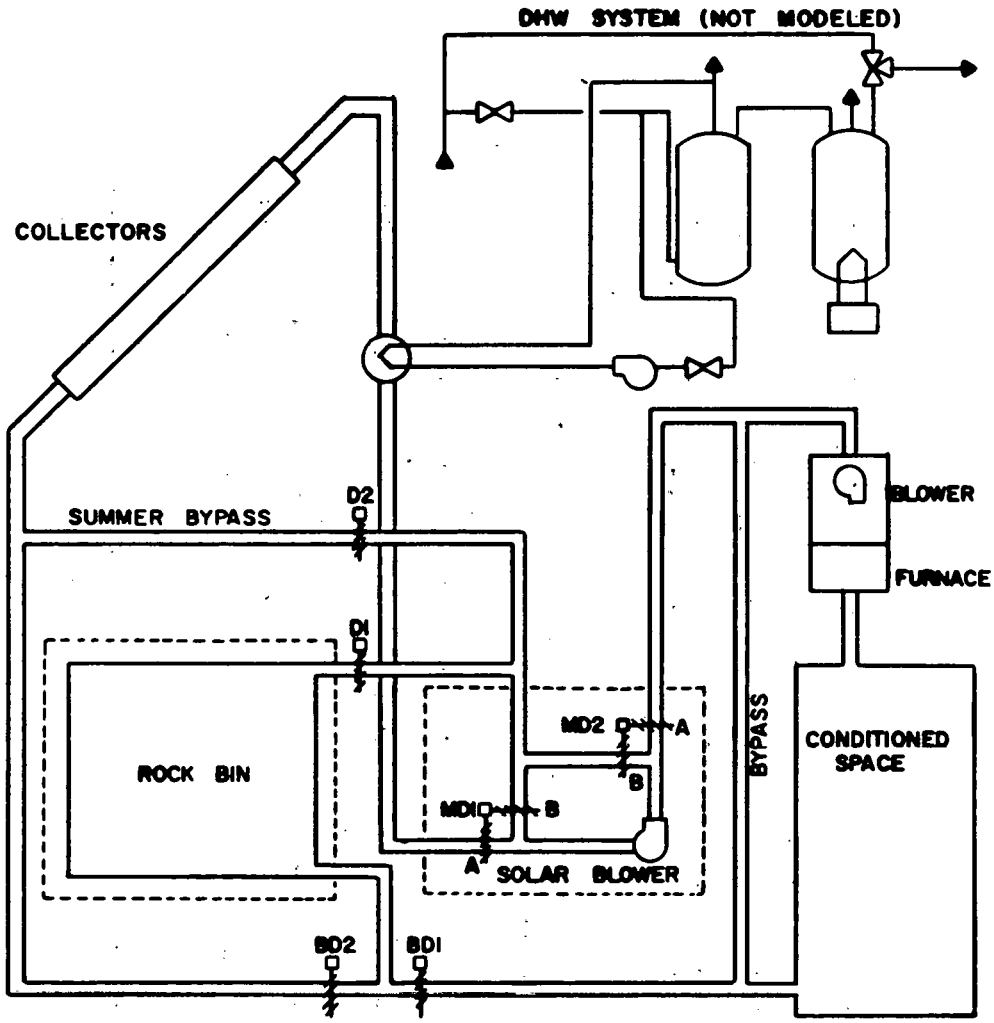


Fig. 1. Solar Air Heating System Schematic

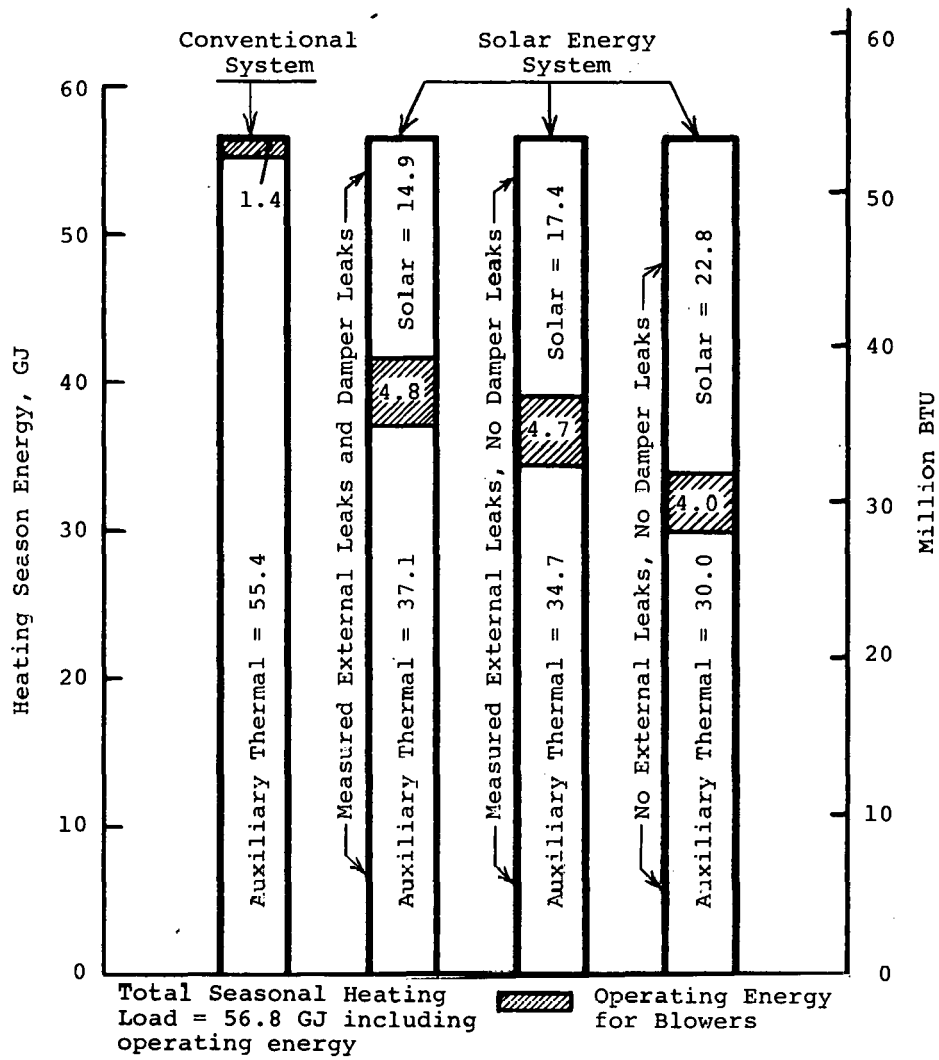
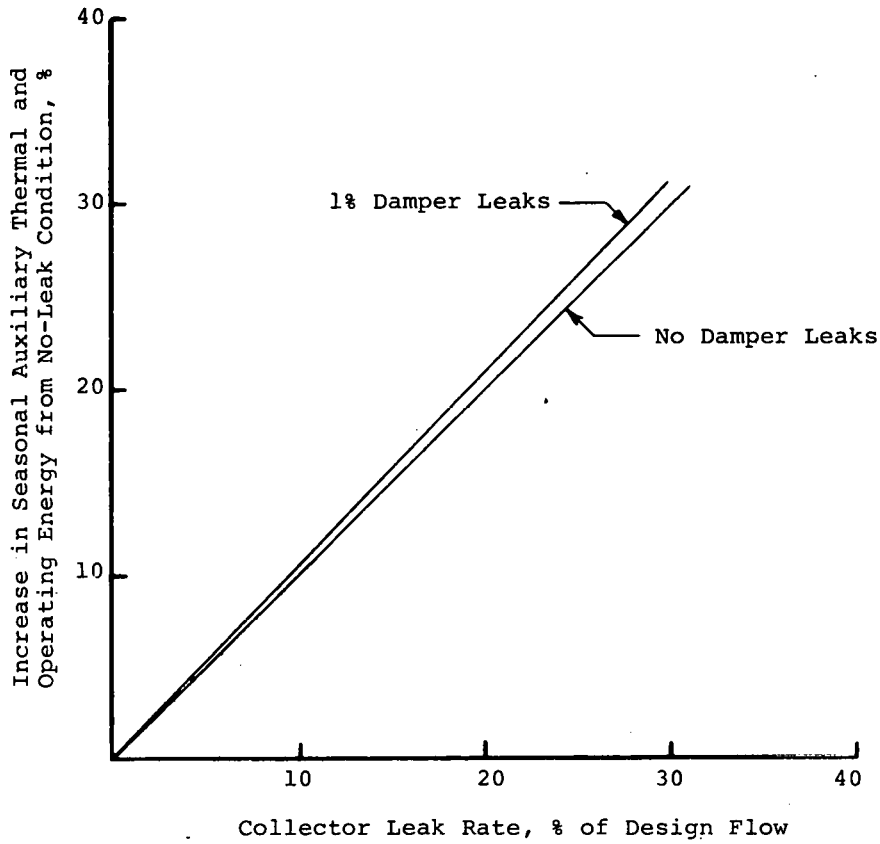


Fig. 2. Computer Simulation Results for a Solar Air Heating System in Madison, Wisconsin With and Without Air Leaks



Notes:

1. In collector-to-storage mode, storage leak = collector leak.
2. In collector-to-house mode, house exfiltration increases by amount of collector leak.
3. In storage-to-house mode, storage leak and house additional exfiltration equal to given collector leak.
4. Mechanical room temperature = 20°C (68°F).

Fig. 3. Relationship Between Collector Leak Rate and Auxiliary Energy Use For Simulated Solar Air Heating System in Madison, Wisconsin

NOTES

OVERVIEW OF FIELD EXPERIENCE WITH THREE ACTIVE SOLAR COMMERICAL BUILDINGS

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ABSTRACT

Utilizing complementary professional capabilities, Bickle/CM INc. and the Burns/Peters Group have designed and built over 20 energy conscious facilities utilizing active solar components. Burns/Peters provide architectural design and planning while Bickle/CM is responsible for the energy analyses and mechanical design. Some of their joint projects include: U.S. Forest Services Tree Nursery, several residences, an office building two apartment complexes, and an Animal Control Center. Three of these projects are described to provide insight into the evolution of solar utilization by Bickle/CM and Burns/Peters.

Experience from the first project identified materials and equipment specification problems which were resolved in the next generation design. Installation or maintenance problems for these designs detracted from system performance until they were corrected. Utilization of field experience and knowledge obtained from past problems-solutions provided an effective solar system for present design stage evolution which exceeds design performance estimates. Proper selection of materials combined with field experience in design and installation creates high efficiency systems in energy conservative buildings which are affordable, effective, and reliable.

INTRODUCTION

Efficient utilization of solar systems requires the effective integration of architecture and engineering in the design process. With complementary professional capabilities, Bickle/CM and the Burns/Peters Group have designed and built over 20 energy conscious facilities with active solar components. Burns/Peters provide architectural design and planning while Bickle/CM is responsible for the energy analyses, mechanical design, and integration of solar components. Some of the joint projects include: several residences, a U.S. Forest Service Tree Nursery, two apartment complexes, an Animal Control Center, and an Office Building. Three of these projects are described in detail to provide insight into the evolution of solar utilization by Bickle/CM and Burns/Peters.

SOLAR VILLAS APARTMENTS

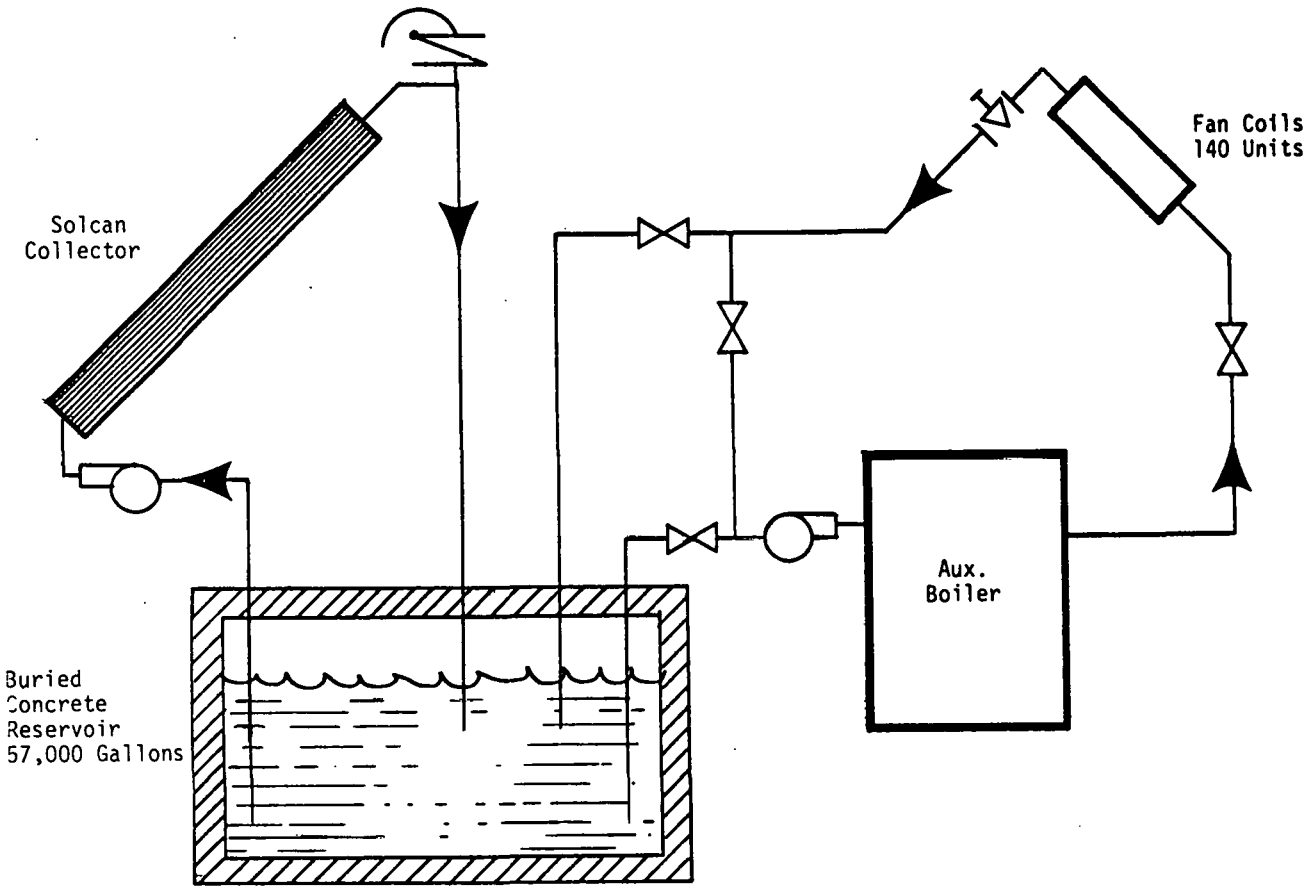
Solar Villas is a 4-story, 101 unit solar-heated apartment building for elderly and handicapped use. Located in Albuquerque, it has 390 Solcan tracking collectors which supply heat through an open loop, automatic drawdown system to a 57,000 gallon buried concrete reservoir. Additional collectors and storage systems provide domestic hot water for the apartments. Operating independently these systems provide 65% of the heating energy. Cooling is provided through utilization of the existing reservoir. During low-load, off-peak periods, chillers produce chilled water which is stored in the reservoir until required by the four-pipe system. This represents an additional savings in cooling costs. Operational data for all three systems is available since May 1977.

Some of the system characteristics, problems, and solutions are described in the following sections and a system schematic is shown in Figure 1.

Reservoir Liner - Originally, the buried concrete reservoir was lined with Hyperlon. During some periods of operation, the collector water temperature was at or above 190°F which resulted in failure of the Hyperlon. The liner separated at the top and fell into the tank. This was noticed during the scheduled biweekly maintenance inspection. After draining the tank during the summer, the liner was replaced with a polypropelene liner.

This material has high temperature stability, high corrosion resistance and was very easy to install. It has lasted over two years without any visible signs of deterioration.

Off Peak Rates - The original design intent was to store chilled water in the 57000 gallon reservoir during high efficiency and off peak rate periods. Mechanical chillers inherently operate with higher efficiencies when there is a large temperature differential between the condenser and outside air temperature. This implies that the best operating period is at night. Additionally, off peak electrical rates are in affect during this period. These rates are not in effect for this area at present, but when implemented, even greater savings will be realized.



Solar System Schematic-Solar Villas
FIGURE 1

Collector Piping Insulation - Since the ambient air temperature in Albuquerque goes into the mid teens during the winter, the collector piping was heavily insulated to reduce transit heat losses. These pipes were wrapped with muslin and sealed with latex paint. The expected lifetime of this wrapping is about five years. Later installations used aluminium jackets to extend the insulation.

Open Loop Fan Coil System - To reduce initial capital costs, an open loop system was installed. This eliminates the first costs of heat exchangers and pumps, but makes the system difficult to maintain. There were also problems with flow balancing and air bleed out which reduce efficiency during operation. The system works in spite of these disadvantages, but unless strictly constrained by the client, experience dictates that closed loop systems are better in the long run.

Series Connection of Collectors - To obtain a higher temperature differential in the collector flow path, the collectors were connected in series. This did result in a high delta T, but problems with the control system resulted in boiling water and creation of steam. In addition, the series connection required shorting lines for drainage. There were also some freezing problems in small lines which did not have enough tilt to completely drain down. After the failure of the liner and the drainage problems were noted, the system was replumbed in parallel.

Submersible Pumps - In the concrete reservoir, submersible pumps were used to lift the water to the pumping station. Because of the high temperature environment, these pumps are very expensive and also have reduced operating lifetimes. They are difficult to maintain and should be replaced with turbine type pumps having the motors above ground and easily accessible.

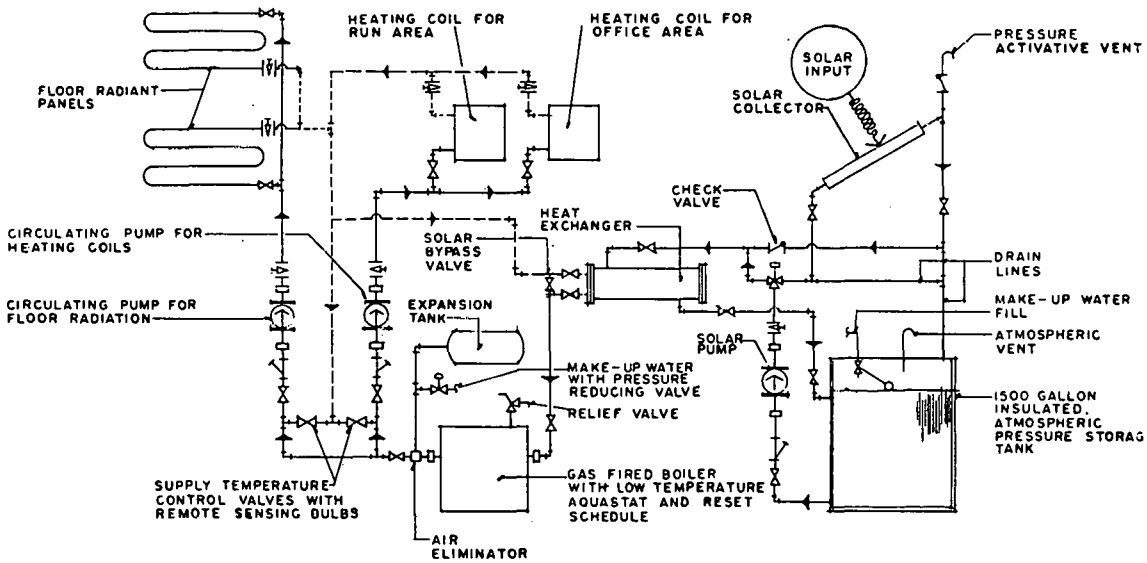
AL CONTROL CENTER

An addition to the Albuquerque Animal Control Center incorporates both active and passive solar elements. The open, active loop has 48 Solcan collectors with 1500 gallons of storage to supply hot water to radiant floor panels and fan coil units. Passive gains provide additional heat for 4300 ft^2 of kennel and 2800 ft^2 of office space with adjustable reflector/insulator panels to regulate the heat flow through 425 ft^2 of glazing. The combined system provides 68% of the heating requirements in the facility.

The schematic diagram for the active solar system is shown in Figure 2, and Figure 3 shows a section through the passive heated dog run area. A brief description of system characteristics, problems, and solutions is provided in the following sections.

Storage Tank Material - Because of the problems with the concrete tank and liner at the Solar Villas, a galvanized steel tank was used to provide storage. This has worked quite well so far with its only liability being a relatively short lifetime, about 8 years, when compared to the building life.

Closed Loop Fan Coil System - To eliminate the maintenance and balancing problems encountered with the Solar Villas open loop fan coil system, a closed loop heating system was used at Animal Control Center. This increases the initial capital cost by requiring a heat exchanger and control valve but significantly reduces downtime and maintenance costs. There have been some problems with the relatively high cost of the control valves and sometimes problems with operation. These problems have resulted in replacement of the valves in newer projects as described in the section about the Burns/Peters building.

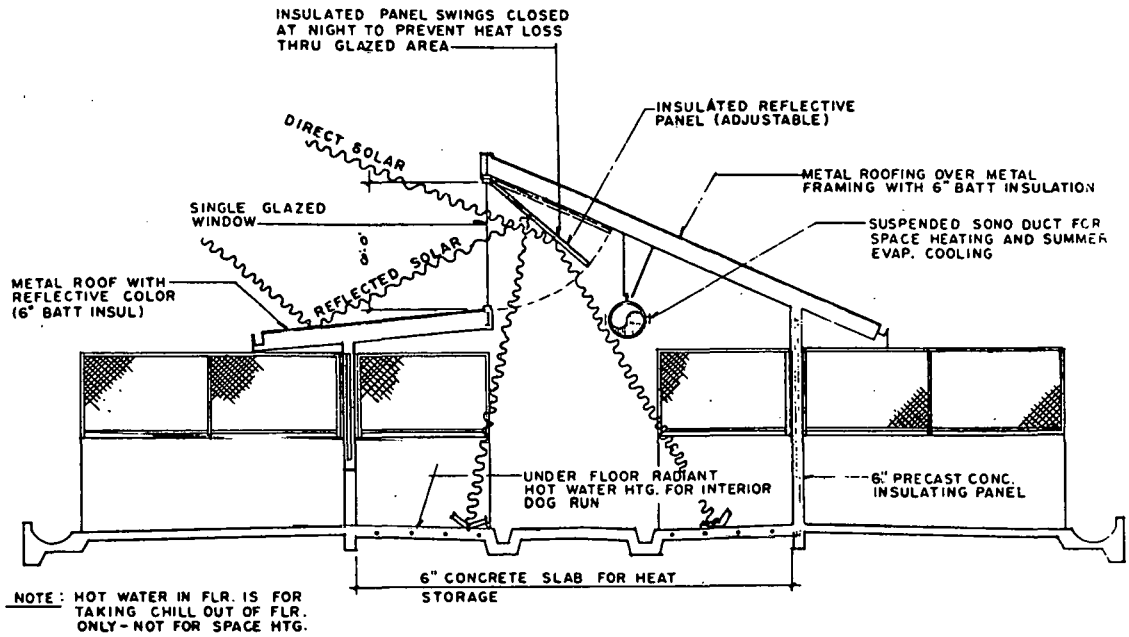


SOLAR AND DISTRIBUTION SCHEMATIC FOR THE ANIMAL CONTROL CENTER

FIGURE 2

Passive/Active Combination - Since the Animal Control Center has two basic use areas, office and kennel, it is very suitable for a combination of active and passive solar. In the kennel area, heat is required both day and night, so a passive system of direct gain and floor storage is utilized. Additional heat for the kennel and conditioning of the office area is supplied by the active system. This combination has worked quite well as the passive system reduces the total load and thus, decreases the size of the active system. This results in smaller initial capital costs and less maintenance are of substantial benefit to the owners.

Collector Piping Insulation - The fiberglass insulation used at Solar Villas was not effective so the insulation was changed to a closed cell material for the Animal Control Center. This was weatherproofed with asphalt paper and then tarred. Cracks have devleoped in the weatherproofing which have increased the amount of maintenance required by this component.



SECTION THRU DOG RUN — PASSIVE HEATED AREA

FIGURE 3

Boiler for Auxiliary Heat - To reduce cycling and fuel consumption, the boiler can be locked out of the solar loop. This requires interface with the solar control network and may not be feasible on large commercial buildings. Also the boiler loop is integrated so that the solar system can be isolated for maintenance without affecting building conditions.

Interfacing Control Systems - On this project, there was no interaction between the solar contractor and the HVAC contractor. Each provided control units which were incompatible with respect to voltages, contact closures, etc. Although a functional block diagram was supplied, no specifications were quoted. This provided too much flexibility for the contractors and resulted in the incompatibility.

BURNS/PETERS SUNSTRUCTURE

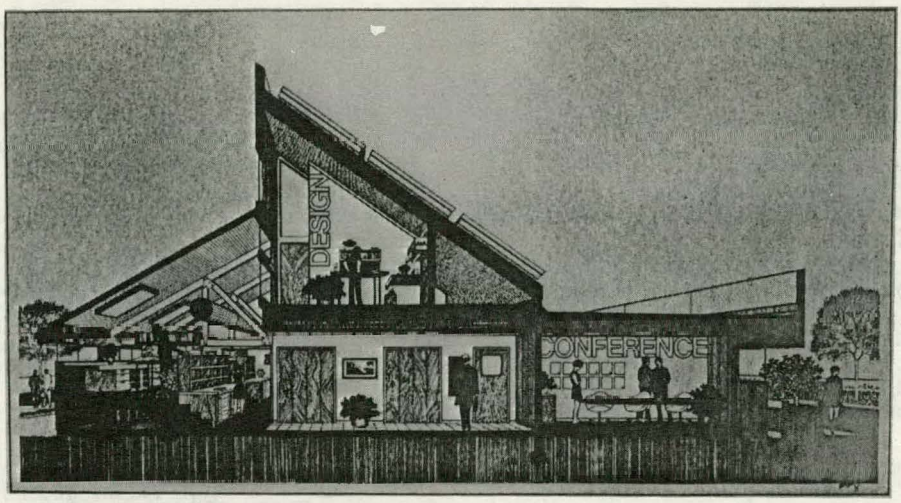
In the 6000 ft^2 Burns/Peters Sunstructure office building, 60 Solcan tracking collectors are mounted on the 35° sloping roof to provide hot water to 2000 gallons of storage. A heat exchanger isolates the pressurized fan-coil loop from the open reservoir. A thermosiphon hot water system, isolated from the reservoir with a double wall heat exchanger, and evaporative coolers have been effectively integrated into the system to provide total conditioning. Approximately 75% of the building heat has been supplied since operation began in early 1979.

A perspective building cross section for the sunstructure is shown in Figure 4. Figure 5 shows a schematic of the solar heating and domestic hot water systems. These and other system characteristics are discussed in the following sections.

Collector Mounting - To eliminate frame material, the building was designed with a 35° slope to accommodate three high collector arrays. The frames were pre-drilled to minimize assembly time and thus allow the structure to be bolted together instead of welded on-site. With large arrays of concentrating collectors, there can be some tracking problems due to reflection from adjacent collectors. These may be minimized by properly locating the tracking sensor, on east side of array to eliminate adjacent collector on that side, and by blacking the back of adjacent west side collectors.

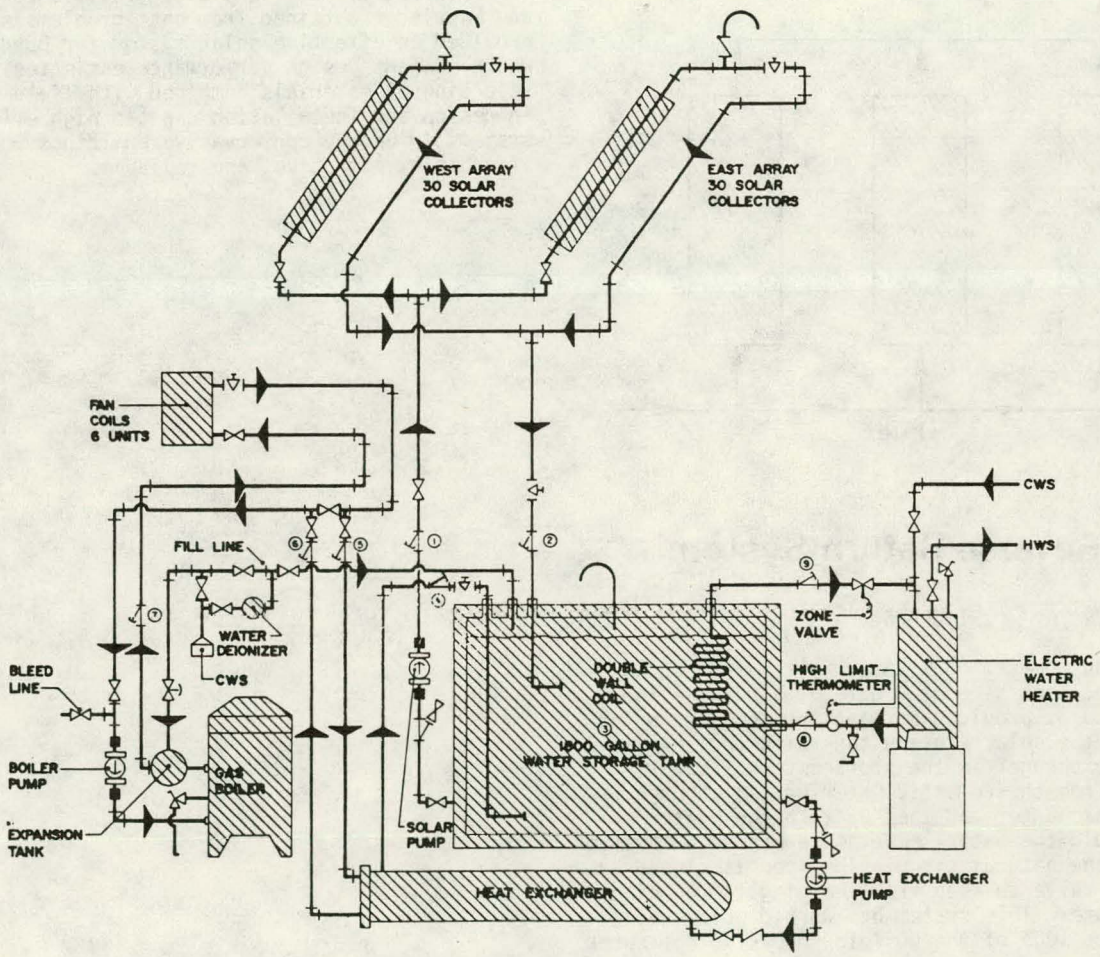
Collector Piping Insulation - The closed cell insulation worked well at Animal Control Center and was again used for the sunstructure piping. However, because the weatherproofing caused problems, the asphalt paper was replaced with aluminum jackets. These were easy to install and have required minimal maintenance.

Storage Tank Materials - Because of the short lifetimes of Hyperlon and steel when utilized in solar applications, a wood tank was built and lined with polypropelene. This was quite easy to fabricate on-site and has performed well. To reduce mineral deposition and corrosion in the open loop system, a deionizer has been included.



PERSPECTIVE BUILDING CROSS SECTION

FIGURE 4

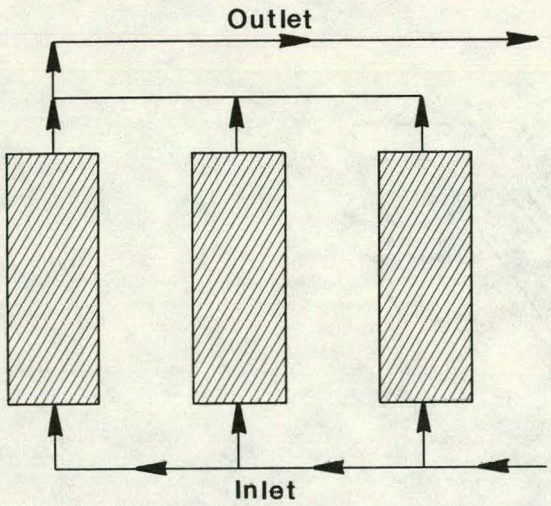


SOLAR SYSTEM SCHEMATIC

FIGURE 5

Closed Loop Fan Coil System - The closed loop system utilizes separate loops for the collector and for the heat exchanger flow paths. This eliminates the need for a three-way valve in the system. Instead of sharing one pump between the collector and building load loops iwth a three-way valve for control, two separate pumping loops are used. This reduces initial costs, minimizes control problems with the boiler/collector interface, and provides more flexibility for system operation. With the three-way valve, single pump system, there were times when the boiler heated the storage tank while heating the building. This problem which exists at Animal Control was eliminated in the Sunstructure building with the two pump, separate loop design.

Collector Flow Balancing - Instead of series connection, reverse return collection is shown in Figure 6 was used. This creates consistent flow path lengths for easy balancing.



Reverse Return System

FIGURE 6

Thermosiphon Domestic Hot Water System - As shown on the right hand side of Figure 5, a natural convection loop os provided between the domestic water heater and the solar storage tank with double wall coil heat exchanger in the storage tank. Water cirulates from the domestic water heater through this heat exchanger and then back to the heater tank. Should the water temperature exceed a maximum of 140°F, the natural circulation loop is closed by a solenoid valve to keep from charging to too high a temperature. This system has worked quite well and provided 100% of the building domestic hot water requirements to date.

CONCLUSIONS

Through an evolution in design processes and field experience, Bickle/CM has been able to eliminate many past problems with active liquid solar systems. Some problems have occured with each system, but these have been corrected on-site and future design modified. An example is the evolution through 3 phases of the collector piping insulation and weatherproofing materials. Starting with fiberglass and painted cloth jackets for Solar Villas, closed cell insulation with asphalt paper and tarring was used at the Animal Control Center since fiberglass was not effective; finally, at Burns/Peters, the asphalt covering was replaced with aluminum jackets to enhance weatherproofing.

Experience from Solar Villas indentified materials and equipment specification problems which were resolved for the Animal Control design. Installation and maintenance problems at Animal Control Center detraced from system performance until they were corrected. Utilization of field experience and knowledge obtained from past problems/solutions provided an effective solar system for Burns/Peters which exceeds design performance estimates. Proper selection of materials combined with field experience in design and installation creates high efficiency systems in energy conservative buildings which are affordable, effective, and reliable.

DESIGN AND OPERATIONAL RESULTS OF A SOLAR SPACE HEATING SYSTEM

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ABSTRACT

This is a disclosure of a space heating system that features a simple approach for capturing solar energy - - a water drain back technique which is fail-safe from freezing and does not have any valves or special venting in the collector loop. Eight systems have been installed with this drain back idea; the original has been in service for more than 3 years with sub-freezing winters.

The system discussed here is a retrofit for a 60-year old wood frame home. The solar collectors and storage systems were necessarily installed on a out building located 100 feet from the home.

Included here will be problems and lessons learned, but will emphasize the design, installation, instrumentation, and performance stated in lay terms.

INTRODUCTION

Solar conferences have historically depicted the results of Federally subsidized projects. These projects have, in most cases, proven to be complex, expensive, burdened with delays, and other problems. For these reasons, most of the Federally assisted projects, while necessary to push the state of the art, have not contributed much to inspiring others to install similiar systems.

What follows is a discussion about a successful, privately funded, simple approach for the design of collecting, storing, and distributing solar heat energy. Solar heating systems have been in use for many years and there has been recent Federal emphasis to promote use of solar energy. Thus, there is wide knowledge of performance of typical solar systems. With this in mind, you will not be exposed to a large amount of detailed measured data. It is more important, considering the present state of the art, that you be provided with the design, installation, and broad performance results based on experience with the 'nuts and bolts' of all phases of this particular system.

Four systems, using the drain back of the collector loop, have been installed for space heating; two of these systems also fire an ARKLA water chiller for air conditioning and all systems provide the heat energy for domestic hot water (DHW). Four additional systems using the drain back technique are in use specifically for DHW. This discussion center around one of these eight systems: a one installed in 1976 for space heating, and in 1978 was enlarged for air conditioning.

LAYOUT

The system was installed near Montgomery, Alabama as retrofit for a 60-year old wood frame home with minimum insulation (2" fiberglass in the walls, 4" fiberglass in the ceiling), 10-ft high ceilings, and 2700sq ft living space. The roof of the building was not properly configured for collector mounting nor proper orientation and was shaded all around with large oak trees. In order that the sun could be more effective, the solar collectors were mounted on an existing out building after being adapted for solar. An underground passageway was installed between the solar building and the home to transfer the solar energy for use.

COLLECTORS

In 1976, 20 collectors (2 rows of 10) were mounted on a slope of 50 degrees from the horizontal and facing south. These collectors were intended primarily for space heating - - especially for the months of December, January, and February in the Montgomery, Alabama area. The collectors each have an apperture of 17.2sq/ft with a glass cover plate that is water white, stipple finish, tempered, and is 1/8-inch thick. The absorber has 1/2-inch OD copper water passages surpentined and encased in aluminum fins. For draining, the collectors were installed with the parallel tubes running horizontally. In 1978 20 additional collectors were installed primarily for air conditioning (to fire the ARKLA 3-ton chiller). The collector building was extended to accept the collectors. For appearance and for effective use in the Winter for space heat as well as Summer for air conditioning, these collectors were mounted on a movable roof hinged at the top along the apex of the East - West roof line. The roof supporting this array can pivot from 20 deg (for air conditioning in the Summer) to 50 deg below the horizontal (for space heating in the Winter), and is moved by a hydraulic actuator operated by a small electric driven hydraulic pump. These collectors have the same case and cover plate but a different absorber. This absorber has 12 verticle ½" OD copper risers encased in aluminum fins; there is a horizontal header at the top and a sloped header at the bottom to promote

complete draining when commanded.

Collector Loop.

Simple design approach and function of the collector loop is probably the most interesting section of the discussion. On command from a differential temperature controller, the pump circulates water from the coldest point in the storage system thru the collectors and back to the hottest storage tank. When the pump stops for any reason, the water is then free to drain back thru the open impeller pump and all the water that is left in the collector loop is in a freeze protected area - - the collectors drain free of water and thus cannot be damaged by freezing ambient temperatures.

Details of the unique features of this collector loop and storage can be followed by referencing Fig. 1.

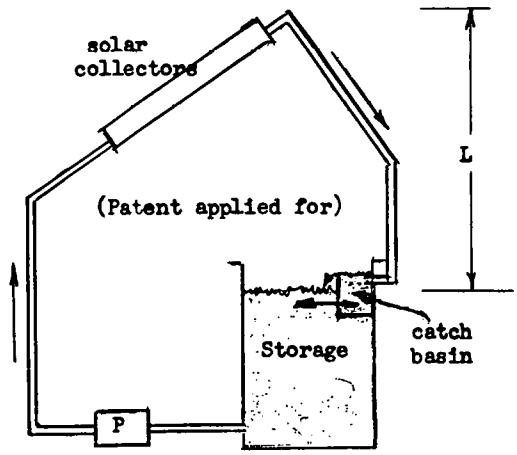


Fig. 1. Collector Loop & Storage

When the pump, P (1/12 HP) is operating, it overcomes the lift, L, primes the collectors and causes overflowing of the small catch basin (about 2-gallon capacity) in the top of the hottest storage tank. As water escapes thru a weir from the catch basin, the water level rises above the inlet from the collectors and results with 'closed loop' operation. When the pump stops, the weir allows water in the catch basin to seek storage water level and uncover the collector inlet. This allows air from the storage system to enter the collector inlet and the water in the collector loop drains back thru the system into storage. This is an inherent 'fail-safe' feature that prevents freezing of the collector system. Note that there are no valves - - vent valves, solenoid valves, nor manual valves necessary for this automatic draining feature. (Patent pending).

The catch basin has the following advantages: (1) it allows closed loop pumping, (2) it preserves temperature stratification in storage by slowing the inrushing water from the collectors into the storage tank, and (3) it provides a source of hottest water available in the system: water is drawn directly from the solar collectors when they are in operation; when the collectors are not functioning, (loop pump idle), the hottest water is still available from the catch basin thru the weir from the upper level of hot water in the storage tank.

STORAGE

Four each 400-gallon, open top, mild steel, tanks were filled to 350 gallons with city water with a Ph of 7 as received from the city water main. The tanks were fabricated with cylinderical sides of 16 gage and the bottom of 14 gage steel (flat rolled). Seams were arc welded. The inside of the tanks were cleaned, primed with zink chromate based primer and then painted with polyurathane. The tanks were insulated from the concrete floor with 3 inches of polyurathane foam board. The sides were wrapped with 6 layers of 1½-inch thick vinyl-backed fiberglass - - common building insulation - - R = 27. The top was covered with a disc made of 3/4-inch plywood with a metal vapor barrier between the plywood and tanks. The top was insulated with 12 inches of batt figerglass; R = 35. The tanks, controllers, ARKLA chiller, and thermometers are located within the enclosed building that supports the solar collectors. See Fig. 2 for interconnection between the storage tanks and plumbing routing to the collectors and other distribution lines.

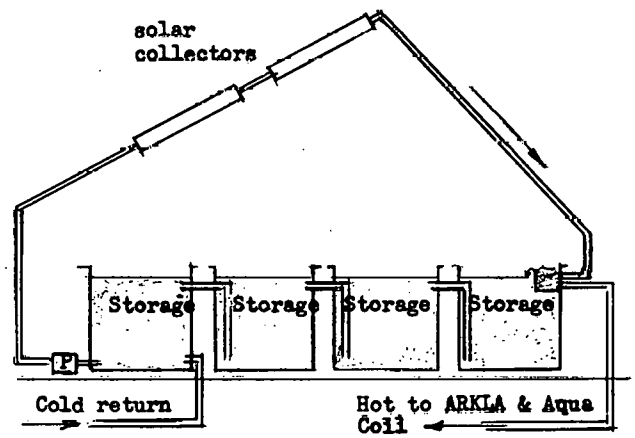


Fig. 2. Storage Interconnects

DISTRIBUTION OF ENERGY

The heat energy is extracted from the hottest storage tank. In the space heating mode, a 1/12th HP pump draws water from the catch basin and forces this hot water thru 3/4-inch CPVC pipe, insulated with 3/4-inch thick Rubatex, in an underground 6-inch plastic passageway. The underground run is 100 feet between the solar facility building and the home. Enclosed in the

ageway is the circuit for DHW as well as the or chilled water for space comfort. In the cooling mode the hot water (from the common space heat pump) is routed to fire the ARKLA chiller. Hand operated gate valves provide for seasonal mode changes so that the same equipment used for space heating (the pump, insulated 3/4 - inch CPVC runs, and heat exchanger in the air handler) is used for firing the ARKLA and supplying chilled water to the home for cooling.

PUMPS and CONTROLS

All pumps in the system are 1/12th HP, except the DHW pump is 1/20th HP, and are open impeller, 120-volt (Grundfos). A 5/15 differential temperature controller commands the function of 3PDT relays. The relays control the pumps and a 24-volt pilot light in the home. A pilot light for each controller-pump provides easy remote monitoring of functioning of the system.

The home comfort thermostats are connected in series with the 3PDT relay control coils (24-volt) to command pumping and air handler fan in the winter. Summer cooling set-up is similar except there is an aquastat in series with the thermostat and 3PDT to prevent ARKLA operation when the firing water temperature is less than 170 deg. F. None of the solar functions will activate the pumps unless the temperature differentials are sufficient to provide useful solar energy. The normal electric back-ups; resistance space heating, air conditioning, and DHW, augment at times when solar energy is not sufficient.

HEAT EXCHANGER

The original heat exchanger was minimum size for sufficient space heating and later proved to be inadequate size for the cooling mode. Therefore an additional aqua coil was installed to provide the necessary capacity for heating and cooling from the common heat exchanger system. The air handler motor was increased from 1/3 to 1/2 HP to maintain proper air flow with the aqua coils added in the air stream.

The water flow thru the heat exchanger is 7 gallons per minute in the cooling mode and 4 gallons per minute in the heating mode (flow is adjusted with a gate valve). This provides up to 3 tons of cooling and about 100,000 BTU of heating with 200 deg. F water.

The aqua coils are upstream from the resistance electric heating and evaporative cooling sections in the air handler. This allows solar heating and cooling to be effective augmentation concurrently when the back-up systems are needed.

INSTRUMENTATION

A direct reading dial thermometer is in the collector feed line near the collector pump outlet another thermometer is in the catch basin to sense the temperature out of the collectors. The thermometer at the catch basin is also used to determine the overnight heat losses from storage. Thermisters are strapped on the collector inlet near the loop pump and a thermister at the outlet of each collector on the top row (20 collectors). Thermisters were installed on the heat distribution lines, one near the pump at the catch basin and one on each side of the aqua coil in the heat exchanger. Thermisters were also installed at the inlet and outlet of the DHW at the solar storage reservoir and the inlet and outlet of the DHW preheat tank to observe the performance of the DHW loop. The temperature sensed by the thermisters was observed remotely (in the home) on an electronic thermometer.

Pump flow rates were determined by measuring the flow into a measured container. The flow thru each of the two 20-collector arrays was 6.7 gal/min. The collectors are connected in parallel at the bottom 10, collectors in series with the top row; thus, the flow thru each collector is .67 gal/min.

DOMESTIC HOT WATER

The DHW is preheated in a 40-gallon tank in the home by a circuit to solar storage. This circuit is a 1/2-inch OD copper pipe insulated with 1/2" thick Rubatex. The heat exchanger in the hottest solar storage tank is a coil of 5/8-inch OD copper 25 feet long. DHW is pumped thru the circuit on command of differential temperature controller that will stop the pump when the DHW reaches 160 deg. F. A 20-gallon electric water heater is placed at a higher level than the pre-heat tank to take advantage of thermosyphoning to allow solar heated water to recover heat losses in the electric water heater -- losses that are normally recovered with electric energy. The electric water heater power is turned off approximately mid-March and not re-energized again until mid-December allowing solar to provide all the heat energy for DHW for 9 months.

PERFORMANCE

Minute and voluminous performance numbers were not obtained; however, the observed performance listed below may prove more meaningful.

Collector loop performance was calculated from manual recordings of inlet and outlet temperatures at random times during steady state operation of the collector loop pumps. The following are some typical Winter, Spring, Summer, and Fall performance data of the collector loop. On numerous occasions in the Spring and Fall when solar heat demand was low, the solar storage water reached boiling temperature - 212 deg. F. There was audible boiling in the collectors during pumping and steam could be seen venting from the overflow from approximately noon until mid-afternoon. Untreated city make-up water was added to replace losses due to boiling.

Date/Time	Temperature ambient/inlet		BTU/hr 40 collectors	BTU/day	Notes
	deg F				
Mid-Jan					
9AM	30	90	60000		Clear sky, light winds.
12noon	45	110	180000	800000	20,000 BTU/hr heat consumption in
4PM	40	170	40000		the home.
Mid-Apr					
12noon	75	205	see note		Low BTU performance because of very little BTU needs (consumption) and high reservoir temperature stagnated at maximum of 212 deg. F.
Mid-Jul					
11AM	90	170	80000		700 gallon storage water. Sky clear.
1PM	95	195	80000	400000	ARKLA operating from 11AM until approx-
3PM	95	200	50000		imately 7PM - - shuts off at 170 deg. F
4PM	95	195	30000		firing water.
Mid-Oct; same as April above.					

Storage heat losses could not be determined with a high degree of accuracy; however, from the data below it is apparent that the heat losses were a minimum due to the insulation of the sides and top of the reservoirs (R27 & R35). The following are observed temperatures at the catch basin:

Reservoir temperature at the end of the collecting day: 200 deg. F. Ambient low temperature during the night: 60 deg. F. No heat consumption during the night. Reservoir temperature at 8AM the next day: 194 deg. F.

There has not been any evidence of leaks developing in the 4 storage tanks. Untreated city water has been in the tanks since 1976. Periodic water samples show no signs of rust or other corrosion (although probably some type of corrosion has taken place by now). The covers have not been removed from the tanks for a visual inspection. Water samples taken from the bottom of the tanks have been visually clear and a Ph = 7.

All pumps, controllers, and associated relays have operated without any malfunction except on one occasion when the pump drawing water from the catch basin would cavitate when subjected to very hot water - - in the order of 200 deg. F. Initially this pump was installed near the top water level at the catch basin. The pump was relocated to a lower level (still drawing from the catch basin) to solve the problem.

The water distribution lines were insulated (with 3/4-inch thick Rubatex) and consisted of copper, galvanized pipe, and for experimental purposes, 3/4-inch CPVC was used in the inlet (lower header) on a collector array and in the underground passageway for the heating and chilled water. There has not been any problem with the distribution system except with the CPVC in the collector array header. This header is supported about 6 inches off the floor at 4 feet intervals. The CPVC performed satisfactorily until the water temperature reached about 200 deg. F. Above 200 deg. the CPVC sagged approximately 4 inches between the center of the span between supports. There were no other problems with CPVC, even at the elevated temperatures, when supported its' continuous length.

Heat losses in distribution was minimum. There was very little distinguishable difference between steady state temperatures at extreme ends of distribution lines between the solar facility building and the heat exchanger in the home.

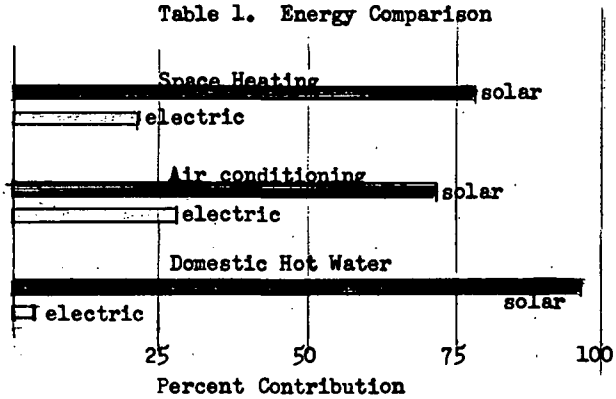
After monitoring this system for 3 years, there has not been any evidence of degradation of performance of any of the subsystems.

ECONOMICS (of this system)

A history of kilowatt-hour (KWH) in this all-electric home is available from the Alabama Power Company from 1962 until the present. A study of this KWH usage shows a dramatic reduction in the use of electric energy since augmentation with solar in 1976. The figures show an annual reduction of 28,000 KWH as a result of the solar influence. This converts to an annual saving of $1400.00 based on current KWH rates - - and these rates are sure to escalate as in the past.

Table 1 on the next page shows pictorily the contribution solar has made in this installation.

Table 1. Energy Comparison



The installed cost of such system is about $25,000.00; this computes to a return on investment in 18 years using current electric rates. Of course this return is going to be much less than 18 years with the expected escalation of utility rates. Worth mentioning is the fact that a quality system is always as valuable as the initial investment - - even after years of use.

With a system such as this, not only are there obvious savings of non-renewable resources and lower utility bills, a most important result must not be overlooked: The home is more comfortable with solar than it was before solar was installed - - especially during periods of ample sunlight.

PRECAUTIONS AND LESSONS LEARNED

1. The collector loop plumbing must be arranged so there is continuous slope from the storage up to the collector inlet, then continuous slope down to the storage tank. This is necessary to assure draining and prevents water traps that might freeze.

2. CPVC plumbing can be used if supported properly if it is to be subjected to high temperature water - - above 200 deg. F.

3. The pumps should be installed as far below the storage water level as possible to provide a positive static water head at the pump.

4. Mild steel storage tanks have proven satisfactory thus far; however, they are not necessarily recommended because of obvious potential for corrosion. Galvanized tanks with annode rods would be a better investment. Not only are they more corrosive resistant, they can be sealed to prevent venting and boil-off; this also allows for a self-pressurizing system resulting with higher attainable storage temperatures.

5. The movable collector array has proven satisfactory; however, considering complexity and cost and if space permits, it is recommended that additional solar collectors be considered at a compromise angle rather than a swing array.

CONCLUSION

A water drainable system can be made durable, to perform reliably, ammune from freezing, and cost effective. The advantages of water over oils and anti-freezes are worth mentioning:

1. Economy - cost of water is much less.

2. Specific heat favors water; it is a more efficient heat transfer medium.

3. Water is not toxic.

4. Water does not contaminate as the others in case of a leak.

5. Water is less viscous and requires less pumping energy than some oils.

6. Water quality does not have to be monitored as critically and replaced periodically as with the others.

7. More efficient with water because a heat exchanger and additional pump are not necessary between the collector loop and storage.

NOTES

PASSIVE ENERGY DESIGN AND HABITABILITY ASPECTS OF EARTH SHELTERED HOUSING IN OKLAHOMA

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ABSTRACT

Identified earth sheltered houses in Oklahoma were examined through a detailed questionnaire during the first phase of a long-range funded project. Preliminary results of energy and habitability aspects are presented herein. Saving energy is reported to be the primary incentive for building such structures. Habitability aspects have generally not received much study until very recently.

Results indicate that although a majority of the respondents feel their energy savings expectations have been reached, over 40 percent feel that their energy consumption is much higher than they expected. Preliminary energy performance studies indicate that in a majority of the projects, the potential thermal mass of the structure has been decoupled by insulation and furred interior surface treatments. This situation can lead to a significant reduction in the amount of free earth cooling available during the summer months. Other factors, not yet studied, undoubtedly contribute additional adverse effects.

Of considerable interest is the finding that the substantial energy savings that are realized have been achieved with little decrease, and often an increase, in comfort and habitability aspects. Most occupants are particularly satisfied with the safety of the structure and the arrangement of the rooms, which in most cases were custom designed by or for the occupants. However, daylighting and privacy of family members, for example, were not as highly rated. A number of other parameters are identified from the survey which present implications for design enhancement in this contemporary type of residential structure.

INTRODUCTION

Project Statement

Earth sheltered architecture is currently being considered in many areas of the country as a viable means to cope with increasing energy costs in buildings. About 80 completed earth sheltered houses in Oklahoma are now being analyzed through mailed questionnaires, limited personal interviews, and limited on-site measurements. The map in Figure 1 indicates the sites identified thus far. The study deals with design, construction, energy, comfort, and consumer parameters.

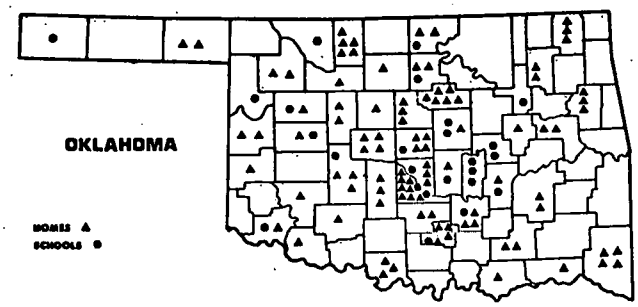


Fig. 1 Identified Earth Sheltered Structures

Specifically, this paper will address the issue of passive design with regard to energy savings as well as the concomitant issue of habitability aspects accruing with such earth sheltered designs.

Background

The present work is the first phase of a long range project on natural energy design in the built environment. It is funded by a Presidential Challenge Grant from the Office of the President of Oklahoma State University to the Departments of Architecture and Housing Design and Consumer Resources. Input from the Office of Architectural Extension and the Department of Statistics has also been utilized. This University program has been established to foster natural energy design improvements, conduct energy performance and habitability research, and demonstrate and disseminate these critical findings to building design professionals and the public.

The emphasis on habitability aspects recognizes that modification of the quality of design will affect the quality of life in more than the aesthetic sense. Consumer attitudes and public opinion toward energy and alternative housing modes have, in fact, been the subject of a number of recent studies (1, 2, 3, 4). Also, detailed earth sheltered design guidelines have been published for several years, although habitability aspects have generally not received extensive treatment (5, 6, 7, 8, 9).

Potential Energy Savings

Energy considerations in passive design have received much attention and also much speculation. In fact, preliminary Oklahoma survey results indicate that energy savings is the primary incentive for the earth sheltered structures built within the past 5 years, replacing tornado and windstorm concerns of earlier times.

A preliminary review of the raw data indicates that earth sheltered designs can reduce energy consumption by 50 to 75 percent compared with conventional above ground designs. However, level of lifestyle can exert a substantial influence on these potential savings. Interior design, finish treatments, insulation, thermal mass, and space planning aspects can also have a significant impact.

PROCEDURE

Questionnaire Development

All seven members of the project team participated in development of the final 17-page questionnaire. Two distinct classes of information were requested - physical parameters and attitudinal parameters. Physical parameters included construction and energy consumption, as well as some physical aspects of comfort and habitability. Attitudinal parameters included public acceptance, lifestyle, and occupant evaluations of comfort, habitability and energy performance.

The first contact with dwelling occupants was an extended telephone call to ascertain basic construction parameters and project location and completion data. These preliminary data formed a critical initial project record for each respondent which was immediately placed into computer storage. At this time (mid-summer 1979) virtually all of the six dozen respondents contacted agreed to participate in a subsequent questionnaire program. These participants were initially identified through the efforts of the Office of Architectural Extension over a 3-year period. Thus, the survey sample represents all readily identifiable earth sheltered dwellings in the State of Oklahoma.

A preliminary questionnaire was initially pilot tested in June 1979 by the project team in a near-by earth sheltered residence. In early July 1979 a national pilot test program was conducted with a sample size of 27 projects which were located in 13 states throughout the nation. The return rate on this pilot study was nearly 50 percent. The final questionnaire was mailed to the Oklahoma sample of 70 projects in early August 1979.

Interim Status of Project

This paper is based on the analysis of selected data from 34 completed questionnaires which were received by mid-September 1979. Due to time constraints only a limited number of questions could be analyzed and only in a preliminary manner. The particular questions to be addressed here comprise approximately 25 percent of the questionnaire. Topics to be covered include insulation, thermal mass, special energy systems, energy consumption, thermal and lighting comfort, and various consumer attitudes. Topics not addressed in this interim project study are: occupant profiles, financial considerations, maintenance aspects, structural systems, and storm protection design. Also, ongoing interview and site measurement studies were not available for inclusion in this report.

It is anticipated that these interim results can be more fully developed and confirmed when more questionnaires are received and more in-depth study is completed. A final Phase I report is expected to be completed in early 1980. Detailed field instrumentation of selected samples is scheduled for Spring 1980 to be followed by a nationwide survey later in the year. Prototype demonstration structures and laboratory mock-ups are anticipated for 1981.

Methods of Analysis

A number of habitability and comfort parameters were rated on a Likert scale. Some items were simply rated on a nominal value scale. For both of these types of attitudinal measures a basic frequency analysis was conducted for which means and standard deviations are presented. A number of physical parameters were evaluated using various other techniques.

A preliminary effort was made to establish a composite value index for selected attitudinal parameters which is discussed in detail in the next section. It is anticipated that this index may be later utilized in dealing with various physical factors. Such an index should be useful both in evaluating proposed designs and in analysis of existing situations.

Energy consumption and performance factors were examined in terms of the single number composite measures of thermal mass of the building shell and annual energy usage per sq. ft. of floor area. These composite numbers were determined by the experimenters from construction data and utility data provided in the questionnaires.

In addition to providing opinions about attitudes, building occupants provided non-traditional input by serving as measuring instruments to quantify certain physical performance characteristics. At this early stage of the investigation, this procedure was selected as a viable alternative to extensive and expensive field instrumentation. It is anticipated that the above approach will be substantiated by ongoing interview and site measurement studies.

Since all questionnaire information was provided by building occupants-who typically were personally involved in design and construction of their own dwelling, a certain degree of bias is to be expected. In the words of one respondent: "after all, would I give my own design a bad rating?" As previously mentioned, tandem research efforts have been formulated to involve independent skilled observers and actual instrument measurements at

lected sites to provide substantiation of questionnaire data where appropriate.

An intuitive effort to counter the effect of occupant bias has been made by the experimenters. Prior to the present study, the researchers did visit more than two dozen earth sheltered structures over a three year period. This past experience has provided a context from which the validity of the various responses can be gaged. The more sensitive and controversial responses are not included here and will be fully studied only when other substantiating evidence is on hand for comparative analysis.

ANALYSIS AND DISCUSSION

Decision Process

Winter and Morris (10) state that housing adjustment behavior occurs when a family's housing produces a significant reduction in their satisfaction level. One of the major causes for adjustment or change in housing apparent today definitely relates to rising energy costs. This factor appears to have been very important to the decision process of the families studied in the present project.

Respondents were asked to rank several items regarding their decision to build an earth sheltered house on a scale of 1 to 10, with 1 representing unimportant and 10 meaning very important. The major reasons expressed for building an earth sheltered structure were energy related as shown in Table 1. Three items were ranked as most important: reduced heating, reduced cooling, and enhanced alternative energy potential. The respective mean ranking values were 8.0, 7.9, and 6.8. Another major item was related to storm protection which received a mean of 7.8. The means and standard deviations are recorded in the table for all twelve items suggested.

TABLE I

MEAN SCORES OF DECISIONS FOR BUILDING AN EARTH SHELTERED HOME

Factor	Mean	S.D.
Reduced Heating	8.0	1.6
Reduced Cooling	7.9	1.7
Storm Protection	7.8	2.0
Enhanced Alternative Energy Potential	6.8	2.9
Reduced Maintenance	5.6	2.9
Security from Vandalism	5.5	3.3
Insurance Reduction or Elimination	5.4	4.1
Personal Privacy	5.1	3.9
Demonstration or Experimentation	4.9	3.6
Environmental Noise Reduction	4.6	3.8
Improved Lifestyle	3.3	3.6
and Preservation	2.1	2.3

Habitability Parameters

The respondents were asked to rate how the overall habitability of their present living environment compares with their previous above ground dwellings. All respondents stated that the level of habitability of the earth sheltered dwelling was much better than their previous above ground dwelling.

Occupants were also asked to respond to a number of habitability parameters from two perspectives: the relative importance of the parameter from an ideal setting, and an appraisal of how their earth sheltered home rated, on a scale of 1 to 5. The specific habitability parameters of spatial arrangement, acoustical features, and lighting aspects are discussed. In general, respondents ranked the spatial arrangement as having the greatest importance as shown in Table II.

TABLE II

TOTAL MEAN RATINGS OF HABITABILITY PARAMETERS

Parameter	Total Mean for Relative Importance	Total Mean Rating for Own Home
Spatial Arrangement	3.9	3.8
Acoustical Features	3.3	3.8
Lighting Aspects	3.3	3.6

Individual ratings for seven items forming a subset of the spatial arrangement parameter are shown in Table III. The safety of the structure received the highest ratings, and flexibility of the interior space received the lowest ratings.

TABLE III

MEAN RATINGS OF SPATIAL ARRANGEMENT PARAMETERS

Parameter	Mean for Relative Importance	Mean Rating for Own Home
Safety of Structure	4.7	4.6
Arrangement of Rooms	3.9	4.2
Interior Storage Space	4.3	3.9
Sufficiency of Fire Exits	4.0	3.8
Accessibility to Bring in Groceries	3.7	3.7
Adequate Exterior Views	3.3	3.5
Interior Flexibility	3.1	2.9

Five acoustical features were evaluated. Generally the respondents rated the importance of these features lower than they rated their own home. This is the only set of parameters which indicated a reversal in ratings between the two tabulated columns. Although acoustics were not rated as highly important, occupants were quite satisfied with their existing acoustic environment. However, in an entirely separate question many respondents indicated that the environment was much too quiet, causing difficulty in sleeping during the initial periods of occupancy.

TABLE IV

MEAN RATINGS OF ACOUSTICAL PARAMETERS

Parameter	Mean for Relative Importance	Mean Rating for Own Home
Elimination of Outside Noise	3.7	4.3
Elimination of Echo Sounds	3.4	4.1
Control of Plumbing Noises	3.2	3.7
Privacy of Family Members	3.0	3.7
Control of Mechanical Noises	3.2	3.4

The lighting parameters consisted of five items. The item of greatest importance was natural lighting design as shown in Table V.

TABLE V

MEAN RATINGS OF LIGHTING PARAMETERS

Parameter	Mean for Relative Importance	Mean Rating for Own Home
Uniformity of Daylighting	3.4	3.9
Elimination of Contrast	3.3	3.9
Arrangement, Color & Style of Artificial Lighting	3.3	3.4
Natural Lighting Design	3.6	3.3
Artificial Lighting Design	3.1	3.3

A composite index formulated for later detailed study is presented here for the previous tabulated data by way of example only. This index is computed by multiplying 4 times the product of the mean importance and mean home ratings. For example, in Table III "safety of structure" would have a rating of $4(4.7)(4.6) = 86.5$ on a possible 100-point scale. Similarly, from Tables IV and V "elimination of outside noise" and "uniformity of daylighting" would have composite indexes of 63.6 and 53.4, respectively. It is not yet clear what, if any, meaning can be attached to cross-product index values of less than about 50. Certainly very low values would indicate insignificance of those particular parameters for design.

Lifestyle Parameters

Attitudes toward changes in lifestyle were also probed. Respondents were asked to rate the importance of several lifestyle parameters on a scale of 1 (unimportant) to 5 (very important) for a perceived ideal situation. Each of the parameters was then rated by the respondent for their own dwelling as part of their personal daily routine. All of the parameters were related to energy reduction methods used in the home.

The respondents ranked each parameter as being more important than they ranked the actual situation in their earth sheltered home. The item receiving the highest rating both in importance and in the actual living environment was related to turning off lights when not in use. The parameter of making a conscious effort to change or modify work patterns in order to save energy had the widest spread between respondents' ranking of importance and their daily routine ranking. The findings for five suggested parameters are shown in Table VI.

TABLE VI

MEAN RATINGS OF LIFESTYLE PARAMETERS

Parameter	Mean for Relative Importance	Mean Rating for Daily Routine
Turn off lights when not in use	3.8	3.5
Adjust type of clothing	3.4	3.0
Shut off spaces when not in use	2.6	2.3
Modify work patterns	3.2	2.1
More fully utilize most comfortable spaces	2.6	2.1

The respondents were also asked to rate how their routine had changed since moving into their earth sheltered home. Nearly 86 percent stated that their routine was about the same with very little change. Of those responding only 14 percent had modified their behavior to save energy.

Energy Expectations

Respondents were asked to rate how their energy consumption in earth sheltered housing compared with their expectations. Of those responding, 44 percent stated that their energy consumption was much higher than expected. About 36 percent stated their consumption was the same as expected, with about 20 percent stating their energy consumption was much lower than expected.

Energy Performance

Special energy system installations were reported in about one-third of the cases. Approximately 10 percent of the total sample utilize solar assisted domestic hot water heaters. Another 10 percent report attached greenhouses. About 25 percent deliberately utilize windows for passive solar space heating.

Conventional heating systems are installed in about one-half of the projects and the remainder utilize heat pump systems. Roughly two-thirds of the total sample have either a wood stove or a fireplace as an additional heat source. Including the heat pump installations, mechanical space cooling systems are provided in 80 percent of the projects. A full 20 percent have no mechanical cooling whatsoever.

Building energy performance was ascertained by two methods. Occupant evaluations indicate that a full 40 percent of the total space conditioning

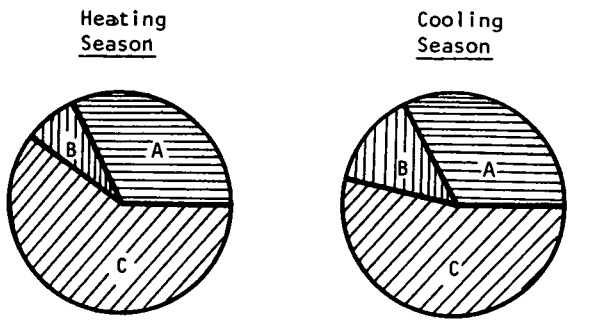


Fig. 2 Occupant Evaluations of Environmental Control Energy Sources

energy needs of the building is being provided by alternative or passive systems. As shown in Figure 2, the occupants perceived mechanical space conditioning to account for about 60 percent of the total environmental control requirement in both the heating and cooling seasons. In winter, wood accounted for about 33 percent of the heating total, and in summer earth heat sink effects account for about 30 percent of the cooling total. With respect to lighting energy performance, the mean estimate of the occupants was that about half of the needed lighting energy is provided by daylighting.

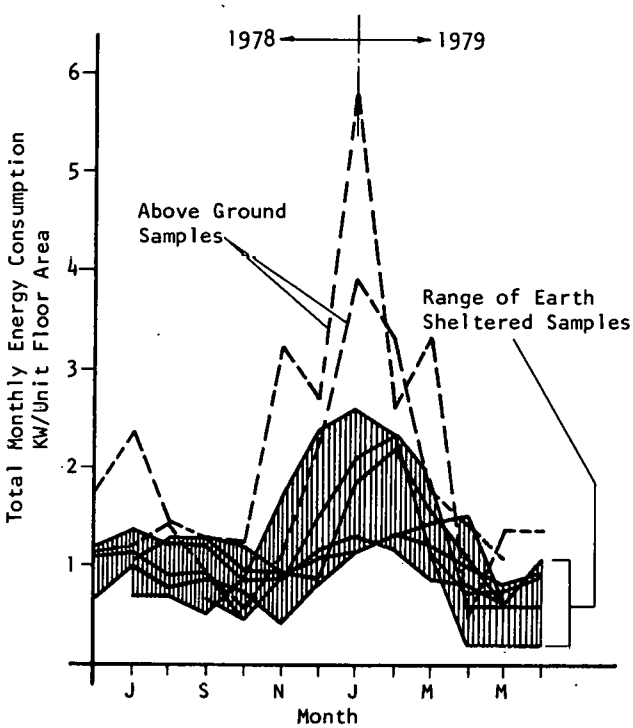


Fig. 3 Annual Energy Consumption Comparison

Actual energy consumption was obtained with the use of utility billing records. Figure 3 shows the monthly energy use for seven projects over a one year period. The data have been plotted in Btu per sq. ft. of occupied living space. For comparison, two contemporary above ground dwellings are represented on the same plot. Although the analysis of energy utilization patterns is extremely complex, Figure 3 does provide preliminary verification of the energy expectations and perceptions of the respondents. That is, their anticipated levels of energy savings generally have been realized, and the savings are substantial compared to contemporary above ground housing.

Of considerable interest is the finding that this substantial energy savings has been achieved with little decrease, and often an increase, in the comfort, livability, and habitability aspects in earth sheltered dwellings. In fact, actual improvement in lifestyle appears to be an attribute sought by some people who have decided to live in earth sheltered structures.

A majority of questionnaire respondents have realized their expected level of energy savings, however, many stated that consumption was much higher than expected. Although it is difficult to predict why the expected savings were not achieved in these cases, there are several factors which may have contributed to this situation. Over-expectation could, of course, be one such factor. In addition, several non-optimum design features became apparent to the experimenters during reviews of the questionnaires. As shown in Figure 4, the average installed mass of the identified projects far exceeds that of a standard above ground house. Unfortunately, the full potential of this installed mass is often not realized, as when furred wall or ceiling treatments are used.

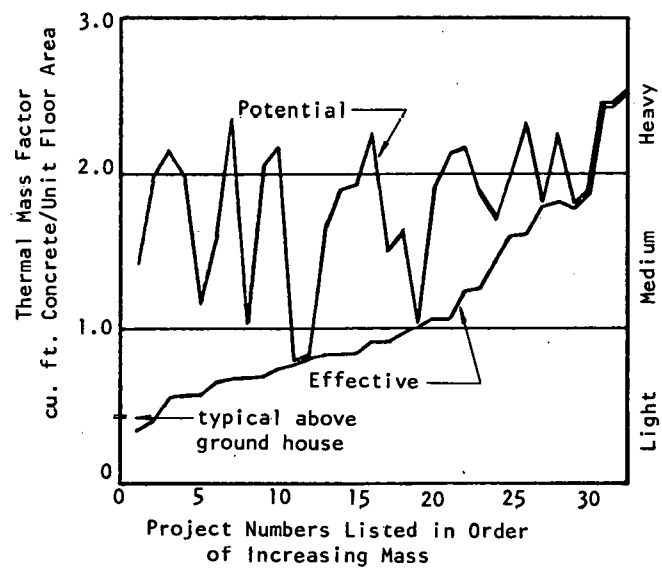


Fig. 4 Effective vs. Potential Thermal Mass

Figure 4 shows that while most projects realize roughly twice the effective thermal mass of an above ground house, the potential thermal mass may be several magnitudes higher even disregarding the mass of the earth surrounding these structures. The efficient utilization of installed thermal mass can have a drastic impact on energy utilization as well as on habitability.

Design Implications

Although this paper is an interim status report, a few important design implications have been identified based upon preliminary findings and analysis. Perhaps the most visible design implication is the lack of a systems approach to earth sheltered design. Although the standard discipline-by-discipline approach to design has resulted in an earth sheltered housing stock which by all indications is more successful than average, it is apparent that, in general, the earth sheltered structures surveyed do not represent an optimization of the idiom. It is the opinion of the researchers that the full utilization of the potential of earth sheltered housing will only be realized by application of a systems approach to building design.

The under-utilization of thermal mass demonstrated in Figure 4 is an example of the potential for more optimized design. The benefits associated with the enhanced thermal mass characteristic of earth sheltered housing are many. However, the number and diversity of design decisions impacting availability of this thermal mass are equally numerous.

Habitability aspects of earth sheltered housing are improved by the temperature stability inherent in massive structures. Energy consumption is modulated by capacitive insulation effects associated with massive materials. The potential for economically feasible active and passive solar systems could be improved by the thermal storage capacity found in earth sheltered buildings. The comment of one occupant that "there is very little difference in temperature when we return from work after the heating and cooling has been turned off all day" tends to indicate that the need for expensive backup systems may well be reduced or eliminated by earth sheltered construction. On the other hand, numerous design decisions strongly influence which of these potential benefits may accrue to the homeowner. The selection and installation of interior finishing materials, and selection and application of insulation and waterproofing systems each provide an occasion for decoupling of available mass from the occupied environment. Spatial arrangement and allocation decisions influence the positioning and viability of installed mass and earth contact potentials. A number of other earth sheltered design factors relating particularly to Oklahoma and the Southwest are addressed in several recently published accounts (9, 11, 12, 13). Opportunities for design professionals in the areas of field inspection and post-occupancy evaluation should represent a logical expansion of services currently offered by individuals involved with earth sheltered structures (14).

CONCLUSIONS

Earth sheltered housing presents the potential for substantial energy savings coupled with enhanced alternative energy installation opportunities. Preliminary studies indicate that the habitability and livability aspects of earth sheltered structures offer a viable housing alternative in an age of increasing energy scarcity and evolving comfort expectations. Continued detailed evaluations of both individual and systems parameters associated with earth sheltered design and construction practices should be actively pursued.

ACKNOWLEDGMENTS

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NOTES

SOLAR ANALYSIS BASED ON UTILITY BILLS

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ABSTRACT

The Utility Bill Analysis of solar projects is discussed for a selected number of sites in the HUD Solar Program. We point out what kinds of information can be reliably obtained from the Utility Bill Analysis of solar projects and what are the limitations of this procedure. We further discuss how such an analysis may be supplemented and extended in order to make it an additional useful tool for the analysis of solar system performance.

I. INTRODUCTION

The Utility Bill Analysis is a procedure for using information from the utility bills for estimating or verifying the energy savings in selected solar systems in the HUD Residential Solar Demonstration Program. This is done by a comparative study of utility bills and load calculations. The utility bills from a solar house during the months of solar operation are compared with past utility bills, with a comparison building's utility bills when possible, and with calculated results for loads and solar participation. In all, five different methods of analysis are used. The results are used to obtain an average value of annual energy savings and the solar participation factor for the year, and are compared with f-chart predictions for the system.

The results of the analysis are discussed in detail for a selected example of an instrumented site in the HUD solar program. We point out the merits and the limitations of the Utility Bill Analysis Procedure for obtaining information about solar projects. We further discuss how such an analysis may be supplemented and extended to make it an additional useful tool for the analysis of solar system performance.

The analysis is based on a comparative study of results obtained using 5 different methods.

A. Dubin-Bloome Associates (DBA) analysis of site utility bills.

B. F-Chart calculations with the 30 year average weather data, and with the solar contribution broken up into the space heating and DHW components.

F-Chart calculations with weather data for the
r in question.

D. Measured data for the instrumented projects (monitored by IBM), from the monthly performance reports.

E. Use of data from a 'comparison site' without the solar system.

The results obtained by using these methods and comparing them will be discussed for a selected instrumented project in the HUD Solar Program.

II DISCUSSION OF THE DBA UTILITY BILL ANALYSIS METHOD

For each site, a one year period was selected during which utility bills, weather data, life style data from the occupants, and other necessary data could be obtained. The comparative study was then used for estimating the yearly energy savings and the annual solar fraction.

The calculations made by us at Dubin-Bloome Associates (DBA) consists of the following:
a. Utility bills were obtained from the utility companies
b. Space heating loads and the DHW load were calculated using the load calculation methods used with the f-chart program, with the space heating load calculated using degree days. Cooling loads were calculated manually.

c. $$\text{Usable Heating Energy} = \left[\text{Energy Charged in the Utility Bill}\right] \times \left[\text{Burner Efficiencies}\right]$$

$$\text{Usable Cooling Energy} = \left[\text{Usable Heat Used for Cooling}\right] \times \left[\text{The Appropriate C.O.P.}\right]$$

for air conditioners that use heat energy input.

$$\text{Solar Contribution} = \text{Total Load} - \text{Usable Heating Or Cooling Energy}$$

$$\text{Solar Savings (Dollars)} = \left[\text{Solar Contribution in BTU}\right] \times \left[\text{Billing Rate per BTU}\right]$$

d. The DBA analysis procedure takes the utility bill (which is generally for the different loads combined), and separates out the relevant load component (heating, hot water, or cooling) using an empirical "kilowatt hour usage factor" (KWHUF). The latter is the sum of billable electrical energy used in the house for all applications other than heating, cooling or hot water, and was determined by interviews with occupants about their life style, along with the Utility Company's estimate of appliance energy usage.

When the data were insufficient, an average value of 650 kWh/month was used for a single family residence.

e. When the KWHUF was used, we have

$$\text{Solar Contribution} = \left[\text{Heating, DHW and Cooling Load}\right] - \left[\text{Energy Usage from Utility Bill}\right] - \text{KWHUF}$$

III. OTHER ANALYSIS METHODS

1. f-Chart Calculations

These have been done in two forms:
(a) Using the 30-year average monthly ambient temperature, heating degree days and horizontal solar radiation.
(b) Using updated monthly weater data from NOAA for the actual year in question (rather than 30-year average weather data).

In each calculation, the f-Chart method is used for obtaining the solar fraction, which when multiplied by the total load gives the solar energy used. The solar contribution in the f-Chart method is broken up into space heating and DHW components.

2. Measured Data from Instrumented Sites

For the instrumented sites, the measurements have been collected together into Monthly Performance Reports. The loads, solar contribution and solar fraction are taken from these reports. In using these, it should be noted that the load, solar fraction, etc. may be defined differently in these reports from the definitions commonly used, so that any comparison must be made with care.

3. The Comparison Site Method

A comparison site is selected which is similar to the instrumented site, the utility bills for both sites are obtained from the utility companies.

$$\text{Solar Savings} = \left[\text{Comparison Site Energy Usage}\right] - \left[\text{Instrumented Site Energy Usage}\right]$$

Solar Fraction = Solar Savings/Load

The value of this method depends on how closely comparable the two systems and houses are. Even if the building, equipment and appliances are equivalent, a difference in the life styles of the occupants can result in a large difference in the loads.

IV Example of Utility Bill Analysis of HUD Solar Projects

To illustrate the results of the Utility Bill Analysis and the comparative study, we shall discuss one example in detail. This is the A-Frame Industries site, which is a single family residence in Kaneohe, Hawaii, with a solar DHW system. The system has 77 sq. ft. of flat plate collectors, with a net aperature of 67 sq. ft. Potable city water is the heat transfer medium used; the storage tank is a 120 gallon tank.

No freeze protection is needed because of the warm climate. Auxiliary energy is provided by a 4 kW electrical heating element in the storage tank. The control system is a simple differential controller, which turns the collector loop pump on when the collector storage temperature difference is 5°F, and off when this difference is 2°F. The hot water design temperature is 104°F; the usage is taken to be 40 gallons per day. The yearly average supply water temperature is 74°F.

The results of the analysis of this solar system using different methods and comparing them are summarized in Tables 1 and 2.

There are small differences among the values of the DHW load values obtained using the different methods; however, they are small enough to be ignored.

Table 1 Analysis of the A-Frame Solar System Using Different Methods
Electricity Price Assumed = $0.04/kWh, 1 MBTU = 1 Million BTU

Month	KWHUF (MBTU)	Calculated DHW Load (MBTU)	Utility Bill (Elec) (MBTU)	DBA Analysis (With 1978 Weather Data) Solar Contribution (MBTU)	$ Solar Savings (DOLLARS)	Solar Fraction %	F-Chart Calculations - I Solar Contribution (MBTU)	$ Solar Savings (DOLLARS)	Solar Fraction %
Jan.	1.55	0.68	1.62	0.61	$ 7.15	89	0.68	$ 8.16	100
Feb.	1.55	0.62	1.62	0.55	6.45	89	0.66	7.92	100
March	1.55	0.68	1.50	0.68	8.76	100	0.68	8.16	100
April	1.55	0.66	1.50	0.66	8.52	100	0.62	7.44	100
May	1.55	0.68	1.42	0.68	8.76	100	0.68	8.16	100
June	1.55	0.66	1.42	0.66	8.52	100	0.66	7.92	100
July	1.55	0.68	1.49	0.68	8.76	100	0.68	8.16	100
Aug.	1.55	0.68	1.49	0.68	9.00	100	0.66	7.92	100
Sept.	1.55	0.66	1.48	0.66	8.52	100	0.68	8.16	100
Oct.	1.55	0.68	1.48	0.68	8.76	100	0.68	8.16	100
Nov.	1.55	0.66	1.67	0.54	6.33	82	0.66	7.92	
Dec.	1.55	0.68	1.67	0.56	6.56	82	0.68	8.16	100
Yearly Total	18.60	8.05	18.36	7.64	$96.09	95%	8.02	$96.24	100%
Yearly Average					8.01			8.04	

Table 1 (Continued)

Month	F-Chart Calculations With Weather Data Solar Contribution (MBTU)	Solar Savings (DOLLARS)	Solar Fraction %	Data From Measurements DHW Load (MBTU)	Solar Contribution (MBTU)	Solar Savings (DOLLARS)	Solar Fraction %
Jan.	0.60	7.20	88	*	*	*	*
Feb.	0.55	6.60	89	1.01	1.01	11.83	100
Mar.	0.62	7.44	90	1.13	1.13	13.24	100
April	0.59	7.08	90	0.85	0.85	9.96	100
May	0.62	4.44	91	*	*	*	*
June	0.61	7.32	93	0.82	0.82	9.61	100
July	0.63	7.56	92	0.88	0.88	10.31	100
Aug.	0.64	7.68	94	*	*	*	*
Sept.	0.64	7.68	96	0.95	0.95	11.13	100
Oct.	0.64	7.68	94	*	*	*	*
Nov.	0.57	6.84	86	*	*	*	*
Dec.	0.57	6.84	83	*	*	*	*
Yearly Total	7.29	$84.36	91 %	5.64	5.64	$66.08	100%
Yearly Average		$ 7.03		0.94	0.94	$11.02	

NOTE: * Indicates data not available

Table 2 Comparison of Instrumented Site and Comparison Site

Month	Utility Bill: Energy Usage (MBTU) Instr. Site	Utility Bill: Energy Usage (MBTU) Comp. Site	Utility Bill: $ Charges Instr. Site	Utility Bill: $ Charges Comp. Site	DHW Load (MBTU)	Saolar Savings: Energy Savings (MBTU)	Saolar Savings: $ Savings	Solar Fraction %
Jan.	1.50	2.66	17.58	31.18	0.68	1.16	13.60	100
Feb.	1.50	2.66	17.58	31.18	0.62	1.16	13.60	100
Mar.	1.42	2.25	16.64	26.37	0.68	0.83	9.73	100
Apr.	1.42	2.25	16.64	26.37	0.66	0.83	9.73	100
May	1.49	2.45	17.46	28.71	0.68	0.96	11.25	100
Jun.	1.49	2.45	17.46	28.71	0.66	0.96	11.25	100
Jul.	1.48	2.41	17.36	28.46	0.68	0.93	10.90	100
Aug.	1.48	2.41	17.36	28.46	0.68	0.93	10.90	100
Sept.	1.67	*	19.57	*	0.66	*	*	*
Oct.	1.67	*	19.57	*	0.68	*	*	*
Nov.	1.62	2.70	18.98	31.64	0.66	1.08	12.65	100
Dec.	1.62	2.70	18.98	31.64	0.68	1.08	12.65	100
Yearly Total	18.36	24.94	215.18	292.72	8.05	9.92	116.26	
Yearly Average								100%

NOTE: * Indicates data not available

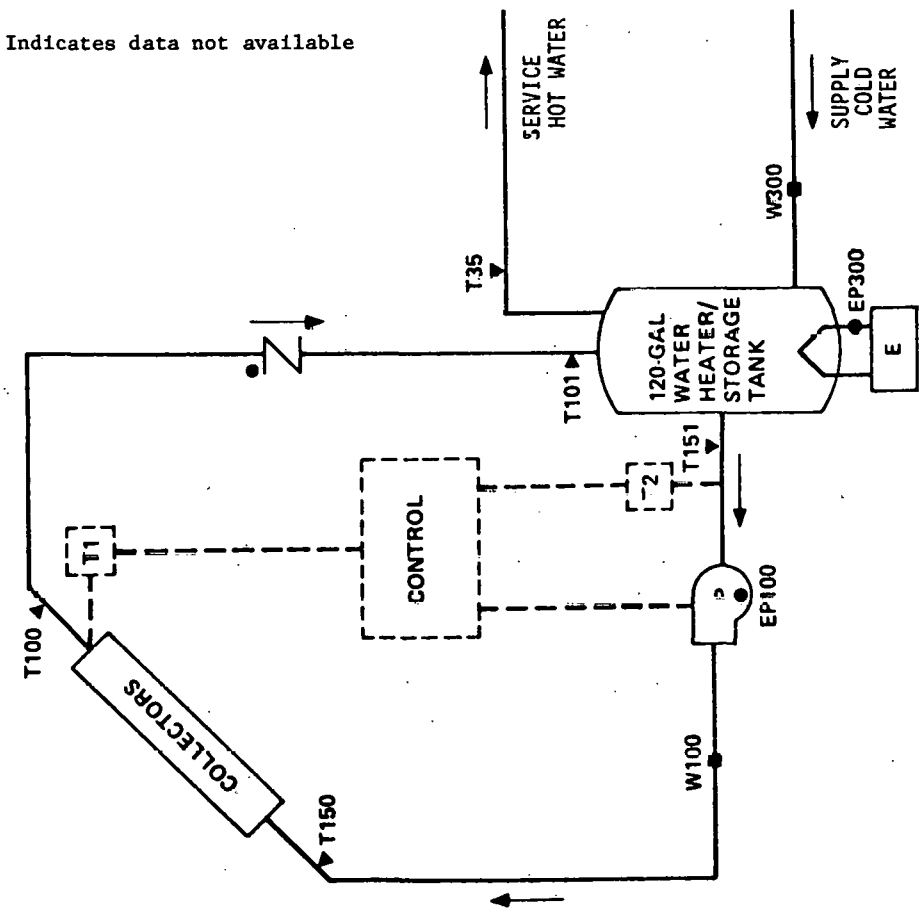


Figure 1. A-Frame Industries Solar Energy System Schematic

DISCUSSION

For this site, the five analyses are in close agreement in the value predicted for the solar fraction. The comparison of the instrumented site with the comparison site is inconclusive. What is shown is a DHW load of 8.05 and a Solar Savings of 9.92. This arises from higher electrical usage in the comparison house, either because of a much higher DHW load or because of a higher KWHUF. This illustrates the uncertainty in the comparison site method. The missing information here is the living habits of the occupants of the house, which is probably responsible for the larger DHW load during certain months.

Note also that the load in the tables given here includes just the demand and not the losses.

V. CONCLUSIONS

The comparative study discussed here has been carried out for 5 instrumented solar projects and a few non-instrumented ones. Space restrictions have allowed us to discuss only one in any detail. For most of the other HUD Solar Projects that we have analyzed, the results obtained by the different methods are not in as good agreement with one another as the results for this project.

The reasons for the differences among the results obtained using different methods and the conclusions that follow from these studies are briefly summarized below.

1) The load calculations show up uncertainties that make proper comparison difficult. These include uncertainties in the (UA) values, in the infiltration loads, and in the correction factor in the degree day method for calculating the load. In some houses, the occupant has a different thermostat setting, e.g. 62°F rather than 68°F in the winter. Using the degree day based information may introduce large uncertainties in the loads here.

2) In some projects, the results of the f-Chart methods differ substantially from those obtained by other methods. The limitations of the f-Chart method make the interpretation of the results uncertain. A method such as the f-Chart does not see many details of a solar system that can give rise to important differences between different systems.

3) The unknown living habits of the occupants in many of the houses makes many of the comparisons difficult. Houses that are otherwise identical can have quite different energy usage because of the differences in the living habits.

4) The separation of the heating, DHW or cooling load from the total electrical load is often very uncertain. The kilowatt-hour usage factor values have been obtained by a method that can have large uncertainties.

5) The efficiency of equipment is often unknown, resulting in an uncertainty in the loads.

6) The somewhat different definitions of the load, solar fraction, etc. used by different sources of information make it difficult to compare them.

The comparative study carried out at Dubin-Bloome Associates shows that the Utility Bill Analysis can be a useful tool for supplementing information obtained by other methods on a solar project. To make it a really useful tool, the method needs to be extended in the following ways:

(a) The metering provided in arriving at the usual utility bills is very inadequate. A minimum requirement is separate metering of each component of the load, e.g. hot water, appliances, space heating, etc. Trying to separate the hot water load from a utility bill that includes all electrical usage is a very uncertain procedure. Separate metering of all equipment would make the method much more useful.

(b) A systematic survey of the living habits of the occupants of the house is needed in order to obtain an accurate picture of the energy usage. Without this, any analysis can have large errors. Without such information, the comparison between a solar and a conventional house may be quite meaningless.

(c) For adequate information about how much solar energy contributes to a house, additional measurements are recommended. One of these would be shutting down the solar system in a house during a specified period, and making measurements during this period. This can provide more reliable data than using a different 'comparison house'.

If the additional measurements and estimates indicated above are made, then the Utility Bill Analysis and the Comparative Study methods discussed in this paper can be used for providing much more reliable information on solar systems in houses.

NOTES

PROBLEMS AND QUESTIONS IN SOLAR ANALYSIS METHODOLOGY

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ABSTRACT

The evaluation of the solar projects in the HUD Solar Residential Demonstration Program has involved examining in detail a large variety of solar systems, and comparing the predicted thermal performance of these systems with the observed performance.

Several questions and problems arise in the course of doing this, regarding the validity of some of the methods of solar analysis used, as well as the meaning of some of the concepts. In this paper, we examine and discuss these questions.

INTRODUCTION

A large number of solar projects have been examined and evaluated in the HUD Solar Residential Demonstration Program. This has involved comparing the predicted thermal performance of a system with the observed thermal performance, and using different methods to obtain estimates of quantities such as the load, solar fraction, etc. The results of this work has raised questions regarding the validity of some of the solar analysis methods used and the meaning of some of the concepts.

In this paper we examine and discuss these questions, including the following:

a) Uncertainties in load calculations and in solar radiation calculations;
b) The f-Chart method and its limitations;
c) The solar fraction, and how it should be defined;
d) Uncontrolled heat losses, and their effect on solar analysis results;
e) Special questions in Passive Systems.

UNCERTAINTIES IN LOAD CALCULATIONS AND SOLAR RADIATION CALCULATIONS

Uncertainties in load calculations arise because of the following:

1. The degree-day method of calculating heat losses has limitations. In particular, the correction factors used with the modified degree-day method are uncertain. The results obtained by using the degree-day method can have uncertainties of up to 25% or 30%.

2. The design interior temperature used in a heating load calculation may be selected differently by different users (e.g. 65°F winter interior design temperature in the BEPS requirements vs 70°F or 68°F). This can give rise to a large difference in the calculated value of the loads.

   Using a different interior design temperature also causes an uncertainty in the use of the tabulated degree-days.

3. Differences between the long-term average data and current data, both for weather data and solar insolation data, can sometimes be large, and can then give rise to difficulties in interpreting the results. An example of the differences is shown in Table 1 for the weather data, and in Table 2 for the solar insolation data.

   Updates in the solar insolation data during the last few years have shown up fairly large errors in some of the data used earlier. Some of the data bases do not have the updated values, so that calculations using different data bases may give results differing by 10% to 30%.

4. Calculation of solar insolation values using calculated values of clear-sky insolation and multiplying it by the mean percentage of possible sunshine gives results somewhat different from those obtained by the Liu-Jordan method (e.g. as summarized in the HUD IMPS Manual). The latter is generally accepted as the more accurate method of estimating the solar radiation. However, the former is often used for quick estimates.

TABLE 1 Differences Between Long-Term Average and Current Weather Data

MONTH	HEATING DEGREE DAYS (°F-Day)		AVERAGE AMBIENT Temperature (°F)	
	Current	Long-Term Average	Current	Long-Term Average
December 77	961	1,004	34	32
January 78	1,302	1,088	23	30
February 78	1,064	902	27	32
March 78	713	868	42	37
April 78	480	525	49	48
Total	4,520	4,387	-	-

TABLE 2 Difference between Long-Term Average and Current Solar Insolation Data

MONTH	TOTAL INSOLATION At Collector Tilt = 30°		
	Current	Long-Term Average	Percent Difference
December 77	32,982	45,712	37%
January 78	32,224	51,773	60%
February 78	28,988	52,246	74%
March 78	47,001	62,304	30%
April 78	53,493	60,885	12%
Total	194,688	272,920	40%

5. When the site of interest is not in the solar data base, the solar radiation data have to be estimated from the data for the nearest site or a similar site. Since differences in cloud-cover can result in substantial differences between nearby sites, this introduces an uncertainty. This question has arisen during the evaluation of several HUD solar projects.

6. For cooling load calculations, use of the average daily temperature, as is sometimes done, can lead to incorrect results. It is necessary to use hourly temperature values over a day to get a reliable estimate.

7. The efficiency values required for an estimate of the equipment load are generally not known to any degree of accuracy. This can give rise to errors of 20% to 30%. If a boiler or burner is known to be regularly and carefully serviced, a better estimate of the efficiency can be made. Note that generally the seasonal efficiency is what is required.

   The efficiency and the load values depend on how the equipment is operated, and on details such as the temperature of the fresh air supply to the furnace, and the insulation on the pipes. Such details are often not available, leading to errors in the estimates.

8. The living habits of the occupants of a house can make a large difference in the value of the heating load of a house. A detailed life-style survey is required in order to obtain the necessary information. Without this information, the values of the load of two houses that are structurally similar cannot be properly compared.

9. Estimating space heating loads is subject to a few uncertainties.

   Firstly, even the transmission heating load calculation is subject to errors, mainly because of the uncertainty in the U-values, and the limitations of the approximate methods used in calculating the heat losses through attics, basements and floor slabs. Secondly, the infiltration heat losses are subject to much larger uncertainties; the uncertainties in the calculated values may be as high as 30% to 50%.

10. The term 'load' is used in a different sense by different users, which sometimes gives rise to a difficulty in interpretation. For example, in a hot water system, one user may define the load as the BTU used in heating the hot water, while another may define it as the energy to be supplied by the hot-water heating system. The latter 'load' will include the losses in the system.

    The space load, the equipment load, and the equipment energy requirements are sometimes not carefully distinguished. For passive solar systems, it should be carefully noted whether the load includes the heat loss through the solar glazing or not.

THE F-CHART METHOD AND ITS LIMITATIONS

The f-Chart method can be a useful method for estimating the long-term performance of a solar system, provided its limitations are recognized. Problems arise when the method is used outside its range of validity. The limits of the applicability of the f-Chart method are as follows.

1. The method is applicable only to certain types of solar systems. These types have been listed, e.g. in the HUD Intermediate Minimum Property Standards Supplement for Solar Systems. For solar domestic hot water systems, there is the further restriction that a certain type of load profile was assumed in deriving the f-Chart correlations. When the load profile is different, the f-Chart results may be in error.

2. The f-Chart method is concerned only with the overall average features of the system, and does not see many details of the system. Information such as the effect of storage sizing is included only approximately in terms of a set of correction factors. One should not expect details of the variation of the different parameters to be shown accurately by an f-Chart calculation. Also, effects depending on the user's lifestyle cannot be properly shown using the f-Chart.

3. The f-Chart method allows any sets of values of the average daily solar radiation to be used as the input. However, in a particular computer program, only a few limited options for the load calculation or the solar radiation input may be available. Shortcomings in the results arising from using, e.g. the degree-day method, to calculate the input loads in an f-Chart calculation should be distinguished from the limitations of the f-Chart method itself. This is often not done.

THE SOLAR FRACTION

The solar fraction was originally defined as the following ratio

$$\text{Solar Fraction} = \frac{\text{Solar Energy Used}}{\text{Load}}$$

To make this a useful definition, both the numerator and denominator must be defined carefully. Questions arise in defining both the numerator and the denominator; these are discussed below [1]

(a) In working with the early solar heating systems, the operating energy was often neglected in estimating the useful energy obtained from solar heat. Measurements showed, however, that the energy used for operating a solar energy system can be significant. The net energy gained by using the solar system is therefore equal to the following

$$\text{Net Energy Gained from Solar} = [\text{Solar Energy Used} - \text{Operating Energy}]$$

The solar fraction must therefore be redefined as the ratio

$$\text{Solar Fraction} = \frac{\text{Solar Energy Used} - \text{Operating Energy}}{\text{Load}}$$

(b) The load used in evaluating the solar fraction has been defined in different ways by different users.

In solar DHW systems, some users have defined the load as the energy used in heating the hot water. However, the energy supplied from solar heat is partly used in offsetting heat losses from the storage and from the piping between the storage and the load. These losses should be included in order to give the total heat required for supplying the hot water.

According as the load is defined to include the losses or not can make a significant difference in the value of the solar fraction.

In solar space heating systems, uncontrolled heat losses cause uncertainties in the magnitude of the load as well as the solar energy used. This is true when the heat losses from the solar system go into the living space. The solar fraction can be different, depending on whether this uncontrolled contribution of the solar input to space heating is treated as an effective reduction of the load or as a part of the solar energy used. Since the magnitude of the uncontrolled transfer of heat is generally not known with any degree of accuracy, there is also an uncertainty in the solar fraction in either method of defining it.

A somewhat different way of defining the load is the following:

Load = (Solar Energy Used + Auxiliary Energy Used)

This will include the losses between the solar storage and the load.

The solar fraction is then defined as follows:

$$\text{Solar Fraction} = \frac{[\text{Solar Energy Used} - \text{Operating Energy Used}]}{[\text{Solar Energy Used} + \text{Auxiliary Energy Used}]}$$

Special questions arise in defining the solar fraction for a passive or hybrid (active + passive) solar system [1].

Firstly, the solar energy used is more difficult to estimate accurately for a passive than an active system. The more accurate methods of calculation for most passive systems are complex enough to require a computer. Simple systems to which the solar-load-ratio method is directly applicable are among the few for which the results of hand calculations (or calculations with programmable calculators) can be regarded as reasonably accurate.

Secondly, the definition of the load is more uncertain and load estimates are more difficult to make accurately for most passive systems than for active systems.

An additional question that arises in passive systems is whether or not the load is defined as including the heat loss through the south-facing glazing which is used for solar collection. This can make a large difference in the value of the solar fraction when the glazing area is large. An example illustrating the questions arising in a passive system is given below.

Example of a Passive System

This is a single-family residence in Longmont, Colorado, with a passive solar space-heating system and a passive solar DHW system. This was an instrumented system, and the following performance data are quoted from a performance report by the IBM group, for the period November, 1978 through May, 1979. The results quoted here refer to space heating alone.

Building heating load (BHL) = 48.1 MBTU

(Note that this as given includes the daytime heat loss through the south-facing glazing.)

Internal Heat Gain (IHG) = 10.6 MBTU

Fireplace Heat Gain (FHG) = 1.6 MBTU

Space Heating Demand (SHD) = BHL - (IHG + SHD) = 35.9 MBTU

'Solar Fraction' of Load = 74%

'Solar Fraction' of Space Heating Demand = 100%

NOTE: The solar fraction as defined in the IBM report does not take the operating energy into account; i.e. Solar Fraction = (Solar Energy Used)/Load.

Using the IBM results, and taking the operating energy into account, the following are obtained.

Solar Energy Used = 35.6 MBTU
(This is defined as the Heating Load minus the Auxiliary Thermal Energy Used).

Operating Energy Used (for beadwall on south glazing) = 1.0 MBTU

Net Solar Energy Used =
35.6 - 1 = 34.6 MBTU

Note that the 'Solar Energy Used' includes all the solar energy entering through the glazing, without subtracting the daytime heat losses through the glazing, as is done for an active collector. The Building Heating Load and the Space Heating Demand both include the daytime heat loss through the south-facing glazing. As a result, the solar fraction is a larger number than it would be if the Solar Energy used and the heating load were defined as for an active system (i.e. if the heat loss through the south-glazing were subtracted from each).

Solar Fraction of Load (corrected for the Operating Energy) = 72%

Solar Fraction of Space Heating Demand = 96%

An estimate of the daytime heat loss through the south-facing glazing daytime heat loss through the south-facing glazing gives about 10.0 MBTU for the period November, 1978 through May, 1979.

Building Heating Load, excluding the daytime heat loss through the south glazing = 48.1 - 10.0 = 38.1 MBTU.

Space Heating Demand, excluding the daytime heat loss through the south-glazing = 35.9 - 10.0 = 25.9 MBTU.

Solar Energy Used, after subtracting the daytime heat loss through the south glazing = 35.6 - 10.0 = 25.6 MBTU

Operating Energy Used = 1.0 MBTU

With the above definitions of the load and the solar energy used, we obtain the following:

$$\text{Solar Fraction of Load} = \frac{25.6 - 1.0}{38.1} = 64.5\%$$

$$\text{Solar Fraction of Space Heating Demand} = \frac{25.6 - 1.0}{25.9} = 95\%$$

This shows how the two different definitions of the load and solar energy used lead to different values for the Solar Fraction. With the first definition, the solar fraction of the load is 72%, while with the second definition, it is 64.5%.

In using the Solar Load Ratio (SLR) method for passive calculations, the first definition is used. However, in comparing a passive system with an active system, the second definition should be used, since this would treat the passive and active systems on an equal footing. (i.e. the daytime heat loss through the collector surface is excluded from the Building Heat Load and from the Solar Energy Used).

UNCONTROLLED HEAT LOSSES

Uncontrolled heat losses create uncertainties in the analysis and interpretation of solar systems, since they cannot be properly estimated. They also give rise to increased cooling loads in the summer.

There are two types of uncontrolled heat loss mechanisms which have been important in solar systems:

1) Uncontrolled conduction losses, especially from storage, uninsulated piping, and heat exchangers.

2) Thermosiphon losses from the storage

Examples of the effects of uncontrolled heat losses in a solar system are given below.

EXAMPLE 1 This is a solar system in a single-family dwelling in Denver, Colorado. The system was designed for both space heating and DHW. The energy collected and delivered to different parts of the system are given below for the period April, 1978 - March,1979.

Solar Energy Incident = 255.1 MBTU

Solar Energy Collected = 79.9 MBTU

Energy Delivered to Storage Tank = 66.7 MBTU

Energy Removed from Storage = 25.6 MBTU

Energy Delivered to Space Heating and DHW Loads = 20.4 MBTU

Uncontrolled Energy Loss (into the Conditioned Space) = 57.1 MBTU

Space Heating Load = 27.5 MBTU

Solar Energy Used in Space-Heating Subsystem = 9.1 mBTU

DHW Load = 20.7 MBTU

Solar Energy Used in DHW Subsystem = 11.3 MBTU

Total System Load = 48.2 MBTU

Solar Energy Used = 20.4 MBTU

Operating Energy in Collection and Storage Subsystem = 3.4 MBTU

SF1 = (Solar Energy Used - Operating Energy)/Load = 35%

SF2 = Solar Energy Used/Load = 42%

Expected Solar Energy Used (from an f-Chart calculation) = 32.4 MBTU.

Maximum Possible Value of SF1 (if heat losses are minimized) = 60%

Maximum Possible Value of SF2 (if heat losses are minimized) = 67%.

The major part of the uncontrolled energy loss is attributed to heat loss from the storage tank. The rest is attributed to losses from the piping and heat exchangers. The heat loss from the storage during the period April 1978 to March 1979 is 41.1 MBTU, as compared to 66.7 MBTU for the energy delivered to the storage tank and 79.9 MBTU for the solar energy collected.

If the heat losses were minimized, the solar fraction for the total system (space-heating and DHW) could be increased from about 35% to about 60%. This takes into account the measured operating energy usage of 3.4 MBTU.

The uncontrolled energy losses also resulted in an additional cooling load of about 27.3 MBTU during the period May 1978 through September 1978.

EXAMPLE 2

This is an instrumented solar system in a single-family residence in Escondido, California. Solar energy is used for both space heating and DHW.

The energy collected and delivered to different parts of the system are given below for the period October 1978 to March 1979.

Solar Energy Incident = 53.4 MBTU

Solar Energy Collected = 14.9 MBTU

Energy delivered to Storage Tank = 8.0 MBTU

Energy removed from Storage = 4.7 MBTU

Energy delivered to Space-Heating and DHW Loads = 6.1 MBTU

Uncontrolled Energy Losses = 8.8 MBTU

Total System Load (Space heating + DHW) = 13.4 MBTU

Solar Energy Used = 6.1 MBTU

Operating Energy in Collection and Storage Subsystem = 0.45 MBTU

SF1 = (Solar Energy Used - Operating Energy)/ Load = 42%

SF2 = Solar Energy Used/Load = 46%

Expected Solar Energy Used (from an f-Chart calculation, with no heat losses) = 12.3 MBTU

Maximum possible value of SF1 (if heat losses are minimized) = 88%

Maximum possible balue of SF2 (if heat losses are minimized) = 92%

Of the total uncontrolled energy losses of 8.8 MBTU, 3.35 MBTU were lost from storage, and the rest from the transport system.

The above systems are examples of large uncontrolled conduction heat losses from the storage and transport systems. There are other systems in which the thermosiphoning heat losses from the storage to the collector have caused problems. In one liquid system, the liquid in the collectors cooled, and flowed back through the collector inlet into the collector-storage heat exchanger, until the water on the other side of the heat exchanger froze. Examples are also known in air systems where thermosiphoning heat losses from a rock storage bin have caused substantial heat losses. The use of a check valve or a backdraft damper can remove these problems.

CONCLUSIONS

In this paper we have discussed some of the main questions arising in solar analysis methods, which can result in errors or difficulties of interpretation. This discussion shows that the numerical results obtained by analyzing solar systems may be uncertain by a fairly large percentage (e.g 30% to 50%), and that care is needed in interpreting them.

We have also pointed out that quantities such as the load, the solar energy used, and the solar fraction need to be defined carefully, especially for passive systems, and that care is needed in comparing these quantities as defined by different users.

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1. Similar questions relating to the solar fraction have also been discussed by Larry Palmiter in publications from the National Center for Appropriate Technology, Butte, Montana.

PERFORMANCE OF THE SOLAR-ASSISTED HVAC SYSTEM FOR TERRASET ELEMENTARY SCHOOL WITH PASSIVE EARTH COVER

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ABSTRACT

Terraset Elementary School in Reston, Virginia, has now been in operation since February, 1977, serving 1000 children and faculty with a unique semi-underground construction, and floor plan totalling 69,000 sq. ft. on one floor.

This paper describes its construction, and the performance modes and characteristics of its HVAC system by which it has achieved an annual energy consumption of 8.75 Btu/ft^2-degree day, or $78.40 per ft^2 per 1000 degree days, which represent a reduction of 43% below the energy consumption of a conventional school of similar size in the same district.

A significant factor in this energy saving is the 36" of passive earth cover on 10" of reinforced concrete over the entire school, plus the surrounding 30° earth bank.

The performance, and some deficiencies in the design of the passive earth cover are reported, along with various improvements proposed for the entire system by which the second generation of schools of this type will show even greater savings as well as comfort standards for tomorrow's school children.

PRINCIPAL DESIGN CHARACTERISTICS OF TERRASET SCHOOL

General

Terraset Elementary School at Reston, Va., 20 miles from Washington, serves 1000 students and faculty, and uniquely combines the benefits of both a passive earth cover and a solar-assisted HVAC system with thermal storage.

Designed by Davis, Smith & Carter, architects, in Fairfax County, it was opened for use in February, 1977. Detailed performance data have now been collected for 30 months both from special instrumentation and from the microprocessor output of the Sunkeeper control and instrumentation system.

The construction cost of the school was $3.68 million of which the HVAC system cost $393,000 and the solar system (with 4822 sq. ft. of collectors) cost $550,000.

The useful area of the school as shown in Fig. 1, is 69,100 sq. ft consisting of four 100 ft. diameter Learning Centers, a 3550 sq. ft. gymnasium, a 7600 sq. ft. multi-purpose area, and 6500 sq. ft. in halls and administrative offices. In addition, there are three mechanical rooms.

All of this is covered overhead with a bed of earth and gravel to an average depth of 36 inches, and on the sides with a 30° earth berm. Each of the four ning Centers receives daylight through 650 sq. of single-pane glass panels in a 125° arc looking through the earth berm. The administrative offices and multipurpose room also looks out on a depressed front courtyard, 60' x 86', under the high space frame carrying the 176 solar collector panels (Owens-Illinois SUNPAK).

The HVAC and Solar-assist System (See Fig. 2 and Ref. 4)

The HVAC system, with Honeywell electro-pneumatic controls, was designed to serve the six Air Handling Units (AHU) in three basic modes, (1) warmup, (2) space heating, and (3) space cooling. Four sources of heat are possible, in economic order as follows: (1) solar hot water; (2) stored hot water, from excess previous solar heat or from heat previously reclaimed from the condensers of the four 372,000 B/hr vapor-compression chillers; (3) direct application of heat reclaim from the chillers operated as heat pumps on low-temperature stored hot water; and (4) hot water from a 480 KW electric boiler.

Similarly the four sources of cooling, in order of use, are as follows, (1) chilled water from the evaporators of the vapor-compression chillers, concurrent with heat reclaim to storage; (2) outside air ventilation; (3) chilled water storage; and (4) solar water cooling by means of two 300,000 B/hr absorption chillers (only possible when solar

hot water exceeds 200°F).

In each mode of operation, the control system logic was designed to commence heating or cooling (as desired) with the best source available, and switch successively to the less desirable energy sources as needed.

Four 26,000 cfm air handling units serve the learning centers with supply and return fans in each. These were designed to supply air through ceiling diffusers at about 60 F (winter and summer) and retrieve it at 70 F from the plenum between the false ceiling and the reinforced concrete roof. A typical air handling unit is shown schematically in Fig. 3.

Stored hot and chilled water for the HVAC system is kept in the three 8200 gallon (31 cu. m) insulated tanks shown in Fig. 2. In winter, all three store hot water: in summer, one stores hot and two store chilled water.

The separate solar system stores hot water from the collectors in two 2500 gal. tanks. These tanks supply heat to the HVAC water loop through the solar heat exchanger, and supply cooling through the two absorption chillers, as shown in Fig. 2, and reported in Ref. 4.

Passive Earth Cover

The passive earth cover over the entire school (except for the 20' x 20' skylight over the Media Center) is made up of four thermal barriers and two moisture barriers from top to bottom as follows: 30" of sodded clay, a layer of straw, 6" of gravel, a 1.5" thick polystyrene bead-board, and a .064" bituthane film on the 10" reinforced concrete roof. This is described in more detail in the report at the Kansas City Conference, Ref. 6.

The thermal performance of this cover is described below.

OVERALL PERFORMANCE COMPARISON BETWEEN TERRASET AND HUNTERS WOODS SCHOOLS

For the two school years July 1977 through June 1979 the energy consumption and electric bill of Terraset was compared with Hunters Woods, a conventional all-electric school in the same district. Results are as follows:

	Hunters Woods 80,254 sq. ft.	
	1977-78	1978-79
Heating deg. days (65F base)	5324	5137
Mwh consumed*	2213	1872
Specific heat consumption (Btu/ft^2-deg-day)	17.7	15.5
Electric bill	$67,423	$56,498

*includes energy for summer cooling

	Terraset 69,100 sq. ft.	
	1977-78	1978-79
Heating deg. days (65F base)	5324	5137
Mwh consumed*	1387	908
Specific heat consumption (Btu/ft^2-deg-day)	12.9	8.75
Electric bill	$42,242	$27,820

*includes energy for summer cooling

The improved performance at Hunters Woods from 1977-78 to 1978-79 was largely due to energy conservation measures introduced. The even greater improvement at Terraset was due both to conservation measures (e.g. reduced lighting) and to better control of the HVAC and solar system learned from the previous year's experience.

PERFORMANCE OF THE SOLAR SYSTEM IN SUPPLYING HEAT AND COOLING

General

A more detailed flow diagram of the solar assist system is shown in Fig. 4. Temperature sensors in the SUNKEEPER control system are shown on the diagram as T124 at solar collector exit, T120 at solar collector inlet, etc. Two flow sensors are employed as shown, one at the LTTST return to the two pumps SCP-1,-2 which circulate the collectors, and one at the HTTST outflow to the two pumps, CP-1, 2, which supply solar water to the "solar converter" for heating the building, or to the absorption chillers for cooling the building.

"Digital" outputs in the Sunkeeper printout show what fraction of each hour these pumps operate. With this information it is possible to determine the periods when a) there was no circulation through the collectors, b) heat was being added to the collectors to keep them from freezing (e.g. on cold winter nights), c) heat was being removed and applied to heating the building, d) heat was being removed and applied to thermal storage in the building, e) heat was being removed and applied to cool the building.

Four periods were studied in detail:
Sun-Fri, April 9-14, 1978
Sun-Thurs, August 13-17, 1978
Sat-Wed, December 16-20, 1978
Wed-Sun, January 10-14, 1979.

Rather than burden the reader with the reams of data involved, and the many changes in HVAC control philosophy tried both before and during this 9-month period, we may summarize the economic logic that now operates in applying solar heat to the building:

1) With negligible insolation, T124 less than 50F. and ambient less than 32F, heat is supplied t the collectors to keep them from freezing.

With T124 between 50F and about 80F, solar heat is stored in HTTST and LTTST but not applied to the building;

3) With T124 between 80F and 120F, and insolation during daylight hours, solar heat is delivered to any of the three hot storage tanks below solar water temperature. These tanks are then available as heat sources by which the Carrier units may heat the building by reclaim heat from their condenser bundles.

4) With T124 above 120F, solar heat is delivered through the solar converter either to heat the building directly, or into thermal storage, for later heating.

5) With T124 above 195F, the hot storage tank up to temperature, and the AHU's calling for cooling, solar water is delivered to the absorption chillers to cool the building.

No representative fraction of time or energy in each of these modes has yet been determined. However, gross "end-use efficiency" defined as (energy transferred to building)/(total insolation on collectors) is now about .40 in winter, and somewhat less in summer due to the need for higher temperature solar water in summer.

During the winter when the school is in session the collectors must be operated above 130F (54C) so that solar heat can be transferred to the hot hydronic loop maintained as high as 128F (53C). This same hot water heats the supply air and provides for convective heating under the peripheral windows. Any excess heat from the hydronic loop is transferred to storage. The winter results reflect typical operation where solar energy is used as a direct energy source.

When school is let out in mid-afternoon and on weekends, school temperature is set back and the hydronic loop can be used to heat the designated hot storage tanks and to provide hot water for maintenance, and morning warm-up. The collectors can then operate at lower temperature and heat the water in the hydronic storage tanks which can serve either for direct heating or as a heat source for operating the air conditioning unit as a heat pump (under false load). During Christmas vacation, 1978, for nine sunny days the school was operated on "set-back" temperature with collector temperature maintained manually just above the storage tank temperature.

Range of Solar Collector Efficiency

The Owens-Illinois SSR SUNPAK collector used at Terraset has published efficiencies with a least-squares fit represented by:

$$\eta_c = .715 - 0.200\ (T_x - T_{amb})/I \pm .005$$

where T_x = solar water exit temperature, °F
T_{amb} = ambient air temperature, °F
I = in-plane insolation, B/hr-ft^2
*30° south-facing tilt, at 39° N. latitude.

The value of I at Terraset measured by the SUN-KEEPER pyranometer and amplifier circuit was calibrated against a standard pyranometer at the official D.O.E. test station at V.P.I. & S.U., Blacksburg, Va., on July 31/79.

Using this calibration, the actual range of operating collector efficiencies on two typical winter days, one bright and one cloudy was as follows:

Time hrs.	T_{amb} °F	I B/hr-ft^2	η_c
	Jan. 10/79 (Cloudy)		
8-9	21	59	.60
9-10	27	158	.66
10-11	28.7	81	.56
11-12	28	40	.43
12-13	28	30	.31
13-14	28.5	32	.31
14-15	28.5	26	.12
15-16	29.2	20	.07
16-17	29.6	14	≈0
	Jan. 15/79 (Clear)		
8-9	18.6	63	.609
9-10	19	165	.651
10-11	20	245	.654
11-12	21	298	.655
12-13	22	320	.65
13-14	22	310	.64
14-15	23	267	.63
15-16	22	144	.57
16-17	20	0	0

Weighted average values of collector efficiency:
for Jan. 10/79, η_c = .43
for Jan. 15/79, η_c = .63

PERFORMANCE OF THE PASSIVE EARTH COVER

A typical thermal resistance of the earth roof cover, winter and summer, is calculated as shown in Table A.

The upper air film is taken as 18 ft/sec in winter, 12 ft/sec in summer. The lower air film, in the ceiling plenum, is taken as 10 ft/sec. Layer B, the wet clay, is assumed 20% moist in winter, 10% in summer. Layer C, the gravel is assumed with 5% moisture at all times, since it tends to drain off into the side berms.

By reducing the thermal resistance to R = 5.4 (assuming the polystyrene bead-board to be crushed and ineffective), a winter heat balance of the air system serving the four Learning Centers, shown in Table B, accounted for the actual supply and return air temperatures which have been measured in winter.

The purpose of such a heat balance was to account for the fact that to maintain winter comfort in this area a supply air temperature of 80F was found necessary, as compared with design expectations that the school could operate year round with a supply temperature of 60 ± 2.5°F.

Table B thus confirms that the massive concrete roof plus earth cover has a cooling effect on the air system much larger than expected. In addition, the window and skylight losses are greater, and the heating contribution from the perimeter fintubes is much less than expected, as reported at the Montreux Symposium, Ref. 5.

IMPROVEMENTS ACHIEVED AND PROPOSED

These are as follows:

a) Addition of temperature sensors and additional flowmeters in the SUNKEEPER system so as to obtain more reliable data continuously on the solar system performance and end-use efficiency.

b) Minor changes of valves and instrumentation in both the heating loop and cooling loop of the HVAC hydronic system.

c) Resetting of control temperatures and control logic in the HVAC system so as to make more effective use of the solar system both winter and summer.

d) A heavy coat of sprayed-on insulation on the underside of the entire concrete slab roof so as to increase its thermal resistance significantly, i.e. from 5.4 to about 10. Alternatively, an insulated return-air duct system would achieve the same result while ensuring that the return air from each space was in correct proportion.

In addition to the above, the massive concrete projections over the windows and in the skylight should be redesigned in future schools to reduce their "wicking" effect which defeats the effectiveness of the earth cover by large heat losses in winter and heat gains in summer.

A noteworthy feature of the roof design is that it has been successfully sealed against any seepage, or even mildew, from the high moisture content of the overlying earth and gravel. In a climate of 39 inches of rain per year, and high summer humidity, the waterproofing cover of bituthane well bonded to the concrete, and fully tested by roof-flooding, is therefore well-proven.

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Figure 1
Floor Schematic
Terraset School

Figure 2
Simplified HVAC and Solar System: Terraset School

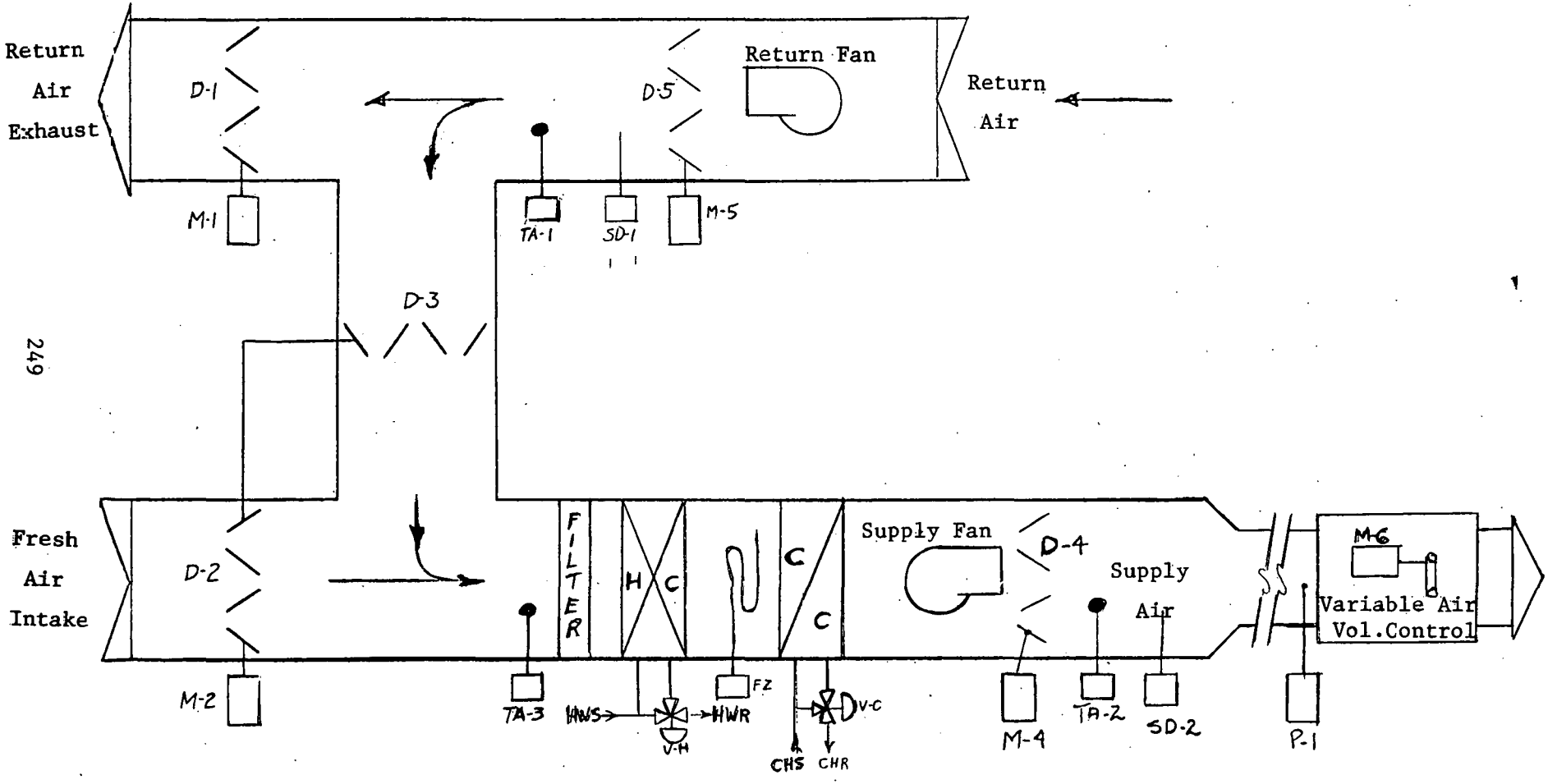


Figure 3 AIR HANDLING FOR LEARNING CENTERS

TERRASET SCHOOL

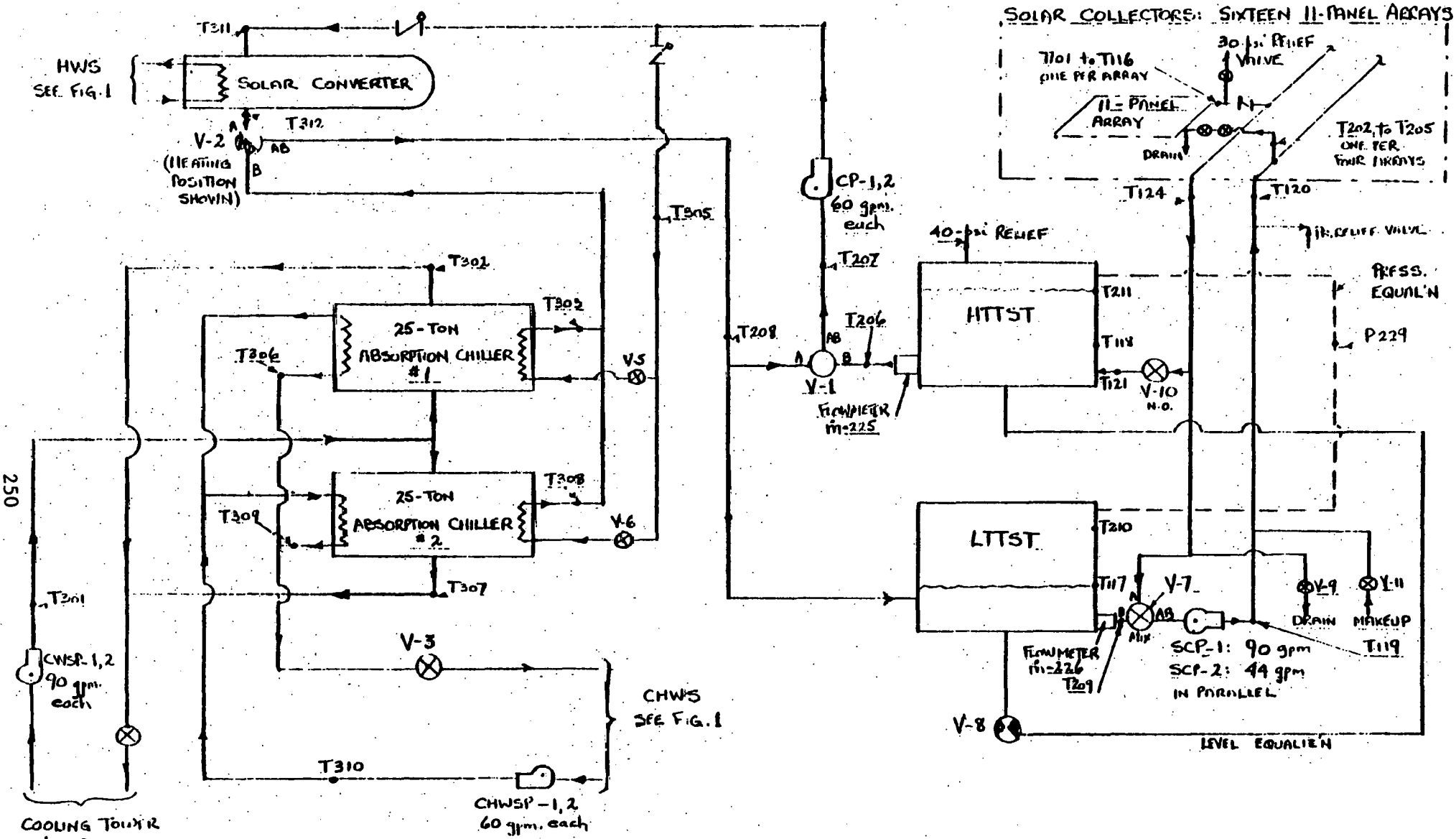


Figure 4
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Table A

THERMAL RESISTANCE OF THE TERRASET ROOF COVER

Layer	ASHRAE Fund'ls 1977	Winter Resistance hr-ft²-°F / Btu	Winter Interface Temp. °F	Summer Resistance hr-ft²-°F / Btu	Summer Interface Temp. °F
			20.0		90.0
A_1: Air film + radiation	p. 22.11	.20		.23	
			21.4		89.8
A_2: Grass	-	0		2.5	
			21.4		87.6
B: Wet clay, 30"	p. 22.28	3.33		5.0	
			44.5		83.2
C: Gravel, 6"	p. 22.28	0.50		0.50	
			48.0		82.8
D: Polystyrene bead-board	p. 22.14	1.80		1.80	
			60.5		81.2
E: Bituthane, .064"	p. 22.16	0.07		0.07	
			61.0		81.1
F: Reinforced conc., 10"	p. 22.15	0.80		0.80	
			66.5		80.4
G: Air film + radiation	p. 22.11	0.50		0.50	
			70.0		80.0
Total R:		7.20		11.4	

Table B

AIR SYSTEM ENERGY BALANCE: Typical Winter Day (20F outside)

(Learning Centers Only: AHU #1-#4, 242,500 lb/hr air)

Supply Air Temp., TA-2 = 80F (See Fig. 3)			
Children Present	0	300	900
HEAT ADDED: Btu/hr. Air Supply to Air Return			
People: (250 B/h children, 330 B/h teachers).	0	76,000	248,000
Lights:	40,000	208,000	208,000
Perimeter Fintubes (Circuits #1 & #2).	132,000	132,000	132,000
TOTAL ADDED, (Btu/hr)	172,000	416,000	588,000
HEAT LOST: B/hr. from L. Centers #1, #2, #3, #4.			
Side Windows (715 ft^2 ea) & Doors (136 ft^2 ea.)	210,500	221,000	242,000
Walls: into earth berms	20,000	20,000	20,000
above & below windows	240,000	244,000	253,000
Floor (periphery, 500 ft)	20,000	20,000	20,000
Windows in front hall (assume offset by fintubes)	0	0	0
Roof: Assume mean temp. in plenum, T_m	(68.4F)	(72.0F)	(74F)
Heat loss, (55,360 ft^2 @ R = 5.4**)	495,000	533,000	554,500
Skylight (2100 ft^2 conc., 113 ft^2 glass).	105,500	110,000	120,500
TOTAL LOST (Btu/hr)	1,091,000	1,148,000	1,210,000
$\Delta T = [T_{supply} - T_{return}] = [Q_{lost} - Q_{added}]/mCp.$	15.8	12.6	10.7
$T_{return} = [T_s - \Delta T]$ = TA-1 in Fig. 3 [These temperatures confirmed by experiment].	64.2	67.4	69.3
T_x to plenum = $T_s - \Delta T$*	72.7	76.6	78.9
T_m = Mean temp. in plenum = $(T_{return} + T_x)/2$, (compare with T_m above)	68.45	72.0	74.1

$\Delta T^* = (Q_{lost} - Q_{added} - Q_{roof})/mCp$

**R = 5.4 based on assuming both the polystyrene & bituthane as ineffective.

COST EFFECTIVENESS - AN ASSESSMENT BASED ON COMMERCIAL DEMONSTRATION PROJECTS

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ABSTRACT

Detailed after-the-fact construction costs are presented for 24 National Solar Demonstration projects. Cost analyses include an evaluation of the cost components and cost factors impacting solar energy systems.

Based on the results of these analyses and detailed performance data obtained from the National Solar Data Network, an economic assessment of nine operating systems is presented. The economic analyses include first year cost of energy, payback period, and rate of return. Implications for the cost effectiveness of similar systems in other areas and under different economic circumstances are investigated to the extent possible.

INTRODUCTION

The use of solar energy systems for heating, cooling or providing process water for commercial applications is commonly considered to be technically feasible. A question that remains is the degree of cost effectiveness and the relative importance of the many significant factors that impact it. This paper reports on continuing work that has utilized the expanding data base within the National Solar Data Program to address the cost effectiveness of National Solar Demonstration projects.

The components of the investigation are system cost, system performance and the selection of appropriate calculation methods and parameters to define cost effectiveness. Accurate total system construction costs have been obtained for twenty-four demonstration projects to date. These total system costs are presented in this paper along with cost breakdowns of major categories for each system. Results of analyses are presented that identify the major factors that impact total system and subsystem costs.

System performance has been determined by the National Solar Data Network (NSDN) for a number of demonstration projects. Because performance aspects of these systems are discussed elsewhere in this conference, only the summary parameters quantifying system performance are presented here. The cost effectiveness of nine demonstration projects as determined by a number of different pro-
ures is presented and discussed at length.
e analyses combine the cost and performance
information for a more complete assessment of cost effectiveness.

COST ANALYSIS

General

The selection of projects for which accurate total system and subsystem costs have been obtained includes a number of different system types as described in Table 1. The 24 projects are identified in Table 2, which also includes information on the type of application, location, type of construction and collector area.

These sites were not selected to be representative of all solar energy systems. A very large sample of systems would be needed to completely understand all the operant factors. Nonetheless, the 24 sites listed do provide an adequate basis for some preliminary conclusions.

The fact that only one passive system is included in the selection reflects the fact that few commercial demonstration projects utilize passive solar energy. Most of the program activity in the passive area has focused on residential demonstration projects.

Table 1. Summary of Project Types

DESCRIPTION	NUMBER OF SITES
Passive	1
Heating	10
Process Water	5
Heating & Cooling	8
New	10
Retrofit	14
Air Type Collectors	5
Liquid Type Collectors	18

Cost data on the subject 24 sites are available in "Solar Project Cost Reports" from the DOE Technical Information Center (TIC). See Table 2 for specific report numbers. Current plans call for a continued program of cost data collection analysis and reporting.

The cost data were obtained during visits to the sites, discussions with system designers, construction contractors, owners, and review of DOE vouchers. In the cost reports, construction costs

Table 2. Basic Information for Cost Analyzed Projects

PROJECT NAME	APPLICATION[a]	LOCATION	CONSTRUCTION[b]	COLLECTOR AREA, SQ.FT[c]	COST REPORT #[d]
1. Aberdeen First Baptist Church	SH,HW (AIR)	Aberdeen, SD	N	1,260	SOLAR/2070-79/60
2. Aratex Industrial Laundry	HW	Fresno, CA	R	6,350	SOLAR/2008-78/60
3. Billings Shipping Warehouse	SH (LIQ)	Billings, MT	R	1,660	SOLAR/2066-79/60
4. Blakedale Professional Center Office Bldg.	SH,HW (LIQ)	Greenwood, SC	N	928	SOLAR/2014-79/60
5. Charlotte Area Health Center	SH,HW (LIQ)	Charlotte, NC	R	3,950	SOLAR/2010-79/60
6. Columbia Gas Office Bldg.	SH & C, HW	Columbus, OH	R	2,978	SOLAR/2068-79/60
7. Concord Municipal Light Building Garage	SH (AIR)	Concord, MA	R	1,737	SOLAR/2048-79/60
8. DuCat Investments Warehouse	SH (AIR)	Kansas City, KS	N	7,000	SOLAR/2069-79/60
9. Hogates Restaurant	HW	Washington, DC	R	5,840	SOLAR/2028-79/60
10. Howard's Grove School	SH (AIR)	Howard's Grove, WI	N	2,357	SOLAR/2041-79/60
11. Ingham Co. Medical Care Facility	HW	Okemos, MI	N	9,374	SOLAR/2056-79/60
12. Iris Images Photo Processing Lab	HW	Mill Valley, CA	N	600	SOLAR/2005-78/60
13. Irvine School	SH & C, HW	Irvine, CA	R	5,000	SOLAR/2021-79/60
14. Kalwall Warehouse	SH (PAS)	Manchester, NH	R	1,750	SOLAR/2015-78/60
15. Loudoun Co. School	HW	Leesburg, VA	N	1,169	SOLAR/2016-77/60
16. Terrell E. Moseley Office Bldg.	SH, HW (LIQ)	Lynchburg, VA	R	376	SOLAR/2011-78/60
17. Mount Rushmore Memorial Visitor's Center	SH & C	Keystone, SD	R	1,725	SOLAR/2019-79/60
18. North Hampton Recreational Center	SH & C	Dallas, TX	R	3,660	SOLAR/2038-79/60
19. Page Jackson School	SH & C	Charlestown, WV	N	10,943	SOLAR/2036-79/60
20. Radian Office Building	SH & C	Austin, TX	R	350	SOLAR/2002-78/60
21. Reedy Creek Utilites Office Building	SH & C, HW	Lake Buena Vista, FL	N	3,840	SOLAR/2018-79/60
22. Scattergood School Gymnasium	SH,HW (AIR)	West Branch, IA	N	2,240	SOLAR/2003-78/60
23. Telex Communications Assembly Plant	SH (LIQ)	Blue Earth, MN	R	10,700	SOLAR/2033-79/60
24. Trinity University Sports Complex & Dormitory	SH & C, HW	San Antonio, TX	R	15,633	SOLAR/2004-78/60

[a] SH = Space Heating, HW = Hot Water, SH & C = Space Heating and Cooling, (AIR) = Air Systems, (LIQ) = Liquid Systems, (PAS) = Passive System.

[b] N = New construction, R = Retrofit construction

[c] Net aperture area

[d] Available from the Technical Information Center, P.O. Box 62, Oak Ridge Tennessee 37830. Project Description Documents have report numbers identical to Cost Report Numbers, except the last two digits are 50.

Table 3. Summary of System Costs

APPLICATION	PROJECT	TOTAL[a] SYSTEM COST, $	TOTAL[b] SYSTEM $/SQ FT	COLLECTOR		SUPPORT STRUCTURE		PIPING DUCTWORK, & INSULATION		HEATING COOLING/ EQUIPMENT		STORAGE		ELECTRICAL & CONTROLS		GENERAL CONSTRUCTION	
				$/SQFT	%[c]	$/SQFT[d]	%	$/SQFT	%	$/SQFT	%	$/SQFT	%	$/SQFT	%	$/SQFT	%
Passive	Kalwall	13,530	7.7	6.1	79	--	--	--	--	0.6	8	0.8	10	0.3	3	--	--
Process Hot Water	Aratex	18,240	29.0	11.7	40	2.8	10	10.4	36	--	--	2.4	8	1.3	4	1.0	3
	Iris Images	18,115	30.2	10.5	35	7.2	24	4.0	13	--	--	2.1	7	1.8	6	4.4	15
	Ingham Co.	337,260	36.0	10.3	29	11.2	31	11.2	31	--	--	1.2	3	1.7	5	0.8	2
	Hogate's	283,955	48.6	18.2	37	8.7	18	13.1	27	--	--	4.1	8	1.1	2	3.4	7
	Loudoun Co.	70,940	60.7	21.8	36	17.2	28	12.2	20	--	--	5.4	9	3.3	5	0.8	1
	Average	---	40.9	14.5	35	9.4	23	10.2	25	--	--	3.0	7	1.8	4	2.1	5
Liquid Space Heating	Moseley	14,295	38.0	13.8	36	3.3	9	6.3	17	5.2	14	4.1	11	4.6	12	0.8	2
	Telex	462,305	43.2	17.3	40	6.6	15	13.9	32	1.0	2	1.5	3	2.0	5	0.9	2
	Billings	77,430	46.6	20.4	44	7.8	17	13.3	29	0.9	2	1.8	4	2.5	5	--	--
	Charlotte	249,195	60.8	18.0	30	5.9	10	24.8	41	0.2	<1	3.3	5	3.3	5	5.1	8
	Blakedale	60,510	65.2	12.0	18	15.4	24	21.0	32	--	--	8.2	13	8.5	13	0.1	<1
	Average	---	50.8	16.3	32	7.8	15	15.9	31	1.5	3	3.8	7	4.2	8	1.4	3
Air Space Heating	Howard's Grove	53,300	22.6	13.4	59	2.4	11	3.2	14	--	--	2.7	12	0.9	4	--	--
	DuCat	252,435	36.1	21.9	61	9.3	26	3.5	10	--	--	--	--	1.4	4	--	--
	Aberdeen	62,595	49.7	24.8	50	1.3	3	13.1	26	--	--	5.6	11	3.3	7	0.9	2
	Scattergood	113,825	50.8	24.5	48	11.5	23	6.0	12	2.1	4	3.8	7	2.7	5	0.2	<1
	Concord	118,740	68.4	18.6	27	25.7	38	12.3	18	--	--	5.2	8	6.2	9	0.4	1
	Average	---	45.5	20.6	45	10.0	22	7.6	17	0.4	1	3.5	8	2.9	6	0.3	1
Space Heating & Cooling	Page Jackson	526,845	48.1	15.9	33	15.8	33	8.4	17	3.3	7	1.5	3	0.3	1	3.0	6
	Irvine	285,190	57.0	28.3	50	7.7	14	15.0	26	--	--	--	--	4.8	8	1.3	2
	Trinity	958,210	61.3	29.1	47	5.5	9	13.2	22	2.9	5	4.2	7	2.9	5	3.6	6
	Mt. Rushmore	130,705	75.8	21.2	28	13.9	18	16.9	22	8.6	11	3.6	5	6.6	9	5.0	7
	North Hampton	384,660	105.1	21.4	20	6.3	6	31.5	30	7.4	7	8.9	8	16.2	15	13.4	13
	Columbia Gas	352,260	118.3	53.0	45	10.2	9	17.1	14	--	--	4.5	4	14.6	12	18.9	16
	Radian	42,150	120.4	37.4	31	15.7	13	30.9	26	14.4	12	7.7	6	9.8	8	4.8	4
	Reedy Creek	625,985	163.0	108.4	67	4.7	3	18.9	12	4.7	3	10.9	7	5.6	3	10.4	6
	Average	---	93.6	39.3	42	10.0	11	19.0	20	5.2	6	5.2	6	7.6	8	7.6	8
	Total Average	---	60.1	24.1	40	9.0	15	13.4	22	2.1	3	3.9	6	4.4	7	3.3	5

[a] All costs in 1977 $'s, costs include overhead and profit

[b] Dollars per square foot of Net collector area

[c] Percent of total system cost

[d] -- No category costs incurred

are divided into cost categories that roughly parallel construction work categories as follows: collector array; support structure; piping; ductwork; insulation; heating and cooling equipment; storage; controls; electrical power; general construction. For the analyses in this paper, piping, ductwork and insulation costs are lumped in a single cost category as are electrical and controls costs. The cost reports also present more detailed cost data if available.

The costs presented are normalized to account for differences in the contractual structures of the projects through the application of a standard overhead and profit rate to bare costs. In addition, all costs are expressed in 1977 dollars.

The cost figures do not include the design, instrumentation or auxiliary energy system costs. More information concerning the cost data collection and analysis process can be found in the cost reports. "Solar Project Descriptions" are also available from TIC for most of the 24 projects discussed in this paper.

The reader is reminded that in comparing the costs of various systems and subsystems, "cheaper" is not necessarily "better." Nine of the 24 projects are subjected to cost effectiveness analysis in another section of this paper. That analysis includes the effect of system performance and is thus a better indicator of the overall success of a system. Nonetheless, terms employed here such as dollars/sq.ft. of collector area can be very useful when the reader is aware of the limitations of their application.

Total System Cost Analysis

Table 3 summarizes the category cost breakdowns for the 24 sites grouped according to type of application, with cost presented as dollars per unit collector area and percent of the total system cost. More detailed cost data are presented in the individual cost reports. In Table 3 it can be seen that the unit cost for the total system for each application type varies considerably. Thus, one should beware of gross summarizations such as "space heating systems in commercial buildings cost X $/sq.ft. of collector area."

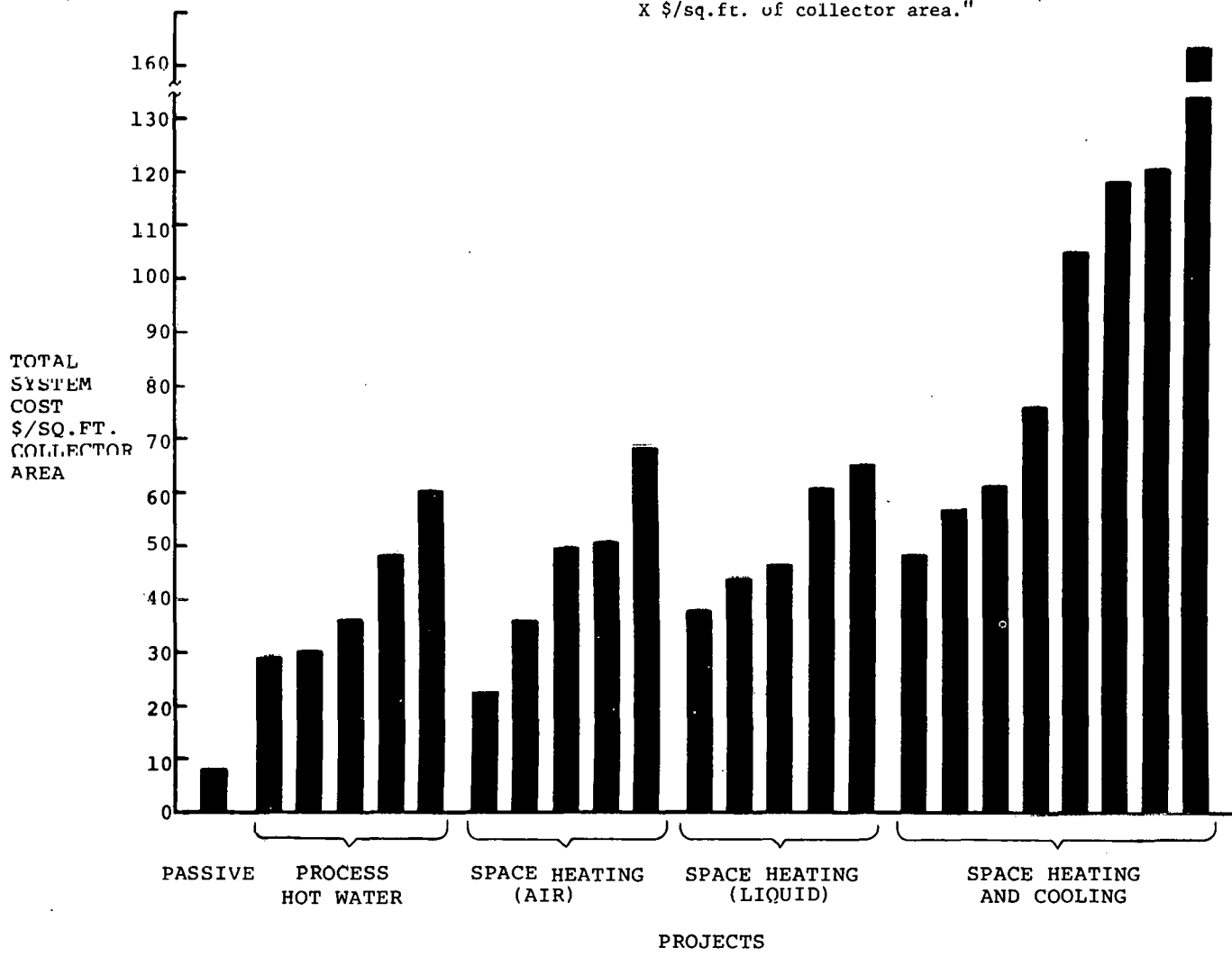


Fig. 1. Total System Costs Grouped by System Type

Also, for most of the sub-categories, very large variations are evident in unit costs. For example, collector category costs vary between $10./sq.ft. and $163./sq.ft., support structure costs vary between $2.40/sq.ft. and $26./sq.ft., and piping, ductwork and insulation costs vary between $3.20/sq.ft. and $32./sq.ft. Each system has unique characteristics that affect these category costs. The system characteristics are described more fully in the Project Descriptions.

While most costs vary, some consistency in the figures in Table 3 is also evident. The percent of total system cost values are relatively uniform within most of the cost categories. Storage costs are also fairly consistent, with only a few systems varying significantly from the average $3.90/sq.ft.

Figure 1 illustrates the relationship between the total system unit costs for the 24 sites for the five different system types; passive, process hot water, space heating (air type collectors), space heating (liquid type collectors), and space heating and cooling. Only one passive system, Kalwall, is listed, and its total cost of less than $8/sq.ft. is much lower than all the other types. The five process hot water systems range from $29 to $61/sq.ft. with an average of $41/sq.ft. The five space heating systems employing air type collectors range between $23 and $68/sq.ft. with an average of about $46/sq.ft. The five space heating systems employing liquid type collectors range between $38 and $65/sq.ft. and average about $51/sq.ft. The space heating systems combined average $48/sq.ft. The eight space heating and cooling projects range between about $48 and $163/sq.ft. with an average of $94/sq.ft. If the Reedy Creek system costs (the highest by more than $40/sq.ft.) are not included, the average cost for the space heating and cooling projects is $84/sq.ft.

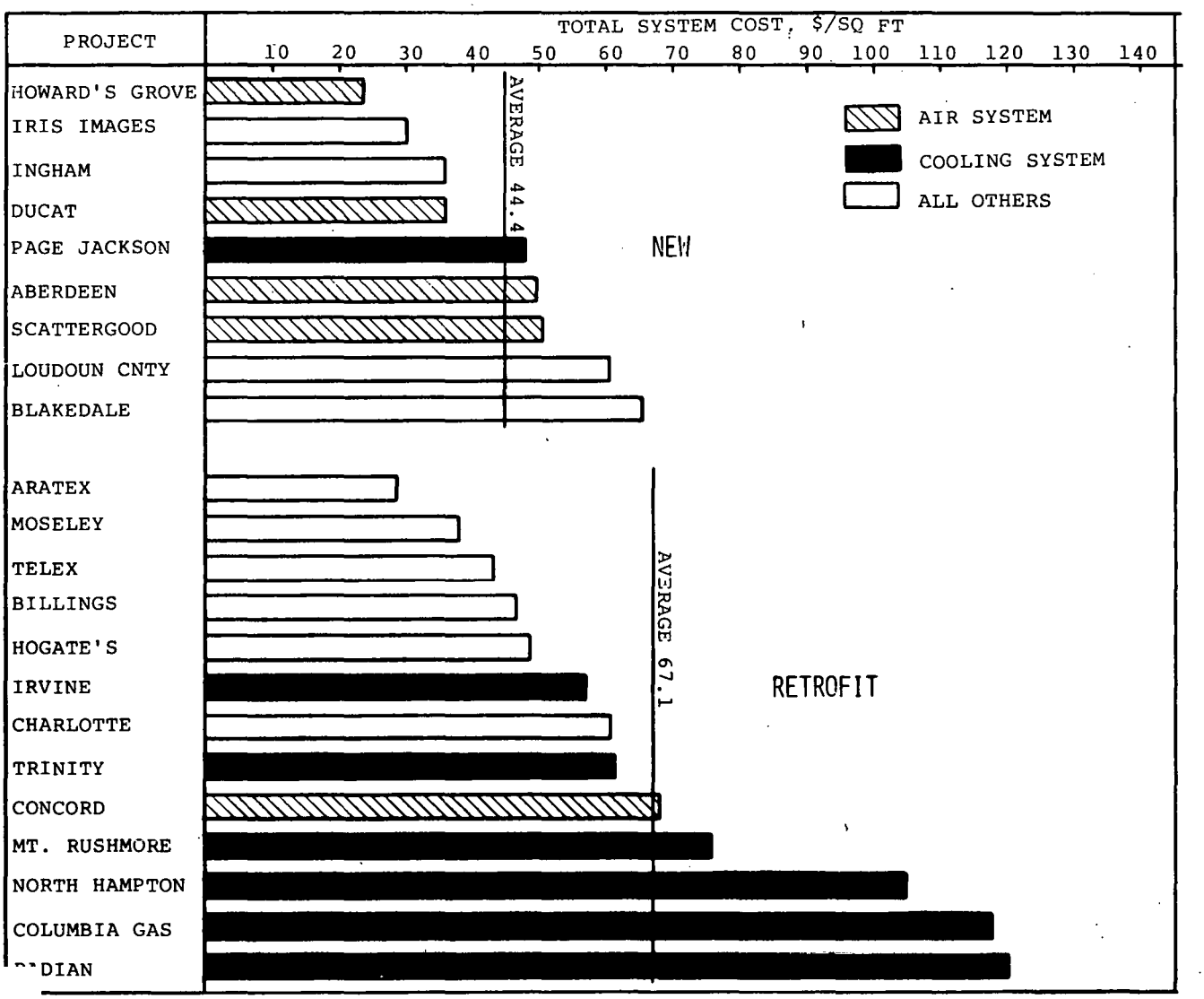


Fig. 2. Total System Cost for New and Retrofit Systems

The data collected to date and as illustrated in Fig. 1 agree with previous conclusions that process water systems are the least expensive active solar application on a per square foot basis, followed closely by space heating applications. Space heating and cooling applications are much more expensive, on the average, than the other system types. However, there are exceptions to these general rules, as can be seen in Fig. 1. Some cooling applications have been installed at lower relative cost than some process water systems. Also, there is a very large range of costs for every application type.

Total system costs are presented again in Fig. 2, this time divided into two main groups, new and retrofit applications. As shown, new systems have an average cost of about $44/sq.ft. and retrofits have an average of $67/sq.ft. However, two factors should be recognized. First, all but one of the cooling projects were retrofit. (This fact may account for some of the higher average costs of the cooling projects in this sample.) Note that Reedy Creek has not been considered in this analysis. If no cooling projects are considered, the average is $44/sq.ft. for new and $48/sq.ft. for retrofit systems.

The second interesting factor in Figure 2 is that all but one of the five air type space heating systems are new systems. This probably reflects the greater difficulty perceived for retrofitting air systems. Three of the four new air systems have total costs significantly below the average for new systems. Given all of the factors, the conclusion of this analysis is that retrofit systems are only slightly more costly than systems in new construction. This difference may even decrease as a larger sample of projects is obtained.

Economies of Scale

The cost data were analyzed statistically to investigate possible economies of scale effects for total system costs. "Economies of scale" is the term used when the unit cost of "large" systems is less than the unit cost of "small" systems. The economies of scale analysis consists of plotting total system cost versus collector area on a log-log graph. Regression analysis is used to find the best fit line through the points. Figure 3 illustrates the relationship between the slope of lines on a log-log plot and the corresponding curves plotted on a linear graph. Note that curve C is the type that would be expected if an economies of scale effect were at work. Curve B indicates no economies of scale, and curve A indicates a reverse economy of scale wherein unit costs increase as system size increases.

The plot of total system cost versus collector area is shown in Figure 4. The solid line is the best fit line determined by regression analysis. The slope of this line is 0.95. The broken line indicates the result of assuming that no economies of scale are involved (slope = 1.00) and that the costs can be represented by the average cost per unit collector area. Note that while some economies of scale effect is indicated, the broken line (no economy of scale) is nearly as good as a fit through the points. Thus, within the range of systems studied, economies of scale are a relatively unimportant cost factor.

Looking further at Figure 4, it can be seen that all cooling projects are very near or above the best fit line. No other project types are above the line.

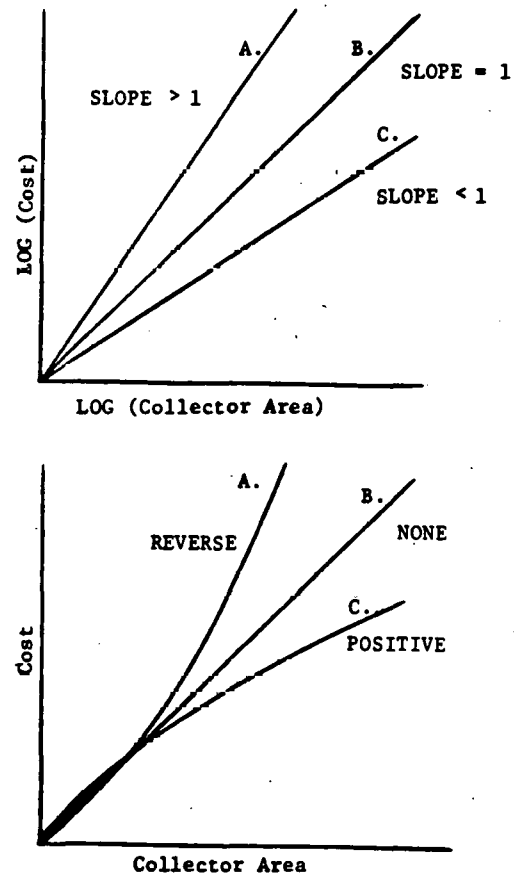


Fig. 3. Lines on Log-Log Scale and Corresponding Curves on Linear Scale

Regional Cost Variation

Another application of regression analysis for identifying cost factors is based on the computation of a correlation coefficient for the best fit line through log costs plotted against log collector areas. The correlation coefficient is a measure of how well the line fits the points. Assume a correlation coefficient is obtained for a sample of cost points. Then, if the cost figu are multiplied by some factor to normalize

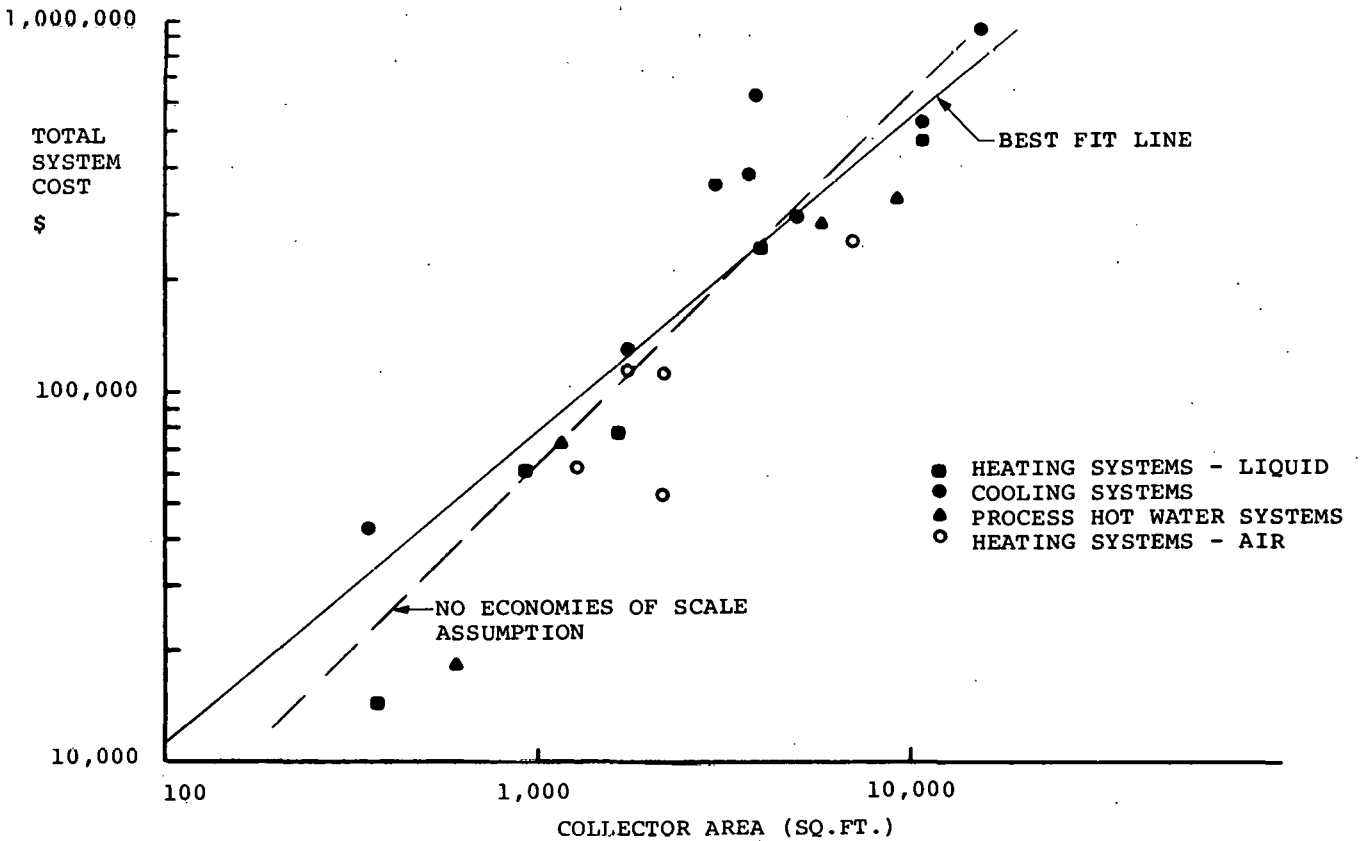


Fig. 4. Regression Analysis of Total System Cost Versus Collector Area

the data and eliminate the effect of some parameter and plotted again, an increase in the correlation coefficient would result. The impact of a cost factor on system cost could be measured by the increase in the correlation coefficient when that normalization factor is applied.

For example, as shown in Figure 4, the log of the system cost was plotted against the log of the collector area for the 24 systems and regression analysis yielded a correlation coefficient of 0.90. The costs were then divided by the mechanical system city cost indexes in Means Mechanical and Electrical Cost Data 1978. These city cost indexes are an indication of the geographic variation in mechanical system materials and installation costs. Table 4 lists the 24 projects, the closest cities for which cost index information was available, and the total mechanical system cost indexes used.

After the system costs were divided by the city cost index, which normalized the costs against regional variations in materials and labor costs, a regression analysis was performed again. The resulting correlation coefficient was 0.89. The fact that the correlation coefficient actually reaced by one percent indicates that regional luences have not been significant in the systems costs studied to date. The slope of the line resulting from this analysis was 0.94, indicating

Table 4. Regional Cost Indexes From 1978 Means Mechanical and Electrical Cost Data

System	Nearest Means City	City Cost Index
ABERDEEN	Sioux Falls, SD	90.6
ARATEX	Fresno, CA	107.0
BILLINGS	Billings, MT	96.5
BLAKEDALE	Columbia, SC	82.7
CHARLOTTE	Charlotte, NC	82.4
COLUMBIA GAS	Columbus, OH	98.2
CONCORD	Lowell, MA	99.3
DUCAT	Topeka, KA	93.5
HOGATE'S	Washington, DC	98.5
HOWARD'S GROVE	Madison, WI	96.6
INGHAM	Lansing, MI	99.4
IRIS IMAGES	San Francisco, CA	119.0
IRVINE	Los Angeles, CA	108.7
LOUDOUN CO.	Richmond, VA	90.9
MOSELEY	Roanoke, VA	88.1
MT. RUSHMORE	Sioux Falls, SD	90.6
NORTH HAMPTON	Dallas, TX	91.2
PAGE JACKSON	Huntington, WV	99.0
RADIAN	Austin, TX	90.7
REEDY CREEK	Orlando, FL	90.5
SCATTERGOOD	Davenport, IA	94.5
TELEX	Minneapolis, MN	94.8
TRINITY	San Antonio, TX	91.9

that even after normalizing the data to account for regional variations there are no appreciable economies of scale.

SUBSYSTEM COST ANALYSIS

Collectors

Much attention has been focused on solar collectors as an area for cost reduction because collectors represent a significant portion of total system cost. Figure 5 presents the cost data for the solar collectors for the 24 sites grouped according to collector type. These costs do not agree with the costs listed in Table 3 under "Collector Array" because the array costs include installation labor; Figure 5 only includes material costs.

In Figure 5, it can be seen that the two site-built flat plate collectors were less expensive than the other flat plate collectors. There appears to be little cost variation between double glazed, flat black surface collectors, single glazed, selective surface collectors, and double glazed, selective surface collectors. The average cost for these three types is $16/sq.ft., which includes overhead and profit. (These costs are in 1977 dollars.) In all cases, the concentrating, tracking collectors and the single evacuated tubular collector were more expensive per square foot than the flat plate collectors. Only one evacuated tubular collector was employed in the systems studied. The prototype collector at Reedy Creek which integrated the roof and the collector reflector troughs was more than twice as expensive as the next highest cost collector.

Also shown in Figure 5 is the percentage of the total system cost represented by the collector materials. Of the active systems studied the collector materials varied from 17 to 64 percent of the total system cost with an average of 34 percent.

Support Structure

Significant variation in system collector support structure costs has been observed. Support structure costs vary from $2.40/sq.ft. at Howard's Grove to $25.70/sq.ft. at Concord. The factor that seems to affect cost most is whether the support structure is used for functions in addition to collector support. These additional

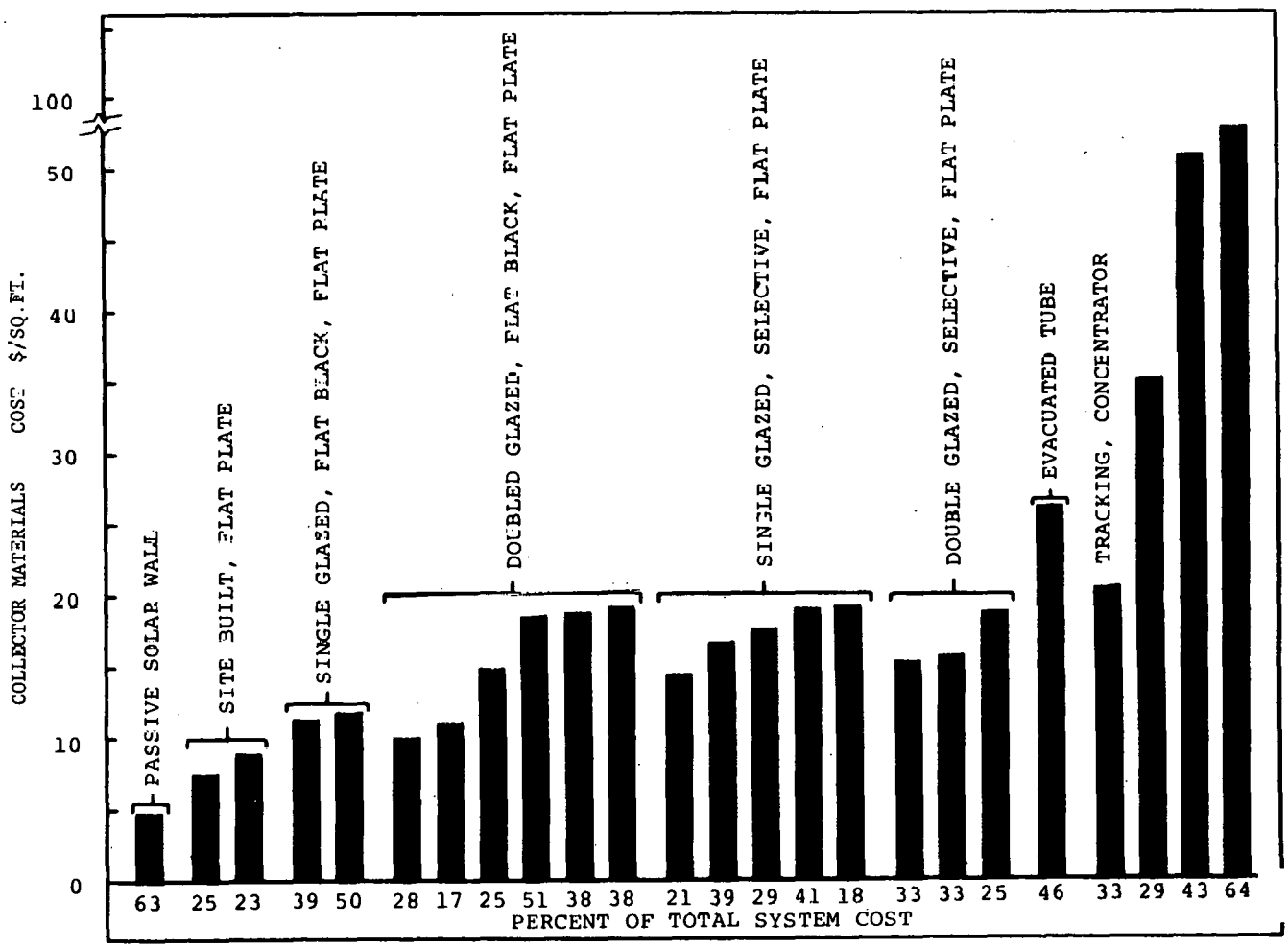


Fig. 5. Collector Materials Cost in Dollars Per Sq. Ft. and Percent of Total System Cost

functions include the following:

- o House equipment
- o Improve system appearance
- o Provide reflectors
- o Form building roof structure
- o Form building waterproof membrane

Figure 6 compares the costs of the support structures that serve multiple functions with those that merely support the collectors. There is a $5.50/sq.ft. difference in average costs to those that groups. Another interesting fact is that all but two of the twelve single function support structures are for retrofit systems. Conversely, only three of the ten multiple function support structures are for retrofit systems. Apparently, the constraints of retrofit systems lead designers to utilize simpler support structures. The flexibility that new systems provide the designer apparently encourages multiple uses for the support structure. The lesson of Figure 6 is that these extra functions will increase the support structure cost. In some cases, the extra functions that the support structure performs result in reduced cost in other areas, and credit would have to be provided. If the support structure provides reflectors, the reflectors will improve system performance. A tradeoff will be required to determine the cost benefit of the reflectors. In most cases, however, the support structure does not contribute to performance, so that least cost to perform the support function should be sought. Thus, for the support structure, except when reflectors are used, $/sq.ft. is a better evaluation factor than $/mmBtu. From Table 3 the support structure on the average represents 15 percent of the total system cost, more than twice as much as the storage subsystem.

Piping, Ductwork and Insulation

If retrofit systems are more expensive than solar

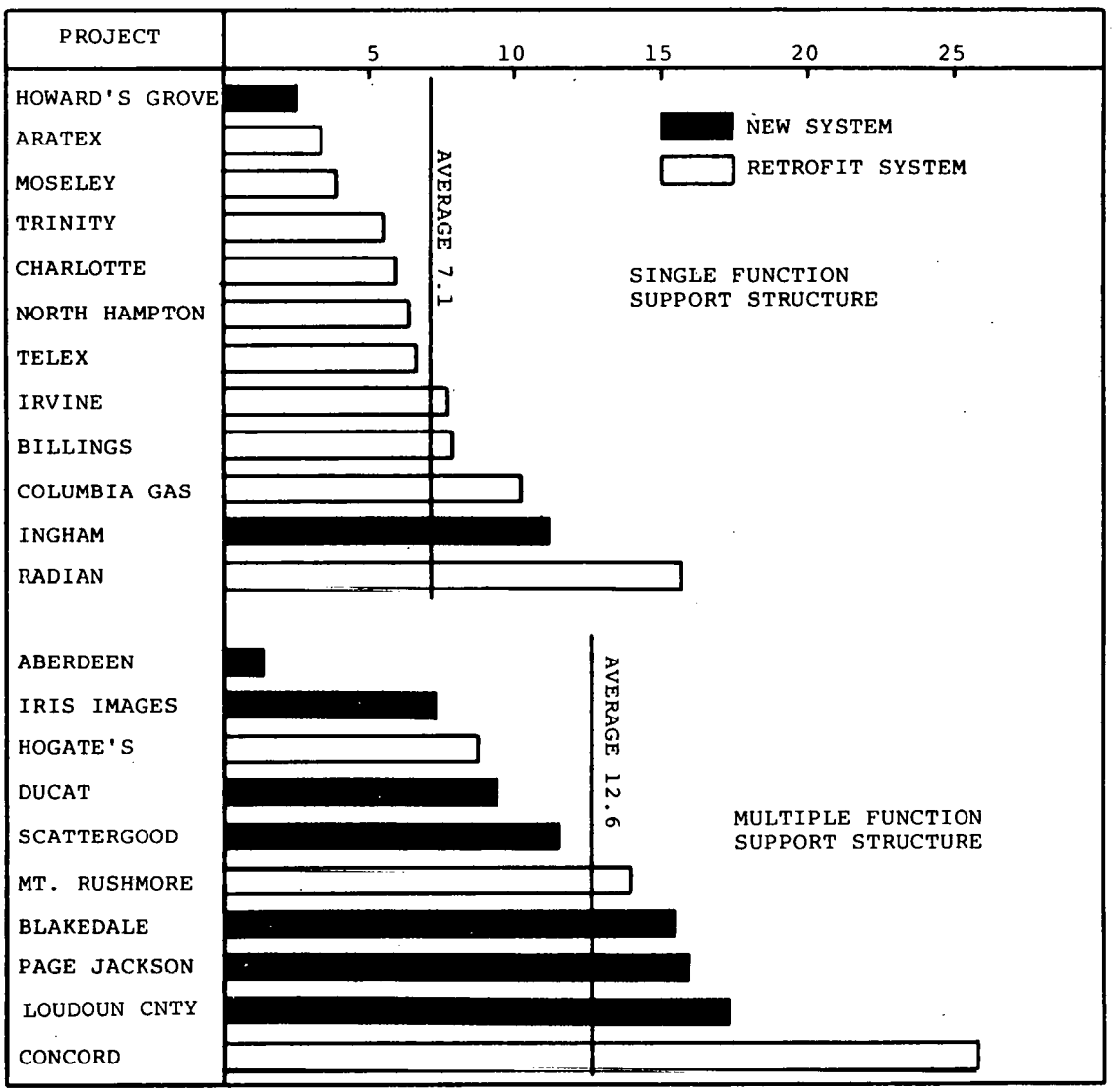


Fig. 6. Support Structure Costs Grouped by Single and Multiple Function

systems on new buildings, one would expect added costs to occur in the energy transport system. Figure 7 illustrates the piping, ductwork and insulation category cost totals. The systems are divided into new and retrofit construction. Approximately an $8 difference exists between the averages for the two groups with retrofit applications higher, as expected. However, most of the cooling projects are again retrofit and their costs in this category are higher. Also, three of the four lowest costs for piping, ductwork and insulation are for air systems. Referring to Table 3, this category average cost is $7.60/sq.ft. for air systems, $15.90/sq.ft. for liquid type heating systems and $19.00/sq.ft. for heating and cooling systems.

In Table 3, piping, ductwork and insulation are seen to represent 22 percent of the total system cost on the average for all system types. For space heating systems using liquid type collectors, this portion is 31 percent and for air type collector systems, 17 percent.

Storage

Storage subsystems are generally easy to define within solar energy systems and their costs have been found to be easily separable from total system costs. On the average, storage subsystems represent six percent of the total system cost. This is a very small percentage considering the importance of this subsystem to overall system performance. A number of systems have been monitored in which twenty to fifty percent of the energy placed in storage is lost from storage. However, it should be noted that other systems indicate "storage efficiencies" greater than 90%.

Analysis shows that storage costs are strongly dependent on the type of storage vessel used. Figure 8 indicates the storage cost per unit storage capacity ($/MBtu/^{o}F). The order of increasing cost for storage types is:

- o Non-pressurized steel tanks
- o Fiberglass tanks

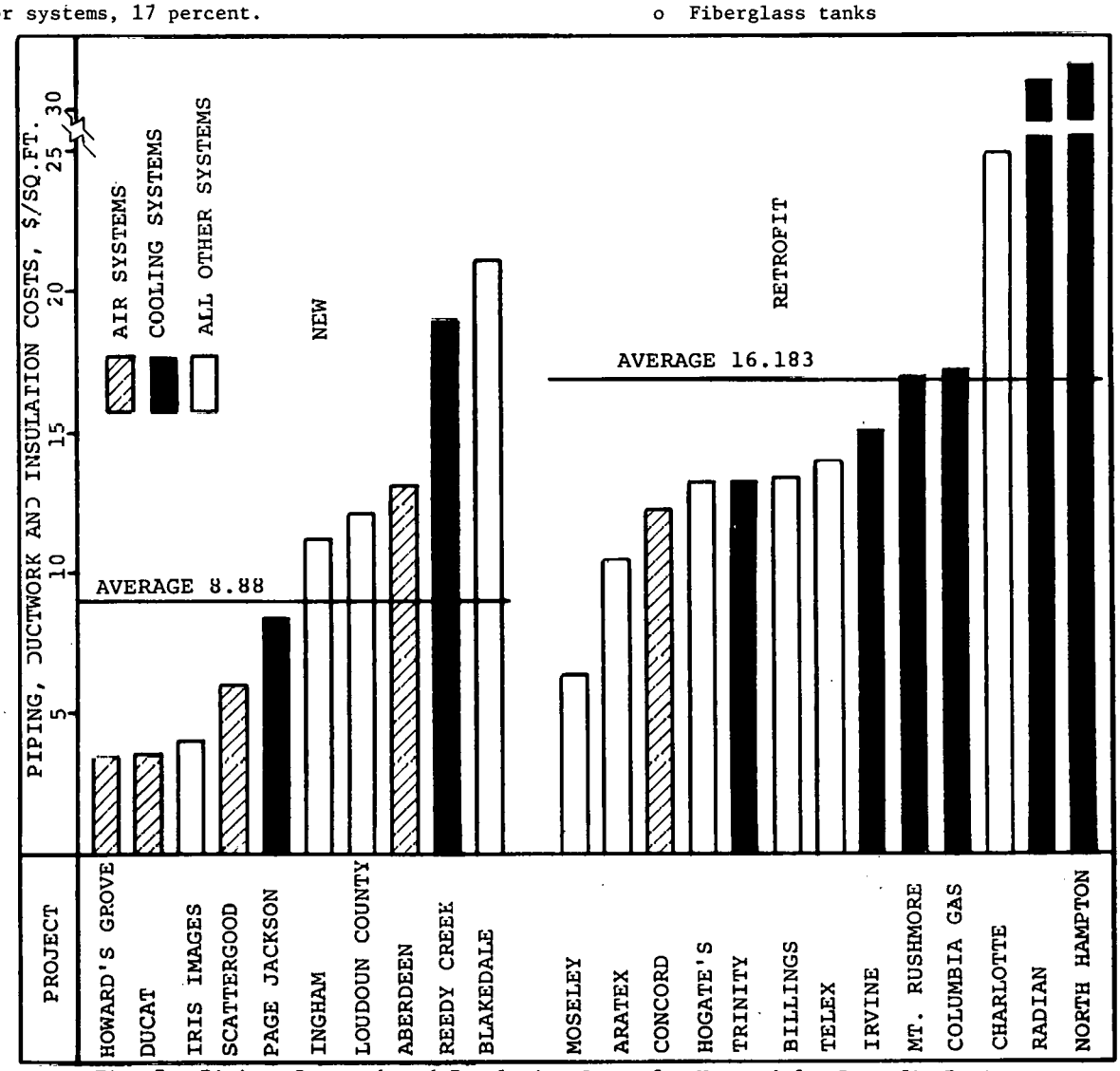


Fig. 7. Piping, Ductwork and Insulation Costs for New and for Retrofit Projects

- o Pressurized steel tanks
- o Rock bins
- o Residential water heater type tank

Referring back to Table 3, note that for unpressurized systems, the storage costs are lower, but they represent a higher percentage of total system cost. This implies perhaps that pressurization impacts other costs as well as storage system costs.

The effect of storage tank location was also investigated although not illustrated here. On the average, buried tanks were found to be least expensive, but only slightly less expensive than exterior tanks. Tanks placed in buildings were, on the average, significantly more expensive than the others. The added cost of piping to and from exterior or buried storage is not reflected here and should be taken into consideration. Also, the analysis does not charge the cost of building space to the costs for interior storage. This finding regarding the effect of storage location is surprising since one would think that interior tanks, without need for waterproofing or protection, would be less expensive.

Controls

Controls costs were analyzed separately from the electrical costs. The range of costs was large regardless of the methods used to normalize the costs. Figure 9 illustrates the controls costs for the 16 projects for which the control costs were separable. The vertical scale is dollars per actuator controlled. Pumps, blowers, automatic valves, and motorized dampers are all considered actuators. The number of actuators for each system is provided on top of the respective cost bar. The cost per actuator controlled can be seen to vary from about $325 to $1800. The highest four costs appear to be exceptionally high. If these four are not considered, the top end of the range is lowered to about $925, almost in half.

Controls costs, when normalized by the number of actuators controlled, reflect less variation than when normalized to other parameters such as collector area. Thus the number of actuators is a better gauge of controls costs.

Controls and electrical costs combined represented between 1 and 13% of the total system cost with an

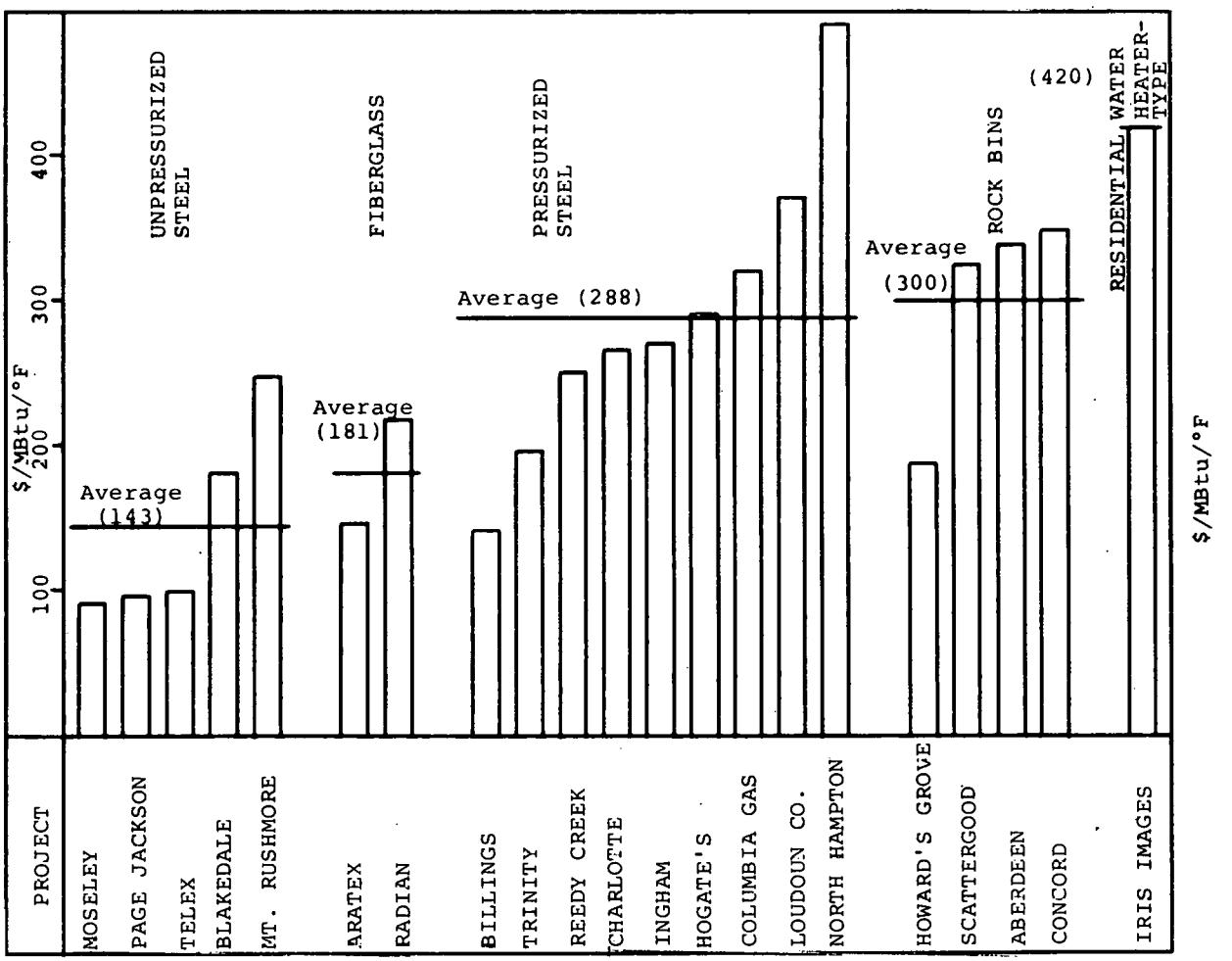


Fig. 8. Storage Costs by Type of Storage Container

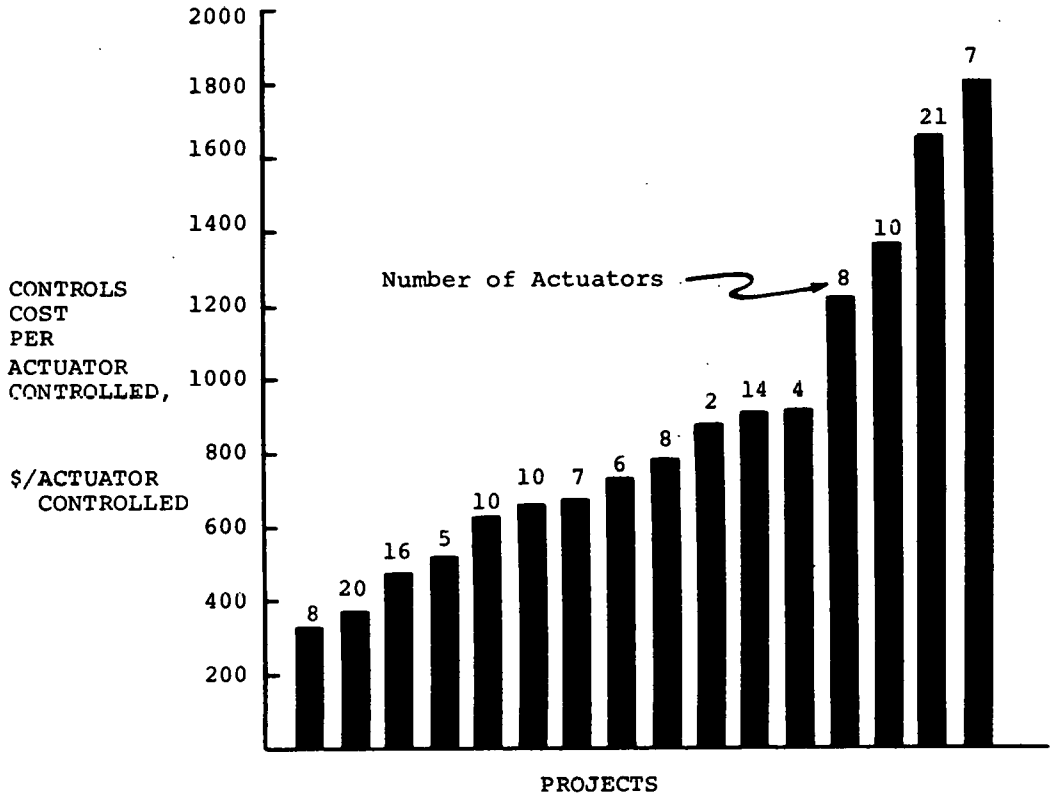


Fig. 9. Controls Costs per Actuator Controlled

average of 7% as shown in Table 3. Controls and electrical costs averaged $4.40/sq.ft. of collector area for the systems studied.

COST EFFECTIVENESS ANALYSIS

General

As has been stated previously, "cost per unit collector area" has definite shortcomings when used to compare solar energy system costs, because it does not account for system performance. A relatively low cost system may actually provide more energy than another, more expensive system. A more complete evaluation of solar energy systems then must take into account the thermal performance.

All systems discussed in this paper are instrumented for performance monitoring by the NSDN. Adequate data to determine annual thermal performance has been obtained for nine commercial systems that do not have major operational difficulties. Table 5 indicates that seven of these systems have twelve complete months of data available. For two other systems, Hogate's and Concord, data for four and three months, respectively, were extrapolated from available data through system simulation and engineering judgement to provide total annual performance values.

Within this section, several economic analytic techniques are utilized to investigate the cost effectiveness of nine solar energy systems. The simplest index of the cost/performance relationship is the ratio of total installed cost to the annual (first year's) solar energy system output. This ratio is useful only as an indicator for comparing systems. It has no counterpart in traditional economic analysis. However, the attractiveness of this approach is that direct outputs of the data program are used and no other assumptions are required. Other methods must assume inflation or interest rates or tax structures, etc. One of the disadvantages is that the ratio does not take into account the effect of operating energy. Figure 10 compares the system ranking under two ratios, total installed cost/collector area and total installed cost/first year energy output. It can be seen that the ranking of the systems changes significantly when energy output is considered. As a first order comparison of cost effectiveness, this simple index is very valuable.

Concord and Reedy Creek will not be analyzed further here because of their low comparative standing in Fig. 10.

Two more sophisticated, commonly used indexes for presenting the cost effectiveness of solar energy systems are payback period and rate of return. The payback period is the length of time required for the accrued net benefits from the solar energy system to equal the net capital investment. The rate of return is the interest rate at which the life cycle costs are equal to the life cycle benefits. Both of these quantities are highly dependent on a variety of economic assumptions.

Table 5. Performance Data Utilized in Economic Assessment

PROJECT	SYSTEM TYPE	NSDN MONTHLY PERFORMANCE REPORTS UTILIZED	EXTRAPOLATED DATA
KALWALL	Passive Space Heating	Sept., '78 - April, '79	
ARATEX	Process Hot Water	Oct., '78 - Sept., '79	
HOGATE'S		Mar., '79 - Oct., '79	November, '78 - February, '79
MOSELEY	Space Heating	Mar., '78 - Feb., '79	
CONCORD		Oct., '78; Dec., '78 - Apr., '79	Sept., '78; Nov., '78; May, '79
SCATTERGOOD		Sept., '78 - Aug., '79	
IRVINE	Space Heating and Cooling	Nov., '78 - Oct., '79	
PAGE JACKSON		Nov., '78 - Oct., '79	
REEDY CREEK		Apr., '78 - Mar., '79	

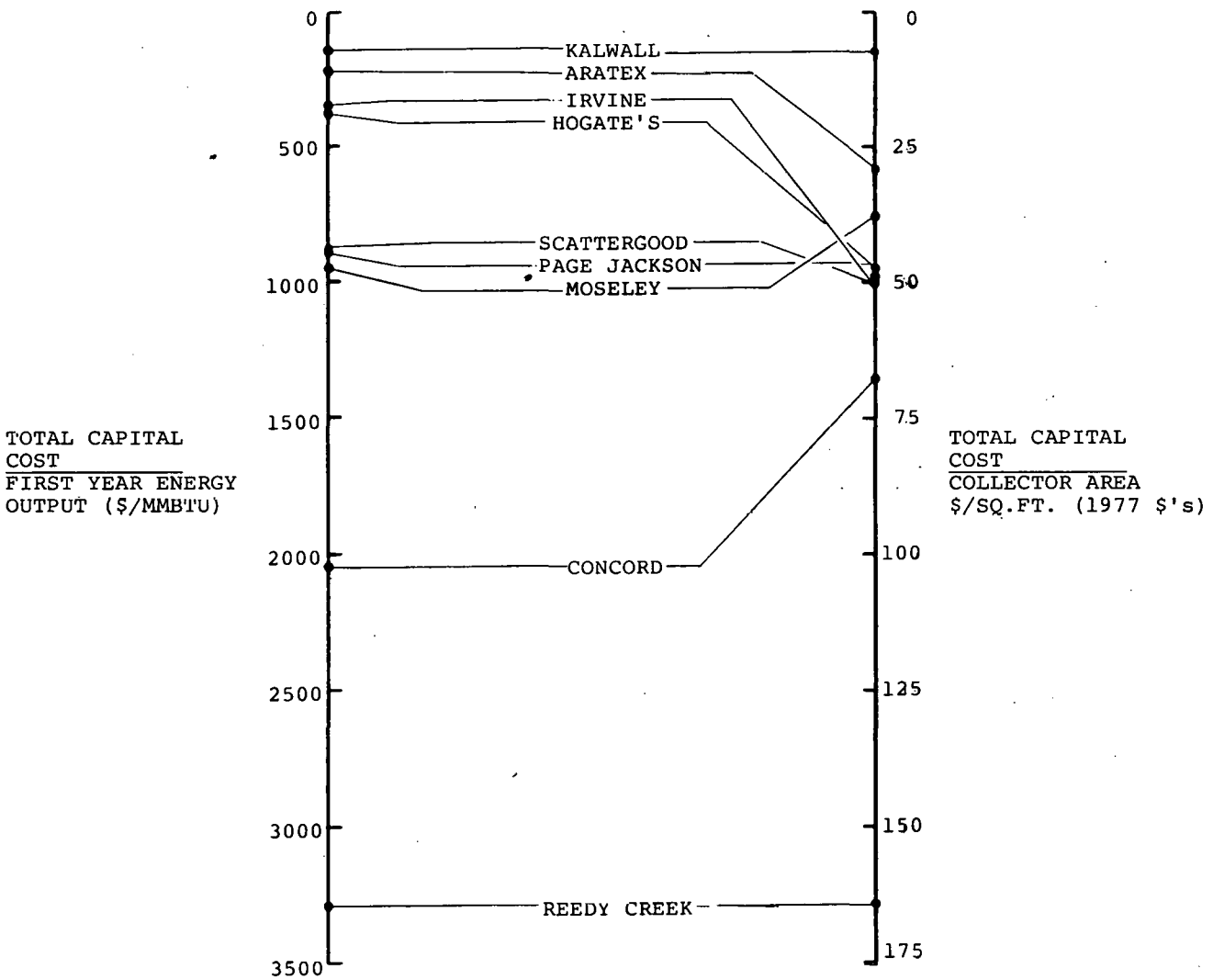


Fig. 10. Comparison of Cost/Collector Area to Cost/First Year Energy Output

The key variables used in the payback and rate of return analyses and also in the first year cost analysis discussed later are listed in Table 6. The nominal values were used in each analysis except where otherwise noted. A zero percent property tax rate was used because most states have passed property tax exemptions for solar equipment. The investment tax credit rate used was 20% (10% normal investment tax credit and an additional 10% now allowed for solar investments).

Table 7 lists the values used in the analyses for capital investment, solar energy used, cost of operating energy, and auxiliary fuel type for the seven systems.

PAYBACK PERIOD

The payback period analyses are based on the equation,

$$\sum_{n=1}^{k} \Big\{ Q\,(Fp)\,(1-\tau)\,(1+e)^n + \tau(rfr + d)\,(CI) - (1-\tau)[\beta_1 + \beta_2 + m\,(1+g)^n]\,(CI) - (1-\tau)O(1+e)^n \Big\} / (1+1)^n = CI - \alpha\,(CI).$$

Briefly, the term in brackets represents the net annual benefit contributed by the solar energy system and the right hand side of the equation represents the capital investment for the solar energy system less the investment tax credit.

Using the nominal values listed in Tables 6 and 7, the only solar energy system that achieved payback within the system lifetime was the Kalwall system, which achieved payback in 13 years. (For the Kalwall case, an investment tax credit rate of 10% was used, since this passive site does not qualify for a solar tax credit under existing law.) If electricity and oil were used as the auxiliary fuel for Aratex, the solar system would pay for itself in 17 and 14 years, respectively.

The payback period calculation is sensitive to the assumed values of the economic parameters in Table 6. Some parameters, such as the fuel escalation and general inflation rates, cannot be predicted with any accuracy for more than a few years. Other parameters, such as "percent mortgaged" and maintenance costs, will vary from site to site. Still others, such as the investment tax credit, are subject to change through legislation. Thus, the accuracy of any economic analysis based on these parameters is subject to the accuracy of the parameters themselves.

Table 6. Economic Analysis Parameters

PARAMETER	SYMBOL	NOMINAL VALUE
o System and Mortgage Life	N	20 years
o Interest Rate	i, r	10%
o Income Tax Rate	τ	50%
o Property Tax Rate	β_1	0
o Insurance Rate	β_2	0.0025 x CI
o Investment Tax Credit	α	20%
o Percent Mortgaged	fr	40%
o General Annual Inflation Rate	g	8%
o Annual Maintenance Costs	m	0.01 x CI
o Efficiency Adjusted Fuel Costs	$\frac{Fp}{CRF}$	See below
o Capital Recovery Factor	CRF	0.1175
o Annual Depreciation	d	*
o First Year Fuel Cost**		
o Electricity		$14.65/mmBtu
o Oil		$10.42/mmBtu**
o Propane		$10.02/mmBtu**
o Natural Gas		$ 5.00/mmBtu**
o Annual Fuel Escalation Rate (includes g)	e	
o Electricity		15%
o Oil		16%
o Propane		15%
o Natural Gas		15%

CI = Capital Investment

* For the payback and first year energy cost analysis, a sum-of-the-years-digit depreciation rate was assumed where d for year n is $2\,\frac{(N-n+1)}{N\,(N+1)}$. A straight line depreciation was used in the rate of return analysis.

**Based on 60% fuel efficiency.

NOTE:
The nominal values listed were meant to be reasonable estimates and no discussion is provided here regarding their appropriateness. The sensitivity of the results to many factors is discussed, however.

Table 7. Cost/Performance Values Utilized in the Analysis

SITE	CAPITAL INVESTMENT, CI (1979 $'s)	ANNUAL SOLAR ENERGY USED, Q_1	AUXILIARY FUEL	ANNUAL OPERATING ENERGY COST, O* (1979 $'s)
SCATTERGOOD	$135,452	154 Btu x 10^6	Propane	$ 44
IRVINE	339,393	976	Gas	1978
KALWALL	16,101	111	Oil	117
MOSELEY	17,011	18	Gas	44
HOGATE'S	337,906	915	Gas	440
PAGE JACKSON	626,945	703	Oil	396
ARATEX	219,246	949	Steam**	293
CONCORD	141,297	69	Electricity	132
REEDY CREED	744,922	227	Steam	513

* Cost is based on $14.65/mmBtu for electricity.
**Analysis for Aratex used variously electricity, gas, and oil as the auxiliary fuel type.

A series of graphs are presented showing the sensitivity of the calculated payback period for the Aratex system to the system cost, the investment tax credit, the interest rate, the fuel escalation rate, and the percent mortgaged. These graphs were developed by holding all parameters at the nominal values and varying one parameter over a range of values. Three curves are shown in each graph, one with electricity as the auxiliary fuel source, another with oil, and the third with natural gas.

The graphs are presented as Figures 11, 12, 13, 14, and 15. Note that in each figure, the sensitivity of payback period to the varied parameter is most pronounced in the case where gas is the auxiliary fuel type. The effect of varying each parameter becomes greater the further in the future that payback is achieved.

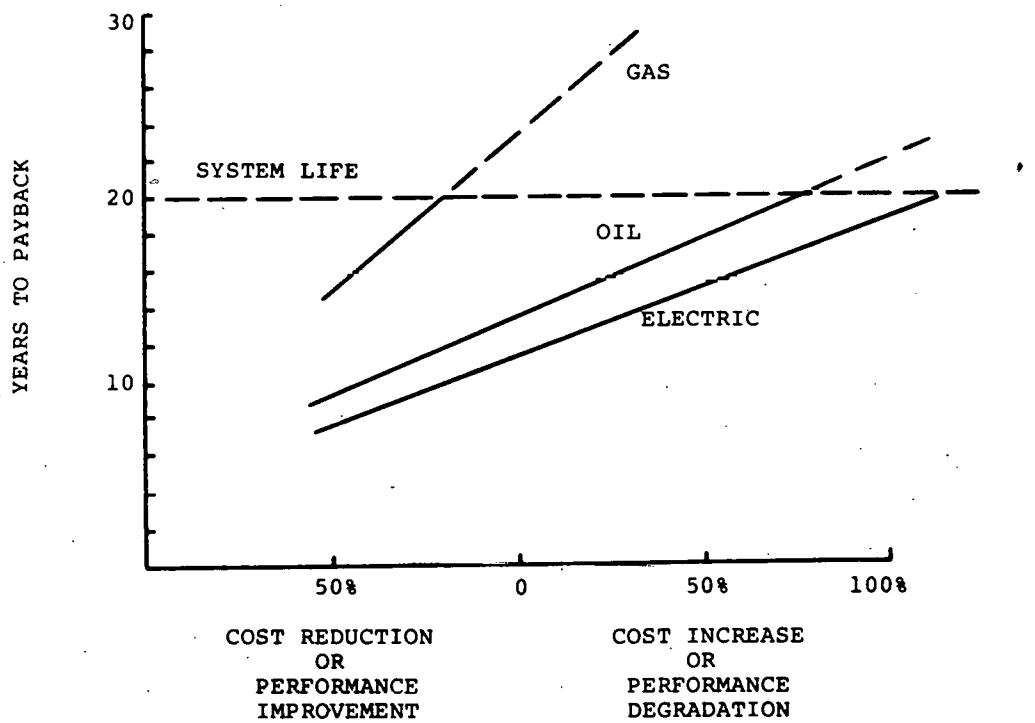


Fig. 11. Sensitivity of Payback Period to Changes in Initial Cost and Performance for Three Fuel Types for Aratex

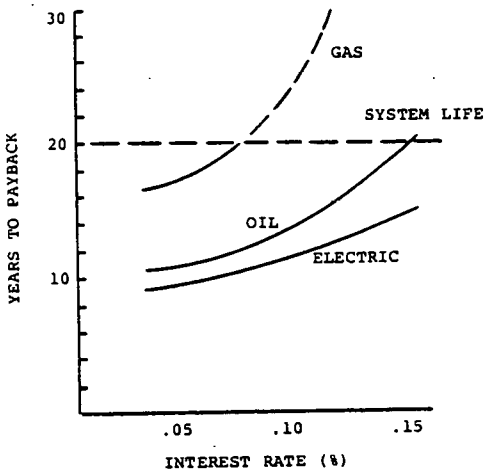


Fig. 12. Sensitivity of Payback Period to Interest Rate for Three Fuel Types for Aratex

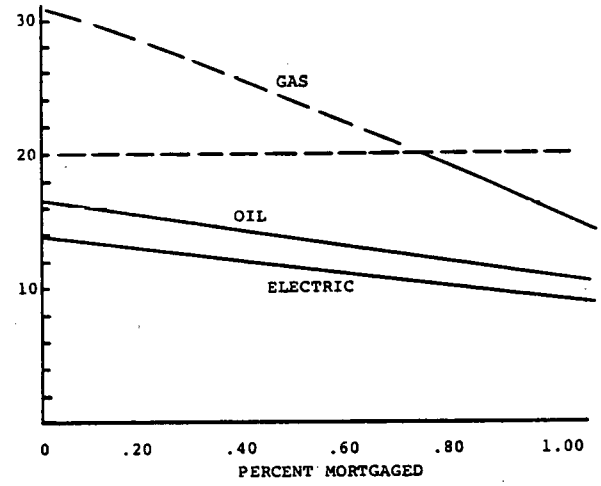


Fig. 13. Sensitivity of Payback Period to Percent Mortgaged for Three Fuel Types for Aratex

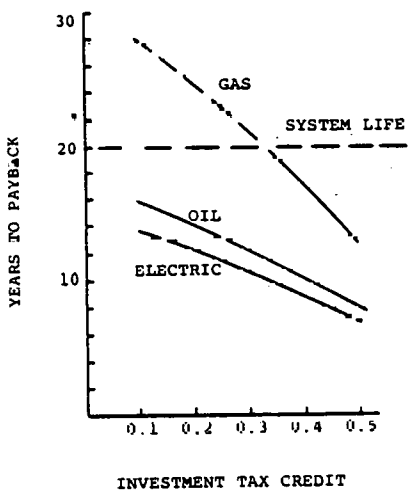


Fig. 14. Sensitivity of Payback Period to Investment Tax Credit for Three Fuel Types for Aratex

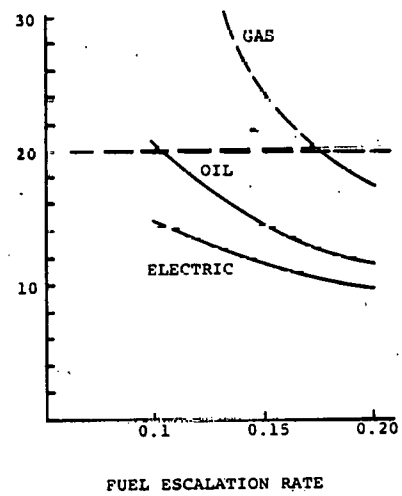


Fig. 15. Sensitivity of Payback Period to Fuel Escalation Rate (Includes General Inflation) for Three Fuel Types for Aratex

RATE OF RETURN ANALYSIS

Rate of return analysis is a method appropriate for evaluating and presenting life cycle costs of solar energy systems. The rate of return (ROR) can be defined as the interest rate at which the solar system life cycle costs are just equal to the life cycle revenues. The life cycle costs include the downpayment, mortgage payments, maintenance and operating expenses adjusted for appropriate tax credits, interest and business expense deductions, and fuel escalation and inflation rates. The life cycle revenues are the yearly energy savings provided by the system and adjusted annually by the fuel escalation rate. This method recognizes the significant effect of the timing of cash flows and the time value of money on the cost effectiveness of an investment. It has been used extensively by the business community to evaluate investment alternatives and has great potential as an indicator of cost effectiveness of solar energy systems.

ROR expresses the cost effectiveness of a solar energy system in terms that the business community understands; and after all, a solar system represents a true long-term "investment".

The disadvantages of this method are that it is very sensitive to the fuel escalation rate and inflation rate chosen and it does not indicate the actual ROR for intermediate years within the investment analysis period.

A computer program, developed by Mueller Associates, was used to calculate the rate of return for each of the sites with performance data. The nominal values presented in Table 6 were used for each analysis. The appropriate fuel types were specified for each system except for Aratex where cases were run for oil, gas and electric auxiliary energy. The ROR was determined for the school systems both with and without taxes. These results are presented in Fig. 16.

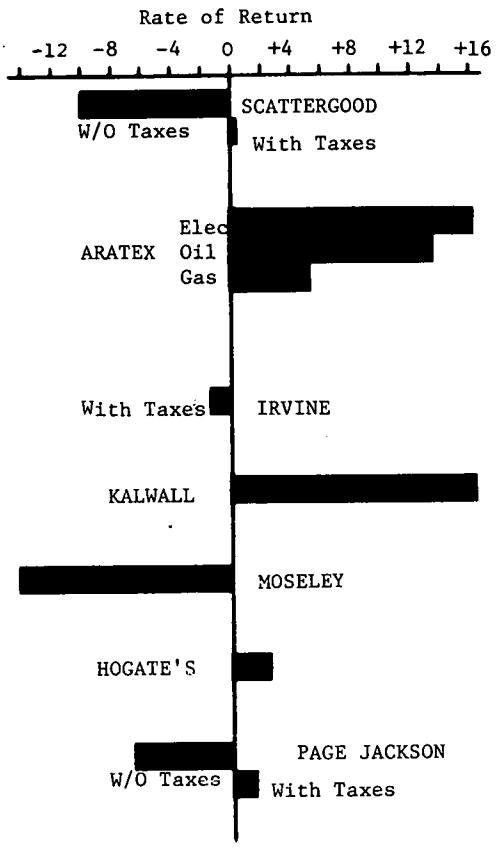


Fig. 16. Rates of Return for Each System with Nominal Values

It can be seen that the fuel type and the availability of tax credits can have an effect on the calculated ROR. By assuming that Page Jackson and Scattergood had both investment tax credits and interest deductions, their ROR's improved to positive values. Even with tax credits Irvine did not have a positive ROR (solar is competing against gas in this case and the Irvine base costs were high compared with its first year energy output.)

In order to compare all the system outputs and costs in relation to one another, the ROR of each system was calculated assuming the nominal values described in Table 6 and that tax credits were available to each system and that the auxiliary energy was electricity. Figure 17 presents the resulting system ROR's as a function of the total capital cost/first year energy output.

It is apparent that each system studied has a positive ROR when competing against electricity. It is also shown that the amount of operating energy used by a system has a significant impact on its ROR. As an example, consider the Hogate's and Irvine systems. Both of these systems have basically the same capital cost. However, Hogate's required only 30 mmBtu operating energy to produce the same amount of energy as Irvine which used 135 mmBtu operating energy. Consequently, Irvine has a 5.8 percentage point lower ROR than Hogate's. The band on Fig. 17 describes an estimate of the ROR of other systems not analyzed here based on their ratio of total capital cost to first year energy output and ratio of operating energy to first year energy output.

FIRST YEAR SOLAR ENERGY COSTS

Another method for determining the cost effectiveness of solar energy systems is the comparison of first year energy costs. These costs (P_{s1}) for the nine systems are presented in this section and are derived from the following equation:

$$P_{s1} = [\overline{CRF} + \beta_1 + \beta_2 + m_1 - \tau(\beta_1 + \beta_2 + m_1 + rfr + d_1)] \frac{CI}{Q_1} + (1-\tau)\frac{O}{Q_1} - \frac{\alpha CI}{Q(1+i)}[\overline{CRF} - \tau rf_r]$$

A description of these parameters and their nominal values are presented in Table 6 and 7, where the values for β_1, β_2, m_1, r, fr, d_1, and α are expressed as a fraction of the initial capital

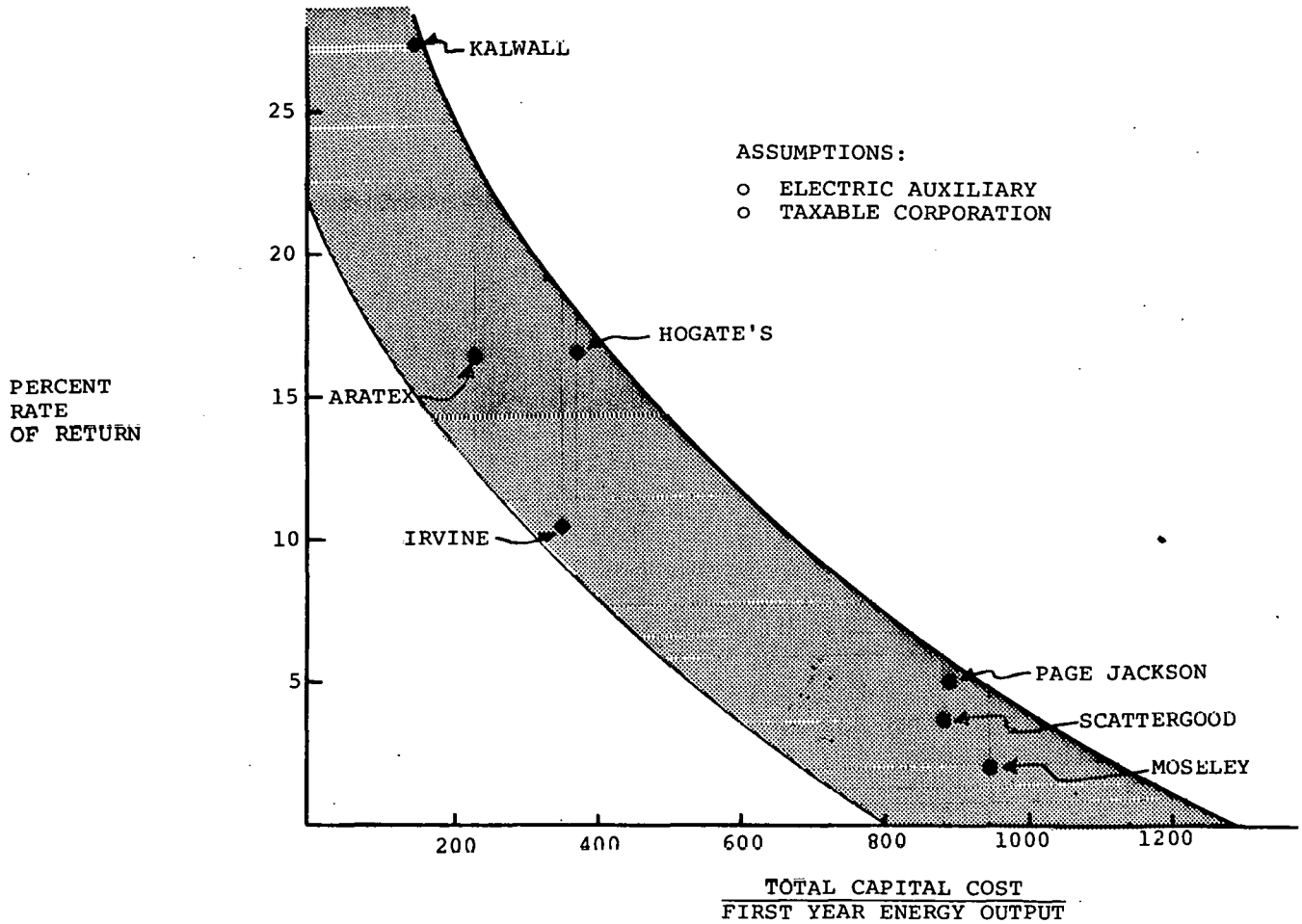


Fig. 17. Rate of Return for Seven Systems

investment. The first year energy cost method has a distinct advantage over other economic comparisons in that it does not depend on assumed fuel escalation rates or a general inflation rate over the life of the solar system. Since it does not include these highly speculative rates, this analysis can be based entirely on current data, prices, and rates. Although payback and rate of return methods may provide other comparisons, the first year cost is a valuable indicator of cost effectiveness. For further discussion of the first year energy cost method, see the papers by Sedmak and Ormasa listed in the Bibliography.

The first year cost of energy was calculated for each solar system for a number of different conditions, shown graphically in Fig. 18. The parameters varied were i (discount rate), τ (corporate income tax rate), and α (investment tax credit). For one case, costs were calculated with τ and α set equal to zero and all other parameters equal to their nominal values in order to simulate the financial situation of a public school. Three of the sites (Scattergood, Irvine, and Page Jackson) are schools. As can be seen from Fig. 18, the loss of all tax deductions and credits has a large impact on the first year cost of a solar system. This cost is on the order of 240% greater than the first year cost for the nominal case.

In the second case, τ and α were held equal to zero but i was lowered to 0.05. This simulates the case of a school receiving a low interest loan (5%) to finance its solar system. The effect of these values on first year energy costs can be seen in Fig. 18. This low interest loan reduces the cost considerably but it is still approximately 140% greater than for the nominal case.

For the case of corporate ownership (τ=.5), the loan interest rate (i) and the investment tax credit (α) were varied while all other parameters were fixed at their nominal values. As shown by Fig. 18, an interest rate of 15% produces a first year cost that is approximately 70% higher than for the nominal case. When either a low interest loan (i = .05, fr = .8) or a high investment tax credit rate (α = .5) is applied to the corporate case, dramatic cost reductions are obtained. A loan interest rate of 5% with 80% debt financing results in a first year cost that is approximately 21% as great as for the nominal case. When the interest rate is held at 10% but the investment tax credit rate is increased to 50%, a first year cost is obtained that is approximately 33% as great as for the nominal case. These reductions underline

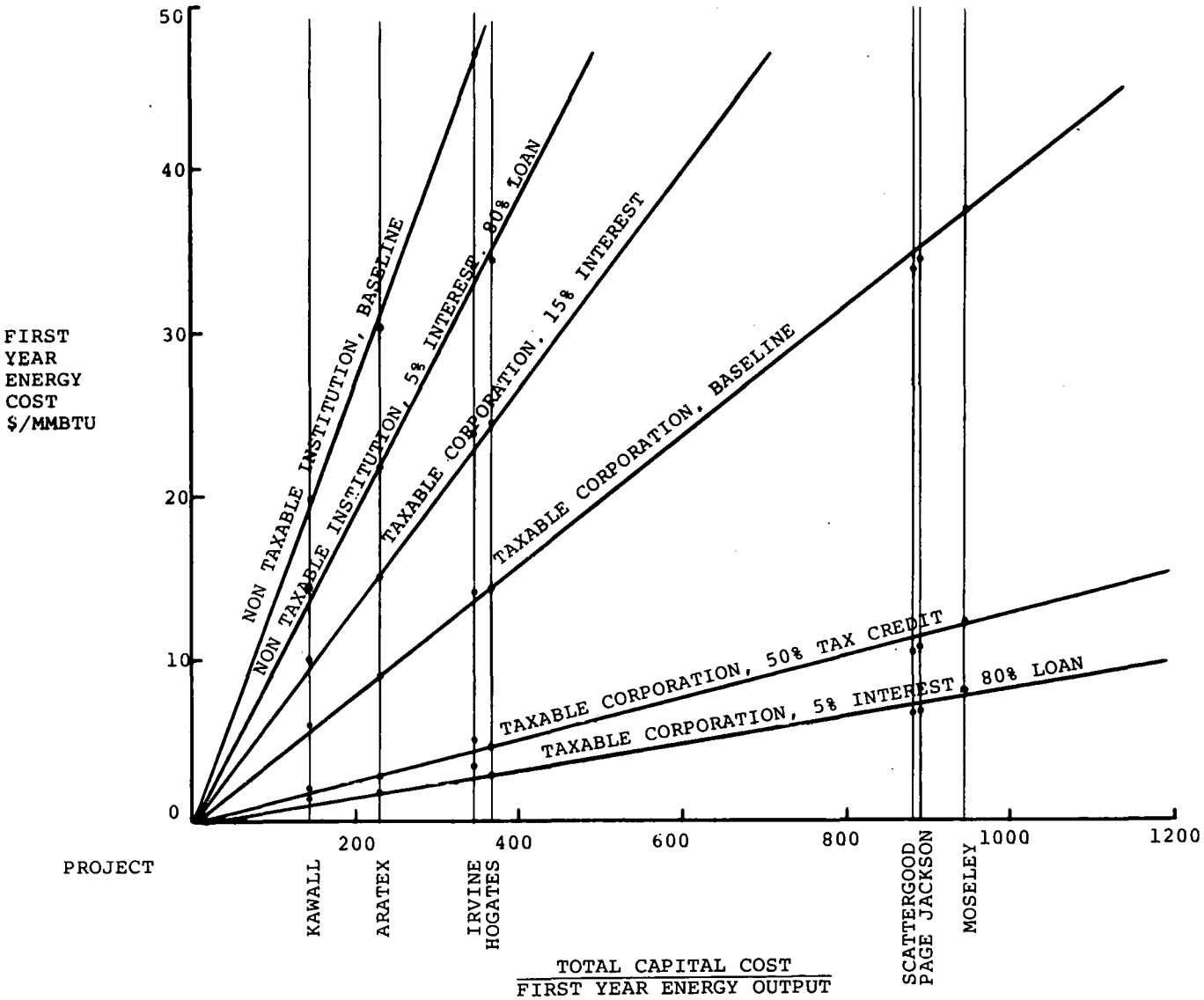


Fig. 18. First Year Cost of Energy for Different Economic Conditions

the tremendous impact that low interest loans or high investment tax credits could have upon the solar industry.

In using Fig. 18, find the cost of the competing fuel on the vertical axis. Note that for taxable corporations the cost of fuel is effectively halved due to tax deductions for operating expenses. Non-taxable institutions must compare their solar costs to the full competing fuel costs.

For the low interest case (i=.05, fr=.8), it can be seen that Kalwall, Aratex, Irvine, Hogate's, Scattergood, and Page Jackson would be cost competitive with electricity. Kalwall, Aratex, Irvine, and Hogate's would be cost competitive with electricity and fuel oil. Only Kalwall and Aratex would be competitive with electricity, oil, and natural gas.

For the high investment tax credit case (i=.10, α=.5), Kalwall, Aratex, Irvine, and Hogate's would be cost competitive with electricity and fuel oil. Only Kalwall would be competitive with electricity, fuel oil, and natural gas. By comparison, for the nominal case (i=.10, α=.2), the only cost competitive system would be Kalwall against electricity. This points out the great influence that loan interest, investment tax credit, and the tax situation can exert on the economic viability of solar energy systems.

The impact of improvements in system performance or reductions in system cost can be assessed by moving to the left as appropriate. For instance, if the Hogate's system could be installed for 37 percent less than it actually cost, its cost effectiveness would be equivalent to the Aratex system.

There has been some question regarding the relationship of federally funded projects to non-federally-funded projects. While little data is available to determine the relationship, the authors feel that except for cooling projects, the projects discussed in this paper represent the same technology and state of the art found in non-federally-funded projects. Most of the projects studied were designed in 1975 and 1976. It may be that lessons learned from these early projects have already assisted in the development of the state of the art.

SUMMARY AND CONCLUSIONS

System and subsystem costs are presented and analyzed for 24 commercial National Solar Demonstration projects. The cost data have been obtained through on-site investigations and review of construction contractor records. The distinction between this study and others is utilization of real cost data obtained after the fact for real systems.

Cost data have been combined with actual performance measurements for nine of the systems to obtain various measures of system cost effectiveness. The following are brief summarizations and conclusions from the findings of the study.

Costs

- System type has a major impact on system cost as summarized below:

SYSTEM TYPE	NO. OF SYSTEMS	AVERAGE COST
Passive	1	$7.70/sq.ft.
Process Hot Water	5	$41/sq.ft.
Space Heating	10	$48/sq.ft.
Space Cooling	8	$94/sq.ft.

  Large variations in total system cost were observed in all types. The reader is advised to be aware of the limited sample size and its implications regarding the validity of the averages presented.

- Retrofit systems were found to be more expensive than systems installed in new construction. The actual impact on system cost is difficult to determine because of the characteristics of the sample. Almost all of the cooling projects were retrofit. However, even when the cooling projects were not included, new systems were still approximately $4/sq.ft. less expensive than retrofit systems.

- Much of the added cost of retrofit construction has been identified with piping, ductwork and insulation categories.

- The support structure category was found to have the opposite relationship; that is, support structures on new buildings were more expensive than for retrofit installation. This is due to the added functions that the support structures are designed to perform on new buildings. These added functions include improving system appearance, housing equipment and providing reflectors. Approximately $5/sq.ft. average cost difference was found between single function and multiple function support structures. The support structure is 15% of the total system cost on the average.

- Economies of scale, within the range of systems studied, are a relatively unimportant cost factor.

- Regional cost variations, typical for mechanical systems, have not been found in the solar energy systems studied.

- The collectors represented on the average 34 percent of the total system cost, although the range was great. The average factory-built flat plate collector cost $16/sq.ft.

- The storage subsystem represented 7 percent of the total system cost on the average for the active systems that utilized storage. The type of storage vessel was found to be the major cost factor. Unpressurized steel tanks were least expensive and residential water heater type tanks were found to be most expensive per unit storage capacity.

- The controls cost for most of the systems varied between about $325 and $925 per actuator controlled.

Cost Effectiveness

- Nine systems were subjected to economic analyses to determine cost effectiveness.

- The relative ranking of the systems was found to change when systems were ranked in order of increasing cost per square foot of collector area and then total cost per first year energy output.

- Only the passive Kalwall system was found to pay back within the assumed system lifetime under expected conditions. The Aratex process water system would pay back if oil or electricity were the auxiliary fuel.

- The sensitivity of pay back to a large number of parameters is illustrated.

- In all economic analyses, non-taxable institutions are at a distinct disadvantage in the attempt to make solar energy cost competitive. This is due to the lack of tax deductions and tax credits in their economic analysis.

- Seven of nine systems studied had a positive rate of return when competing against electricity. Four systems had a positive rate of return against their existing competing fuel.

- Only Kalwall had a first year cost of energy competitive with its existing auxiliary fuel. However, if low interest loans were used or if 50 percent investment tax credits were given, four systems would be first year cost competitive. In addition to the passive Kalwall system, one of those four systems is a cooling project, and two are process hot water systems.

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NOTES

SUMMARY REVIEW: SOLAR PASSIVE SPACE HEATING SYSTEMS*

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What constitutes the performance evaluation of a passive solar space heated building? Such an evaluation should clearly include the following:

EVALUATION CRITERIA

- Thermal Performance
  - Solar Contribution
  - Energy Savings
- Livability
  - Temperature Indoors
    - Averages
    - Extremes
    - Daily Swings
  - Spacial Variations
  - Air Quality
    - Light
    - Ease of Operation
- Costs
  - Add-on Cost to Build
  - Operating Cost
  - Reliability/Longevity

The popular conception is that the evaluation base for passive solar buildings is very scant. By comparison with active solar systems this is undoubtedly so, but there does exist a significant evaluation base based on observations made in passive solar buildings over a period from 1969 to the present time. A few buildings were reported at this conference and a few have been reported at the First Operational Results Conference. In addition there is an extensive literature on passive solar buildings contained primarily in the proceedings of the four National Passive Solar Conferences which have been held and the passive solar sessions at the annual meetings of the American Section of the International Solar Energy Society.

I have not made a comprehensive search to catalog all of the information available for all passive buildings. A brief search provides the information which is contained in Table I giving a few figures of merit for different passive solar buildings. The column headings give the various appropriate values defined as follows:

1. SHF. Solar Heating Fraction. The contribution of the total building load which is supplied by solar energy, computed after the effect of normal internal energy sources such as lights and people have been accounted.

2. Efficiency. The fraction of the total solar radiation incident on the collection wall which is ultimately delivered to the house as a useful contribution to the heating load.

3. Temperature Swing. The peak-to-peak temperature range observed inside the passive solar building during sunny weather conditions.

4. Solar Contribution. The annual amount of heat delivered from the solar wall to the structure, Btu/yr-sq ft.

*Work performed under the auspices of the U.S. Department of Energy, Office of Solar Applications.

5. Solar Savings. The total amount of heating energy which is saved by the presence of the solar wall. The comparison here is with a conventional construction on the south side instead of the solar wall and with the building held at a normal comfort level.

Overall Evaluation.

It is clear that passive solar construction is thermally advantageous and questions pertaining to factors such as comfort, other livability considerations, and cost need to be evaluated building-by-building. Properly done, it appears that passive solar buildings can get high marks on all of the evaluation criteria. There are many examples, however, where one or another factor leaves something to be desired. A comprehensive evaluation of a large sample of buildings against all criteria has yet to be done. Such an evaluation will be difficult, lengthy, and expensive.

Results from buildings which have been monitored indicate performance which equals or exceeds the estimates obtained by simulation analysis. However in these cases the largest unknown is the actual building load; expected performance is extremely sensitive to thermostat setting. Some owners prefer to accept a lower thermostat setting in order to obtain an 85% solar fraction rather than a more conventional thermostat setting which would lead to a 50% solar fraction. However this is decreasingly so as more and more passive solar buildings are occupied by people who did not build them.

The general consensus, based on feedback from hundreds of passive buildings located all over the United States, is that passive solar performance generally exceeds expectations. Annual heating costs of $100 per season or less are commonly reported for moderate-sized passive solar buildings even in cold climates. Such results are certainly influenced by a variety of psychological and physiological factors. Some passive home dwellers (especially in the first generation) have adapted their lifestyles slightly, preferring a sweater to a normal thermostat setting, but this is becoming the exception rather than the rule. Most passive homes are now sold on the speculative market to buyers who are not necessarily solar advocates and it is both inaccurate and a poor strategy to market a passive solar house on the basis that any sacrifice in comfort is implied.

There is a growing body of evidence that correctly designed passive solar buildings have comfort characteristics which equal or exceed their non-solar counterparts. This is thought to be due to several subtle physiological factors. The most important of these is the effect of the thermal radiant environment on comfort. Under still-air conditions the radiant temperature is roughly equal to the actual air temperature in the individual's perception of comfort. In a passive building the enclosing surfaces of a room are usually warmer than in a conventional building which leads to an increase in radiant temperature. This increase in surface temperature results from two effects: well-insulated walls and multiple glazings (conservation measures) and storage of solar heat (passive solar measures). Thus the individual in a well-insulated passive house can experience an equivalent comfort sensation when the air is at 65 F as in a conventional home with a 72 F air temperature. A second effect which enhances overall comfort is the incredible thermal stabilty of a well-insulated building with a large internal thermal mass. Temperatures change very slowly resulting in less physical stress than in a building with the same average temperature but with abrupt cyclic variations.

In conclusion, I find that the early results are very encouraging but that continuing work needs to be done monitoring performance of actual passive solar buildings against the entire list of evaluation criteria which I have presented.

TABLE I SOME MONITORED PASSIVE BUILDINGS

	SHF	Eff.	ΔT Swing	Btu/sq ft-yr Contribution	Saved
Wallasey School	100%		7-16°		Nil
Trombe - Michel House	70%	36%			
Atascadero (Hay) House	(100%)*		2°		
• Balcomb House	89%	37%	5°	189000	118000
• Mobile-Modular II	80%	24%	10°	214000	78000
MIT 5	70%				
Fowlkes House	85%				
Kelbaugh House	80%		15°		
Hunn House	80%				
• Hamilton College	68%				
Green Mountain Home	39%			90000	
Saskatchewan House	73%		11°		105000
• Hullco	100%	25%	5°		73000
• Colorado Sunworks	76%	38%	6°		93000
• Kallwall Warehouse	58%	43%			54000
• Spence - Urban					
• Greenmoss					Negative
• Living Systems				81000	
• Lumsdaine House	75%				
NMSU Skytherm®			2°		

• Buildings reported at the Second Operational Results Conference.

* Data missing for two months in winter.

NOTES

Session IV: Solar Cooling Systems

THERMAL PERFORMANCE OF SPACE COOLING SOLAR ENERGY SYSTEMS IN THE NATIONAL SOLAR DATA NETWORK

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ABSTRACT

This paper presents results derived from analysis of data obtained from monitoring the operation of four solar energy cooling installations in the National Solar Network. It is shown that chiller coefficients of performance (COP) on the order of 0.65 can be easily obtained with existing technology, provided the designer adequately matches the solar energy system to the absorption chiller. It is also shown that flat-plate, concentrating and evacuated tube collectors may each be used successfully to operate absorption chillers in space cooling systems. The results show that appreciation for the systems engineering aspects of solar energy is extremely important to the development of a viable industry.

ACKNOWLEDGEMENTS

The assistance of Messrs. Ingle, Picking and Schmitt of Arkla Industries, Inc., Evansville, Indiana must be acknowledged. Without their patient assistance the performance of the absorption chillers discussed in this report could not have been evaluated. Independent evaluation of specific systems performed by Mr. Don Evans and Mr. Henry Smith of SAI and Mr. Warren Browning of IBM have also been incorporated in the conclusions of this paper.

INTRODUCTION

This paper presents thermal performance data for several solar energy cooling systems. The thermal performance data presented was obtained by the National Solar Data Network (NSDN) which is a part of the National Solar Heating and Cooling Demonstration Program. The performance of four space cooling systems which use single-effect, lithium bromide-water, absorption chillers, is discussed in order to quantify the overall system performance. Detailed performance of the two most critical components for a cooling system, the collectors and chiller, are discussed at length.

The results show that although the performance of the cooling systems have been less than expected most systems, the causes of this low performance are characteristic of all active solar energy systems and are related to the control system design. The problems can be avoided in the future only if information gained from experience is shared freely. This report attempts to share the experience gained from monitoring the performance of cooling systems in the National Solar Data Network.

COOLING SITES IN THE NATIONAL SOLAR DATA NETWORK

At four sites in the National Solar Data Network, all or part of the building's cooling load was satisfied using solar energy during June, July and August of 1979. These four sites, all located in the southern part of the United States, are briefly described in Table 1. The references contain sources of additional information as to the description, cost and performance of these sites. These sites represent a wide range of system designs and include a variety of collector types, various storage capabilities, three sizes of absorption chillers and several methods for auxiliary cooling. All sites, Dallas Recreation Center and Reedy Creek, provide space heating and domestic hot water heating in addition to space cooling while Irvine and Page Jackson schools do not provide domestic hot water heating.

Table 1 also presents the average cooling produced each daily for each square foot of collector during June, July and August of 1979. The performance summarized in these tables is presented in detail in the Appendix and will be discussed in the following section.

SYSTEM DESCRIPTIONS

Dallas Recreation Center

This solar energy system is installed in the North Hampton Park Recreation and Health Center in Dallas, Texas. The building contains an area of 16,000 square feet, which provides room for an 8,000-square foot gymnasium, a locker room and a health care clinic. The solar energy system is designed to provide 30 percent of the annual space heating, 48 percent of the annual space cooling and 90 percent of domestic hot water requirements.

The solar energy system contains 238 single-glazed, flat-plate collectors, manufactured by Honeywell, Inc., providing a gross area of 3,650 square feet. The collectors are mounted in 29 arrays on the roof

TABLE 1. COOLING SITE DESCRIPTION AND PERFORMANCE SUMMARY

Site	Collector Type	Collector Area (ft^2)	Storage Volume (Gal)	Chiller Capacity (Tons)	Auxiliary Cooling Method	Cooling Produced (Btu/ft^2-day)[1]
Dallas Recreation Center Dallas, TX	Flat-plate, Single-glazed, Selective	3,650	6,000 Hot 2,000 Cold	25	Electric (Vapor Compression)	96
Irvine School Irvine, CA	Evacuated Tube, Reflector	4,950	None	200 (Two Chillers at 100 tons each)	Natural Gas	218
Page Jackson Charles Town, WV	Flat-plate, Double-glazed, Selective Absorber, Reflectors	11,000 10,000	20,000 Hot	100 (Derated to 33 Tons)	Electric (Centrifugal)	54
Reedy Creek Utilities Lake Buena Vista, FL	Tracking Absorber, Fixed Reflector	3,840	10,000 Hot 10,000 Cold	25	Central Chilled Water Plant	69

[1] Cooling produced is rated at Btu of cooling produced each day per square foot of collector area.

of the building and face south. The collectors are tilted at an angle of 25 degrees from the horizontal. The heat transfer medium is an aqueous solution of 35 percent ethylene glycol and water.

Space cooling is supplied by using solar energy to operate an ARKLA absorption chiller. Chilled water is stored in a 2,000-gallon tank located above ground in the mechanical room and insulated with four inches of urethane. Auxiliary space cooling is provided by two vapor compression units.

Irvine School

The El Camino Real Elementary School, also known as Irvine School, is located in Central Orange County, California. The school is approximately ten miles from the Pacific Ocean and was built in 1971. The building contains 41,109 square feet of floor area and is normally occupied five days a week by 850 children and 60 adults. Building occupancy is uniform throughout the year since the school operates on a trimester plan. The solar energy system was added to the existing building and was designed to supply 50 percent of the annual building heating and cooling demand.

The solar energy installation includes 4,950 square feet of Owens-Illinois evacuated tubular-glass collectors, a heat rejector and a heat exchanger from the collector loop to the load loop. The collector array faces south at an angle of 25 degrees from the horizontal. Water is used for delivering solar energy from the collector array to a heat exchanger for transfer to the load subsystems. Existing load loop components were unaltered except for controls. These load loop components include a four million Btu per hour natual gas boiler, two 100-ton absorption chillers, 41 heating coils and seven air handlers. Since heating demands are low due to the moderate climate and most of the load demand occurs during the day, no storage is provided. The collected solar energy is transferred directly from the collector loop to the load loop via the heat exchanger. If collected energy exceeds the load loop demand, excess energy is rejected via a heat rejector. The heat rejector is a water-to-air heat exchanger mounted on the roof and situated in the collector loop between the collector outlet and the load loop heat exchanger inlet. If the solar energy does not meet the full energy demand from the loads, a boiler is activated to make up the shortage before the hot water reaches the loads. The hot water is routed to the chillers or heating coils, or both, depending on the building demand. The hot water then returns to the heat exchanger to complete the cycle.

Page Jackson School

Page Jackson School is an elementary school located in Charles Town, West Virginia. The solar energy system is designed to provide approximately 85 percent of the space heating and 50 percent of the space cooling energy requirements of the school. It has an array of Pittsburgh Plate Glass Co. flat-plate collectors, model A529, with a gross area of 11,000 square feet that faces south at an angle of 45 degrees from the horizontal. Water is used as the medium for delivering solar energy from the collector array to storage. The solar heated water is stored in two interconnected 10,000-gallon storage tanks and is used for space heating and cooling. When the solar energy is insufficient to meet the heating demands, an oil-fired boiler is used to provide auxiliary hot water for heating. In the space cooling mode, the hot water from storage is supplied to an absorption chiller to generate chilled water. A conventional centrifugal chiller is used as backup whenever solar energy is insufficient to meet the space cooling demand. Operation during the summer months of 1979 has resulted in the solar energy system supplying the major portion of the energy to provide space cooling.

Reedy Creek Utilities

The Reedy Creek Utilities site is a two-story, 5,625 square foot concrete block office building located in Lake Buena Vista, Florida. The solar energy system is designed to provide space heating, domestic hot water and space cooling. The collector subsystem is composed of a horizontal array of 16 parabolic trough collectors, manufactured by AAI Corporation, with tracking absorber tubes. The collector subsystem is oriented with its major axis in an east/west direction. The total collector aperture area is 3,840 square feet. Water is used as the energy collection, transfer and storage medium. Collected solar energy is stored in a 10,000-gallon hot water tank, located adjacent to the building and shaded by the roof.

A 25-ton absorption chiller utilizes hot water from solar storage to provide chilled water to a 10,000-gallon cold water storage tank. For space cooling, water from this cold tank is circulated through heat exchangers located in the building's central air distribution system. Auxiliary cooling is provided by supplemental cold water from the utility district's central chiller plant, which is powered by fossil fuels.

COMPONENT PERFORMANCE

In designing or evaluating the performance of a solar energy space cooling system, two components cause difficulties in addition to those encountered when designing or evaluating solar energy systems for space heating or domestic hot water heating. First, because of the higher system temperatures nally required for space cooling, the collector ay must be able to efficiently collect energy while operating at temperatures in excess of 160°F. Because the efficiency of collection at these higher temperatures can be significantly below the collection efficiency of space heating or domestic hot water heating system, the collector must be selected with care. In addition, the chillers which are currently available are more complex than other commonly used solar energy system components. These chillers, which are single-effect, lithium bromide-water, absorption chillers contain their own internal control system, therefore the control system for the solar energy system must be designed with care to ensure that cooling is produced efficiently. The performance of a particular energy collection and storage subsystem and an absorption chiller will be discussed to illustrate these two observations. Data from the other systems will be presented for comparison but without elaboration.

Collector Performance

The thermal performance of all collectors in the NSDN has been evaluated based on actual data obtained for each site using a method similar to that which has become an industrial standard [Reference 5]. The methods differ in that the industrial standard is based on controlled tests of single collector panels under steady-state laboratory conditions and the method used in the NSDN is based on data obtained from collector arrays operating in the field under transient conditions. The method used in the NSDN has previously been described [Reference 6] and results have been presented for more than 40 collector arrays [References 7 and 8]. The NSDN collector results present the collector array efficiency as a function of collector operating point as defined in Reference 5.

Typical average operating points during June, July and August for Dallas Recreation Center, Irvine School, Page Jackson School and Reedy Creek Utilities are also presented in Table 2. Although the average operating point is approximately 0.35°F ft^2-hr/Btu, it has a daily variation caused by changing temperatures and insolation of between 0.2 to 0.6°F ft^2-hr/Btu [References 7 and 8].

TABLE 2. COLLECTOR OPERATING POINTS*

SITE NAME	OPERATING POINT (°F-FT^2-HR/BTU)
Dallas Recreation Center	0.35
Irvine School	0.35
Page Jackson School	0.40
Reedy Creek Utilities	0.35

* Collector Operating Point [Reference 5] =

$$\frac{\text{Collector Inlet Temperature} - \text{Ambient Temperature}}{\text{Total Insolation in Collector Plane}}$$

The collector array efficiencies derived from data collected by the NSDN for the evacuated tube collector at Irvine School and the concentrating collector at Reedy Creek Utilities and the two flat-plate collectors at Dallas Recreation Center and

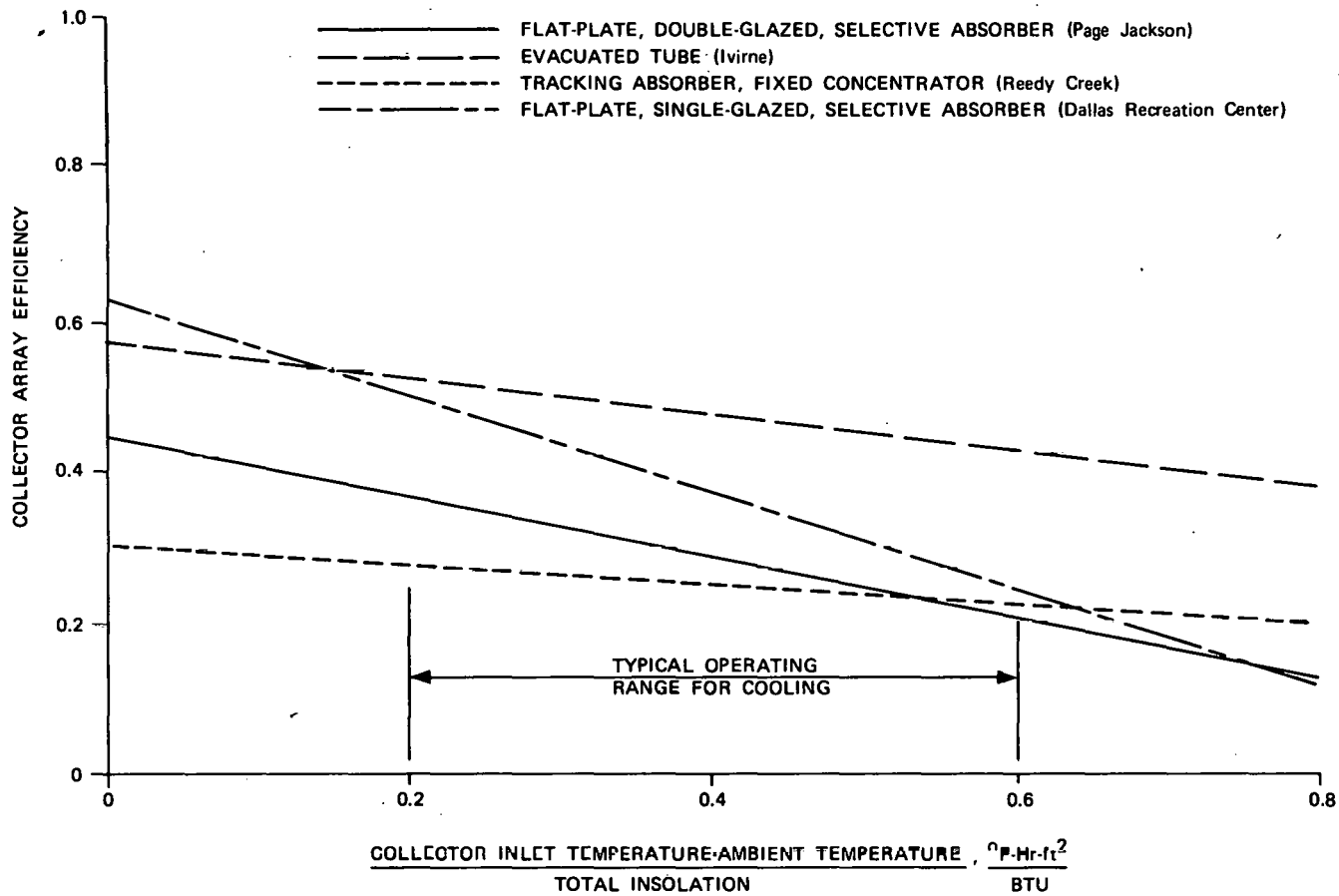


Figure 1. Collector Array Efficiency

Page Jackson School are shown in Figure 1. The data shown in the figure agrees with that supplied by the collector manufacturers within 10 percent which compares favorably with the results for other collectors evaluated in the NSDN. Figure 1 also shows that between 25 and 50 percent of the insolation incident on the collector plane while the collector pump is operating can be collected by a wide variation of collector designs. These efficiencies are higher than those presented in the Appendix because those efficiencies are based on the total insolation on the collector plane during the day and include the energy incident on the collector when the collector was not able to increase the temperature of the heat transfer fluid. The cost of the collectors whose efficiencies are shown in the figure vary significantly; and, as a general rule, the higher collector efficiencies require a greater initial investment. Therefore, the actual collector selected for a site should be based on a design trade between collector efficiency and collector cost, including maintenance. However, Figure 1 does show that several collectors can be used for space cooling systems and the choice is not limited to a particular type.

Absorption Chiller Performance

An absorption cycle chiller contains two internal circulation loops (refrigerant and absorbent) and three external circulation loops (hot water, chilled water and condensing water). Most absorption chillers contain internal controls which monitor the temperatures of these external circulation loops and control the flow of the internal loops. This internal control system is used to maximize the chillers performance and to prevent damage to the chiller when these temperatures rise above or fall below the chiller's operating range. Most system designers recognize the minimum temperature requirement of the energy supplied to the generator; however, many do not recognize the effects on the chiller's performance due to variations in the temperatures of either the chilled water or the condensing water. Two sites, Dallas Recreation Center and Irvine School, have been selected to illustrate these effects.

The performance of the 25-ton chiller at the Dallas Recreation Center site is shown in Table 3 for May 17, 1979. On this day, all conditions were essentially constant except the temperature (T_g) of the hot water entering the generator which decreased slightly at the end of the day and caused a slight decrease in the amount of cooling produced. Under the conditions experienced on May 17, the chiller averaged producing approximately 10.6 to of cooling each hour or slightly better than tha expected by the chiller manufacturer with 164°F water entering the generator (T_g), 82°F condensing

TABLE 3. DALLAS RECREATION CENTER ABSORPTION CHILLER PERFORMANCE - MAY 17, 1979

Hour Ending	T_g (°F)	T_c (°F)	$(T_e)_{in}$ (°F)	$(T_e)_{out}$ (°F)	COP	Q_g[1] (Btu/hr)	Q_c[2] (Btu/hr)	Q_e[3] (Btu/hr)	Q_e[3] (TONS)	Energy Imbalance (Btu/hr)
12 NOON	165	82	54	48	0.68	0.192 x 10^6	0.318 x 10^6	0.129 x 10^6	10.75	0.003 x 10^6
1 PM	166	82	55	49	0.69	0.199	0.338	0.138	11.50	-0.001
2 PM	166	82	56	50	0.70	0.200	0.344	0.140	11.67	-0.004
3 PM	166	82	57	51	0.69	0.202	0.334	0.140	11.67	0.008
4 PM	164	82	57	51	0.72	0.181	0.315	0.130	10.83	-0.004
5 PM	161	82	56	51	0.69	0.159	0.276	0.110	9.17	-0.007
6 PM	158	81	55	51	0.65	0.034	0.056	0.022	8.46	0.0

[1] Generator Input Energy
[2] Energy Rejected to Cooling Tower
[3] Evaporator Input Energy (Cooling Produced)

water (T_c), 55°F chilled water entering the evaporator $(T_e)_{out}$ and 50°F chilled water exiting the evaporator $(T_e)_{out}$. Also shown in the table are the results of an energy balance on the absorption chiller. This balance was obtained by adding the energy input to the generator and evaporator and subtracting the energy rejected by the cooling tower. The maximum imbalance (due to either chiller transients or measurement errors) is less than 2.5 percent of the energy being transferred through the chiller. This close agreement implies a high confidence in the data system.

In addition to low chilled water temperatures decreasing the performance of an absorption chiller, the condensing water temperature entering the chiller (T_c) can also affect the chiller's performance. Table 4 presents the Irvine School's absorption chiller performance for March 5, 1979. As can be seen from the table, the COP of the chiller was below 0.5 until 12 noon even though the hot water temperature (T_g) was approximately 190°F and the temperature of the chilled water exiting the evaporator $(T_e)_{out}$ was greater than 50°F. It is important to note, however, that the condensing water temperature (T_c) was below 80°F until 12 noon. The internal controls for this chiller monitor the condensing water temperature and bypass the absorbent around the absorber if this condensing water temperature is less than 80°F, thus causing a significant reduction in the amount of cooling produced. Under full 100-ton load conditions, sufficient energy would be rejected by the cooling tower in the first hour of operation to quickly raise the condensing water temperature to 80°F. However, under the partial load conditions caused by operating chiller at below the design hot water loop temture (T_g) of 230°F, four hours were required to raise the condensing water temperature above 80°F. The performance of this chiller can be increased by presenting the condensing water temperature to remain below 80°F for this extended period.

The thermal performance of the energy collection and storage subsystem (ECSS) and the absorption chillers at the four cooling sites are presented in Table 5 and Table 6. This summary information has been compiled from data taken on the NSDN and rep-

TABLE 4. IRVINE SCHOOL ABSORPTION CHILLER PERFORMANCE - MARCH 5, 1979

Hour Ending	Cooling Produced (TONS)	COP	T_g	$(T_e)_{out}$	T_c
8 AM	5.2	0.05	180	49	68
9 AM	7.8	0.07	188	53	72
10 AM	19.2	0.18	190	53	75
11 AM	35.5	0.37	189	56	77
12 NOON	41.1	0.49	190	60	84
1 PM	44.6	0.55	190	61	84
2 PM	46.6	0.53	190	61	83
3 PM	43.4	0.51	190	61	83
4 PM	43.5	0.52	189	60	83
5 PM	42.1	0.59	189	60	83
6 PM	34.2	0.52	188	59	82

TABLE 5. AVERAGE COLLECTOR AND STORAGE SUBSYSTEM PERFORMANCE - JUNE, JULY AND AUGUST 1979.

SITE NAME	TOTAL AVAILABLE INSOLATION (BTU/FT^2-DAY)	COLLECTED STORED ENERGY (BTU/FT^2-DAY)	TOTAL SOLAR ENERGY UTILIZED (BTU/FT^2-DAY)	THERMAL LOSSES (BTU/FT^2-DAY)	AVERAGE COLLECTOR OPERATING PT. (°F-FT^2-HR/BTU)	COLLECTOR EFFICIENCY	AVERAGE TEMP. (°F)	AVERAGE DAYTIME TEMP. (°F)
Dallas Recreation	1,854	542	460	--	0.32	28	83	92
Irvine School	1,970	759	557	--	0.37	39	73	84
Page Jackson	1,483	219	128	88	0.40	14	71	79
Reedy Creek	1,681	201	123	82	0.35	12	82	91

TABLE 6. PERFORMANCE OF ABSORPTION CHILLERS - JUNE, JULY AND AUGUST 1979.

SITE NAME	CHILLER INPUT	COOLING PRODUCED	COEFFICIENT OF PERFORMANCE	AVERAGE COOLING SOLAR FRACTION
Dallas Recreation	460	96	0.366	13
Irvine School	557	218	0.339	21
Page Jackson	128	54	0.423	73
Reedy Creek	118	69	0.585	31

resents average data for these four sites for June, July and August 1979.

It may be observed that the Irvine School chiller system operated at an average COP of only 0.37 and supplied but 13 percent of the cooling load, when the design called for 50 percent. While it was observed that the available insolation was 90 percent of the predicted long-term insolation, the primary cause of the low COP and resultant low solar fraction was the abnormally low condensing water temperature as previously discussed.

The thermal performance of the Reedy Creek solar energy system has been monitored and reported in the NSDN since March 1978. During the time period June, July and August 1979, the cooling load was essentially 100 percent of the total load except for minor hot water heating requirements. The solar energy system satisfied 31 percent of these space cooling demands. While the chiller COP at Reedy Creek was nearly twice that of the Irvine chiller array, the significantly lower collector array efficiency kept the cooling produced, normalized to size of the collector array, to less than one third of Irvine School. Basic design philosophy has also affected the cooling produced per square foot of collector since Irvine utilizes neither hot or cold storage and Reedy Creek uses both.

Although chiller performance has been presented in this section which is below that expected, based on manufacturer's data, in all instances the low performance was caused by the designers lack of consideration of the chiller's internal control system. A COP of 0.6 to 0.7 can easily be obtained for an absorption chiller activated by solar energy. This will require, however, recognition of the complexities of the chiller during the design of the solar energy system. In addition to recognizing the influence of the temperature of the hot water supplied to the generator, the influences of the chilled water temperature and the condensing water temperature on the chiller's performance must be recognized. If these influences are recognized, a COP of 0.6 to 0.7 can easily be obtained.

SUMMARY

Performance data from four space cooling sites in the Solar Data Network have been presented. These data show that the operational problems with cooling sites are caused by the same factors which cause operational problems with heating and hot water sites. The factors which cause the majority of the operational problems in all active solar energy systems are not related to the individual components, such as collectors, storage tanks or ab-

sorption chillers, but are related to operational problems caused by inexperience in integrating components into a solar energy system.

This integration requires a control system which must be designed using an overall system approach to ensure the system efficiency is maximized. This task becomes more difficult when designing a space cooling system because the absorption chillers currently used in such systems typically contain an internal control system. The performance characteristics of these chillers must be well understood to correctly interface the chiller with the solar energy system.

Performance results have been presented which show that if proper attention is given to the interfacing of the solar energy system with the absorption chiller, a coefficient of performance of 0.6 to 0.7 can be easily obtained with solar energy temperatures as low as 160°F. However, in many instances, either the chiller's performance characteristics were not recognized or the chiller's internal control system was not functionally compatible with the solar energy control system and the actual coefficient of performance was much lower. This low performance could have easily been avoided if these factors had been recognized.

Three collectors were shown to be capable of operating at the high temperature requirements of cooling system. The efficiency of these collectors, when operating in a cooling system, ranged from 17 to 41 percent based on the total insolation in the collector plane. Since the cost of these collectors vary, the choice should be based on the particular application. It was shown, however, that a choice does exist.

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11.* Smith, H. T., "Solar Energy System Performance Evaluation, Irvine School," SOLAR/2021-79/14 (September 1978 through March 1979) Department of Energy, Washington.

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18.* "Solar Project Cost Report, Reedy Creek Utilities," SOLAR/2018-78/60, Department of Energy, Washington.

* Copies of these reports may be obtained from Technical Information Center, P. O. Box 62, Oak Ridge, Tennessee 37830.

TABLE A-1. MONTHLY PERFORMANCE DATA.

DALLAS RECREATION		JUNE	JULY	AUGUST
1. $\bar{I}_T$	(kJ/day·m²)	21151.6	20569.9	21408.8
2. $\bar{I}_T$ during collector operations	(kJ/day·m²)	17331.3	17054.7	17754.9
3. $\bar{Q}_u^A$ (Area = 339.09 m²)	(kJ/day)	2.02x10⁶	1.99x10⁶	2.25x10⁶
4. Collector operating efficiency	(3/2·A_c)	28.2%	28.5%	31.1%
5. Collector daily efficiency	(3/1·A_c)	N/A	N/A	N/A
6. $\bar{Q}_L^{Ext}$ [piping]	(kJ/day)	N/A	N/A	N/A
7. $\bar{Q}_L^{Ext}$ [storage+ground, etc.]	(kJ/day)	N/A	N/A	N/A
8. $\bar{Q}_L^{Int}$ [piping]	(kJ/day)	N/A	N/A	N/A
9. $\bar{Q}_L^{Int}$ [storage-building]	(kJ/day)	0.03x10⁶	0.03x10⁶	0.06x10⁶
10. $\bar{Q}_{Load}$ [solar]	(kJ/day)	1.70x10⁶	1.55x10⁶	2.07x10⁶
11. $\bar{Q}_{Cooling}$ [solar]	(kJ/day)	0.40x10⁶	0.12x10⁶	0.57x10⁶
12. $(COP)_c$ [solar]		0.335	0.343	0.42
13. $\bar{Q}'_{Load}$ [auxiliary] CAE	(kJ/day)	0.87x10⁶	1.58x10⁶	0.93x10⁶
14. $\bar{Q}'_{Cooling}$ [auxiliary]	(kJ/day)	2.17x10⁶	3.95x10⁶	2.32x10⁶
15. $(COP)_c$ [auxiliary]		2.5	2.5	2.5
16. f - Fraction of cooling load by solar		16%	3%	19.85%
17. Electrical power usage				
a. Collector/exchanger pumps	(kJ(elec)/day)	86490	84270	88430
b. Circulating pump [solar]	(kJ(elec)/day)	298282	446369	345079
c. Circulating pump [auxiliary]	(kJ(elec)/day)	0	0	0
d. Cooling subsystem [solar]	(kJ(elec)/day)	0	0	0
e. Cooling subsystem [auxiliary]	(kJ(elec)/day)	867440	1581000	930239
*f. Load pump/blower [solar]	(kJ(elec)/day)	395250	538170	452160
*g. Load pump/blower [auxiliary]	(kJ(elec)/day)			
h. Total solar system power	(kJ(elec)/day)	N/A	N/A	N/A
18. $\bar{Q}_{Heating}$ [solar]	(kJ/day)	0	0	0
19. $\bar{Q}_{Hot\ Water}$ [solar]	(kJ/day)	-214	-1833	-627

* Same Fan Used.

TABLE A-1. MONTHLY PERFORMANCE DATA. (CONTINUED)

REEDY CREEK		JUNE	JULY	AUGUST
1. $\bar{I}_T$	(kJ/day·m²)	20733.4	20004.4	16510.6
2. $\bar{I}_T$ during collector operations	(kJ/day·m²)	12053.4	13491.3	11148.1
3. $\bar{Q}_u^A$ (Area = 356.7 m²)	(kJ/day)	74.4254	774327	933334
4. Collector operating efficiency	(3/2·A_c)	0.173	0.160	0.235
5. Collector daily efficiency	(3/1·A_c)	0.101	0.109	0.158
6. $\bar{Q}_L^{Ext}$ [piping]	(kJ/day)	N/A	N/A	N/A
7. $\bar{Q}_L^{Ext}$ [storage+ground, etc.]	(kJ/day)	252145	315445	433605
8. $\bar{Q}_L^{Int}$ [piping]	(kJ/day)	N/A	N/A	N/A
9. $\bar{Q}_L^{Int}$ [storage-building]	(kJ/day)	0	0	0
10. $\bar{Q}_{Load}$ [solar]	(kJ/day)	472333	439871	499516
11. $\bar{Q}_{Cooling}$ [solar]	(kJ/day)	283972	307163	479559
12. $(COP)_c$ [solar]		0.548	0.661	0.550
13. $\bar{Q}'_{Load}$ [auxiliary] CAE	(kJ/day)	150733	434226	492871
14. $\bar{Q}'_{Cooling}$ [auxiliary]	(kJ/day)	376428	1095031	1233151
15. $(COP)_c$ [auxiliary]		N/A	N/A	N/A
16. f - Fraction of cooling load by solar		0.43	0.22	0.28
17. Electrical power usage				
a. Collector/exchanger pumps	(kJ(elec)/day)	41767	50129	43258
b. Circulating pump [solar]	(kJ(elec)/day)	9395	8312	15161
c. Circulating pump [auxiliary]	(kJ(elec)/day)			
d. Cooling subsystem [solar]	(kJ(elec)/day)	54985	68114	54523
e. Cooling subsystem [auxiliary]	(kJ(elec)/day)			
f. Load pump/blower [solar]	(kJ(elec)/day)	71574	106557	130530
g. Load pump/blower [auxiliary]	(kJ(elec)/day)			
h. Total solar system power	(kJ(elec)/day)			
18. $\bar{Q}_{Heating}$ [solar]	(kJ/day)	0	0	0
19. $\bar{Q}_{DHW}$ [solar]	(kJ/day)	54231	49281	69032

IRVINE SCHOOL		JUNE	JULY	AUGUST
1. $\bar{I}_T$	(kJ/day·m²)	22146	24172	20765
2. $\bar{I}_T$ during collector operations	(kJ/day·m²)	20742	23969	20749
3. $\bar{Q}_u^A$ (Area = 460.1 m²)	(MJ/day)	3.85	4.36	3.68
4. Collector operating efficiency	(3/2·A_c)	40%	40%	39%
5. Collector daily efficiency	(3/1·A_c)	38%	39%	39%
6. $\bar{Q}_L^{Ext}$ [piping]	(kJ/day)	122380	66113	102335
7. $\bar{Q}_L^{Ext}$ [storage+ground, etc.]	(kJ/day)	N/A	N/A	N/A
8. $\bar{Q}_L^{Int}$ [piping]	(kJ/day)	244760	132227	204670
9. $\bar{Q}_L^{Int}$ [storage-building]	(kJ/day)	N/A	N/A	N/A
10. $\bar{Q}_{Load}$ [solar]	(kJ/day)	2710295	3149175	2871710
11. $\bar{Q}_{Cooling}$ [solar]	(kJ/day)	1069200	1184617	1155094
12. $(COP)_c$ [solar]		0.315	0.327	0.376
13. $\bar{Q}'_{Load}$ [auxiliary]	(kJ/day)	14071390	13078835	12059705
14. $\bar{Q}'_{Cooling}$ [auxiliary]	(kJ/day)	4022230	3965893	4096396
15. $(COP)_c$ [auxiliary]		0.315	0.327	0.376
16. f - Fraction of cooling load by solar		0.21	0.23	0.22
17. Electrical power usage				
a. Collector/exchanger pumps	(kJ(elec)/day)	379800	363975	372415
b. Circulating pump [solar]	(kJ(elec)/day)	1073875	1003527	1006818
c. Circulating pump [auxiliary]	(kJ(elec)/day)			
d. Cooling subsystem [solar]	(kJ(elec)/day)	1545292	1444049	1448786
e. Cooling subsystem [auxiliary]	(kJ(elec)/day)			
f. Load pump/blower [solar]	(kJ(elec)/day)	134373	125569	125981
g. Load pump/blower [auxiliary]	(kJ(elec)/day)			
h. Total solar system power	(kJ(elec)/day)			
18. $\bar{Q}_{Heating}$ [solar]	(kJ/day)	28485	0	12525
19. $\bar{Q}_{DHW}$ [solar]	(kJ/day)	N/A	N/A	N/A

PAGE JACKSON SCHOOL		JUNE	JULY	AUGUST
1. $\bar{I}_T$	(kJ/day·m²)	18459	15483	16572
2. $\bar{I}_T$ during collector operations	(kJ/day·m²)	14688	12052	13722
3. $\bar{Q}_u^A$ (Area = 1022.3 m²)	(kJ/day)	2866435	1936980	2428610
4. Collector operating efficiency	(3/2·A_c)	0.191	0.157	0.173
5. Collector daily efficiency	(3/1·A_c)	0.152	0.122	0.143
6. $\bar{Q}_L^{Ext}$ [piping]	(kJ/day)	10550	5627	2011
7. $\bar{Q}_L^{Ext}$ [storage+ground, etc.]	(kJ/day)	550007	768040	600300
8. $\bar{Q}_L^{Int}$ [piping]	(kJ/day)	5275	2813	1055
9. $\bar{Q}_L^{Int}$ [storage-building]	(kJ/day)	275003	384020	304192
10. $\bar{Q}_{Load}$ [solar]	(kJ/day)	1967900	760645	1503258
11. $\bar{Q}_{Cooling}$ [solar]	(kJ/day)	974941	268354	544361
12. $(COP)_c$ [solar]		0.494	0.359	0.364
13. $\bar{Q}'_{Load}$ [auxiliary]	(kJ/day)	231233	1548	68548
14. $\bar{Q}'_{Cooling}$ [auxiliary]	(kJ/day)	766025	2711	219704
15. $(COP)_c$ [auxiliary]		3.326	1.541	4.653
16. f - Fraction of cooling load by solar		0.56	0.99	0.63
17. Electrical power usage				
a. Collector/exchanger pumps	(kJ(elec)/day)	88200	79355	89548
b. Circulating pump [solar]	(kJ(elec)/day)	59670	14601	31819
c. Circulating pump [auxiliary]	(kJ(elec)/day)			
d. Cooling subsystem [solar]	(kJ(elec)/day)	59670	14601	31819
e. Cooling subsystem [auxiliary]	(kJ(elec)/day)			
f. Load pump/blower [solar]	(kJ(elec)/day)	626543	153313	334097
g. Load pump/blower [auxiliary]	(kJ(elec)/day)			
h. Total solar system power	(kJ(elec)/day)			
18. $\bar{Q}_{Heating}$ [solar]	(kJ/day)	0	0	0
19. $\bar{Q}_{DHW}$ [solar]	(kJ/day)	0	0	0

TABLE A-2. COMPOSITE PERFORMANCE FACTORS.

SITE NAME / PERFORMANCE FACTOR		JUNE	JULY	AUGUST
Page Jackson School				
I_c	(kJ/hr-m^2)	951	1185	1806
ξ		2.50	2.50	2.50
$(COP)_c$		.494	.359	.364
β		3.02	3.79	3.75
$\overrightarrow{Q_u}$	(kJ/day)	1459438	-302784	673540
η_{st}		3.8%	-0.69	1.4%
SCOP = SCOP'		1.75	-1.16	1.38
$\overline{Q}_c$	(kJ/day)	-625769	-957459	-544668
η_s		-3.3%	-6.1	-3.2%
Reedy Creek Utilities				
I_c	(kJ/hr-m^2)	1294	1236	661
ξ		2.50	2.50	2.50
$(COP)_c$		0.548	0.661	0.550
β		2.82	2.51	2.82
$\overrightarrow{Q_u}$	(kJ/day)	492109	458882	499719
η_{st}		3.6%	4.2%	4.7%
SCOP = SCOP'		2.77	1.97	2.05
$\overline{Q}_c$	(kJ/day)	47807	-123898	-108876
η_s		0.6%	-1.7%	-1.8%

TABLE A-2. COMPOSITE PERFORMANCE FACTORS.(CONTINUED)

SITE NAME / PERFORMANCE FACTOR		JUNE	JULY	AUGUST
Dallas Recreation Center				
I_c	(kJ/hr-m^2)	964	1133	1017
ξ		2.50	2.50	2.50
$(COP)_c$		0.335	0.343	0.420
β		3.98	3.92	3.38
$\overrightarrow{Q_u}$	(kJ/day)	1900600	1872400	2047200
η_{st}		10.5%	9.2%	11.8%
SCOP = SCOP'		0.82	0.60	0.97
$\overline{Q}_c$	(kJ/day)	-16567	-274271	-70128
η_s		-0.2%	-3.9%	-1.0%
Irvine School[1]				
I_c	(kJ/hr-m^2)	230	461	230
ξ		2.50	2.50	2.50
$(COP)_c$		0.315	0.327	0.376
β		4.17	4.06	3.66
$\overrightarrow{Q_u}$	(kJ/day)	2706971	3757045	2828573
η_{st}		8.4%	11.0%	11.1%
SCOP = SCOP'		0.86	1.28	0.96
$\overline{Q}_c$	(kJ/day)	-5126379	-3585754	-1713216
η_s		-0.50	-0.32	-0.18

[1] No thermal storage - assume $T_{f,1}$ = 185°F.

NOTES

SOLAR HEATING AND COOLING RESULTS FOR THE LOS ALAMOS STUDY CENTER*

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ABSTRACT

Solar energy system performance data for two heating seasons and two cooling seasons are given for the Los Alamos National Security and Resources Study Center, a 5576 m^2 (60,000 ft^2) conference center and library at the Los Alamos Scientific Laboratory, New Mexico.

The solar energy system for the Study Center consists of a 716 m^2 (7705 ft^2) array of selectively coated, single-glazed collectors, a 19 m^3 (5000 gal) pressurized tank for hot storage in the cooling mode, and a 38 m^3 (10,000 gal) tank, which is used for hot storage in the heating mode and cold storage in the cooling mode. Either of two chillers may be used in series with the cold storage tank, a 1075 MJ (85 ton) absorption unit, or a 973 MJ (77 ton) Rankine cycle unit. Night evaporative cooling is also used to cool the large tank. A heat recovery unit is used to preheat fresh air in the winter, and, by means of spraying the exhaust air, to pre-cool fresh air in the summer.

Daily, monthly, seasonal, and yearly energy summaries are presented for the system. Performance data for the two chillers inlcude tabulation of thermal and system coefficients of performance.

The solar fraction for the system was 69% for the 1978-79 heating season, which was down from 79% the previous winter. This was due to a more severe winter as the solar performance was about the same. The solar cooling fraction was 97% in 1978 versus 94% in 1979.

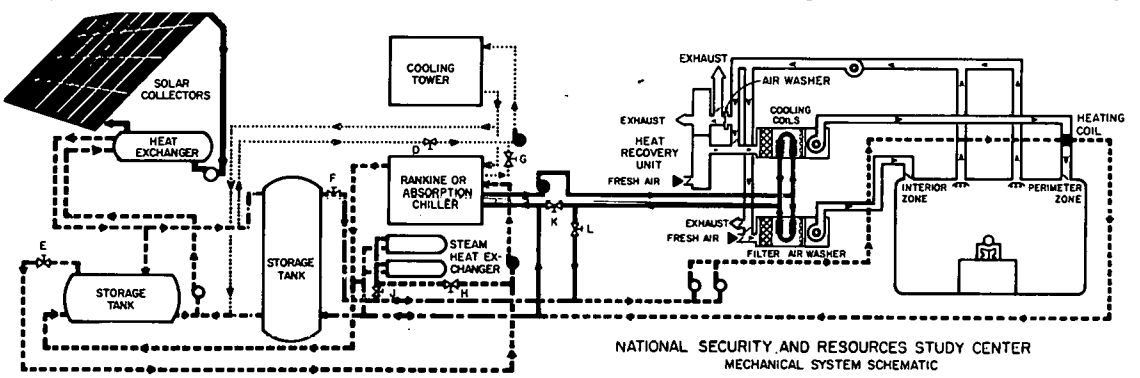
SYSTEM DESCRIPTION

A schematic of the energy, heating, ventilating, and air conditioning system is shown in Fig. 1. The 716 m^2 (7705 ft^2) collector array is architectually and structurally integrated into the building, forming the roof of the mechanical equipment room. The total air-conditioned floor area of the building is 5576 m^2 (60 000 ft^2) on three floors.

HVAC System

The HVAC system is a two-zone (perimeter and interior) variable air volume system with separate supply and return fans and cooling coils for each zone. Hot water coils for heating are provided in the perimeter zone only.

The air handling system features recirculation of inside air, and a heat pipe heat recovery unit in the perimeter zone system serves two functions: to preheat outside air in the heating mode; and, by means of spraying the exhaust air, to precool outside air in the cooling mode. The light fixtures are cooled by the



1. Schematic of Study Center System.

*This work performed under the auspices of U.S. Department of Energy, Office of Solar Applications.

return air. The main air supply units each have a cooling coil, an air washer, and a supply fan.

Energy System

The energy system consists of the solar collector array and heat exchanger, two storage tanks, and two water chillers--a lithium-bromide absorption chiller, and a Rankine cycle unit, either of which can be used for comparative studies of solar air-conditioning. The collectors are fabricated of mild steel, have a selective surface of black chrome, and are single-glazed with water-white glass. Four hundred and seven collectors are connected in parallel to form the array. The collectors are cooled with a light paraffinic oil heat transfer fluid.

In the heating mode, a 38 m^2 (10,000 gallon) tank is used to store hot water for heating. In the cooling mode, a 19 m^2 (5000 gal) pressurized tank is used to store solar hot water to operate either of the chillers, and cold water is stored in the larger tank. Cold water is generated either by one of the chillers, which are in series with the larger tank, or by night evaporative cooling with the cooling tower, which is normally used to reject heat from the chillers.

The solar energy system is backed up by auxiliary steam heat exchangers that generate hot water directly for heating or to power the chillers. Domestic hot water is generated by means of a 0.38 m^3 (100 gal) tank connected to the solar hot water tank, and augmented by an electrically heated tank downstream.

Heating Controls

The building control system demands hot water for heating at a temperature of 60°C (140°F) if the ambient temperature is -18°C (0°F), and 21°C (70°F) if the ambient is 21°C (70°F), with a linear profile as a function of ambient temperature. If the stored solar hot water drops below the required temperature, the auxiliary system is used for heating.

Cooling Controls

The cooling control system regulates the fan discharge temperature to satisfy the building cooling load. The building cooling sequence is activated in the following order:

1. Full Fresh Air
2. Air Washers Start
3. Exhaust Spray
4. Chilled Water from Cold Storage
5. Water Chiller Starts

The chiller is started when the hot storage tank has been heated above 82°C (180°F), regardless of whether the building is calling for cooling. The chiller continues to operate until the hot tank drops 77°C (170°F) or until the cold tank drops below 9°C (48°F). Auxiliary operation is prohibited if the cold storage tank is below 18°C (65° F). This scheme improves system performance by running the chiller when the collectors are operating, thereby reducing collector temperatures; forcing long chiller operating times at full capacity; prohibiting auxiliary operating until cold storage is exhausted.

Solar Collector Controls

A temperature probe on the back of the collector is used to compare collector temperature with the hot storage tamperature. When a 5.5°C (10°F) temperature differential is established, the collector pumps are started after a three minute delay.

SYSTEM THERMAL PERFORMANCE

About 160 channels of instrumentation are installed in the building for the purpose of evaluating the system performance. These measurements consist primarily of flow rates and temperature differences on all of the energy sybsystems. In addition, electrical power consumptions of all of the pumps and fans are measured. Most temperatures are measured using platinum resistance thermometers. Flows are measured with turbine flowmeters and Annubar probes. Solar flux is measured in the plane of the collector and in the horizontal plane. Data acquisition is controlled by a PDP-11/34 computer, which automatically provides periodic energy summaries for display and storage.

Temperature probes were calibrated in the laboratory and as installed, and matched for temperature difference measurements. The resolution and accuracy of the resistance probes are 0.3°C (0.5°F) and 0.1°C (0.2°f), respectively.

Solar System Daily Performance

Figure 2 gives the collector incident and output for the 1978-79 heating and cooling season.

System Load

Figure 3 shows the portions of the building heating load provided by solar and auxiliary for the 1978-79 heating season. Figure 4 summarizes the chiller hot water consumption and chilled water output for the 1979 cooling season.

Monthly Energy Summary

The mean daily values of system energies are given in Table I for the 22 month period. The Nov 77 to Sept 78 period reported in (6) has been repeated to allow a seasonal comparison. Generally, the data reported here are complete. Occasionally, one day or part of a day was lost for any given month. In these cases, estimates were made of the daily energy values to complete monthly records of daily summaries. The months with missing data are Nov 77 when the data

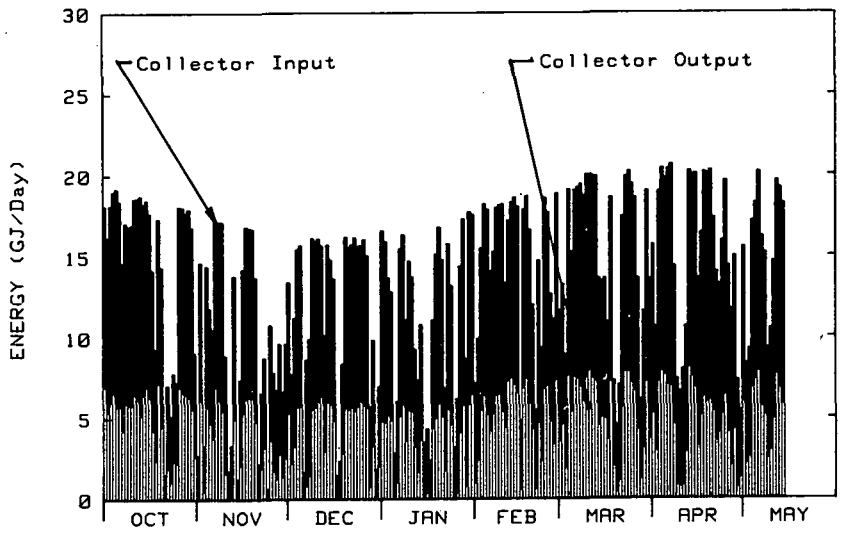


Fig. 2. Daily collector input and output for the 1978-79 heating season.

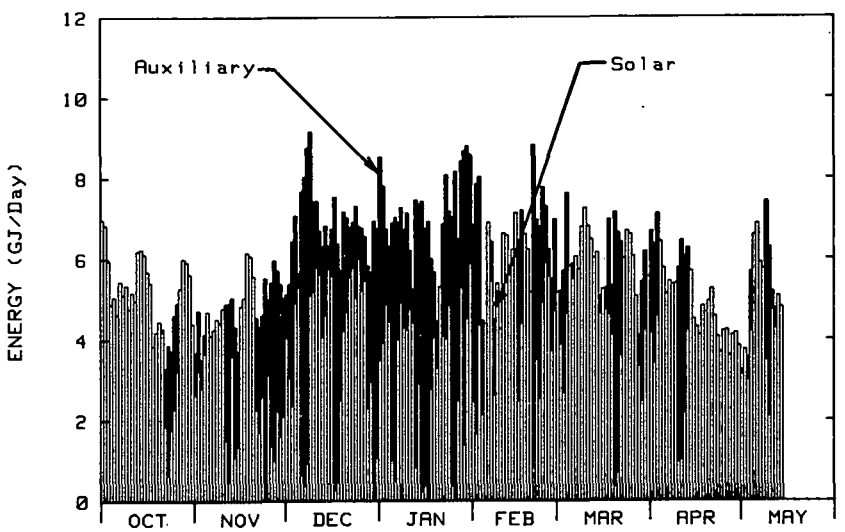


Fig. 3. Daily solar energy delivered and auxiliary energy used during 1978-79 heating season.

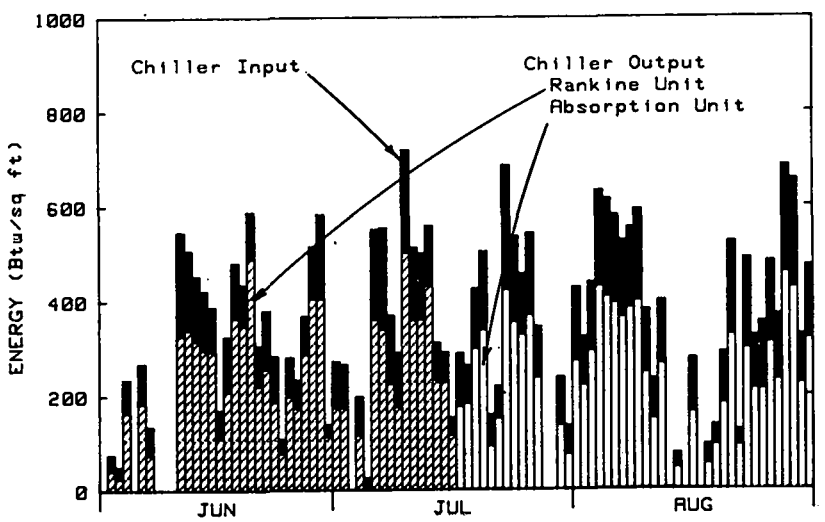


Fig. 4. Daily chiller hot water input and chilled water output for the 1979 cooling season.

TABLE I

MONTHLY SYSTEM THERMAL PERFORMANCE
SUMMARY (MJ/DAY)

Month	Year	Total Incident Solar	Collector Output	Storage to H-C Load	Storage to DHW	Cooling Output	Storage and Piping Losses	Total Operating Electricity	Auxiliary	Percent Solar	Days of Data
Nov	77	13985	4902	3766	0	0	1136	282	580	86.9	28.0
Dec	77	12176	4070	3493	74	0	503	272	1287	73.5	31.0
Jan	78	11439	3780	3138	108	0	534	247	1936	62.6	31.0
Feb	78	13027	4079	3446	108	0	526	270	1550	69.6	28.0
Mar	78	14618	4839	4021	107	0	711	295	632	86.7	31.0
Apr	78	17583	4954	4325	96	0	533	305	27	99.4	30.0
May	78	15537	2990	2298	127	1635	565	445	0	100.0	31.0
Jun	78	15843	3255	2584	106	1880	565	484	158	94.5	30.0
Jul	78	14851	3386	2758	94	2102	535	538	275	91.2	31.0
Aug	78	14798	3557	2927	78	2057	552	536	0	100.0	31.0
Sep	78	15442	2770	2024	85	1328	661	436	0	100.0	28.0
Oct	78	15047	5042	4614	0	0	428	304	343	93.1	31.0
Nov	78	9762	3196	2883	0	0	313	218	1726	62.5	30.0
Dec	78	11754	3941	3434	0	0	517	232	3101	52.5	31.0
Jan	79	11465	3566	3765	97	0	705	211	3891	42.4	31.0
Feb	79	14870	5104	4216	119	0	768	284	1806	70.6	28.0
Mar	79	15264	5312	4585	109	0	618	307	1075	81.4	31.0
Apr	79	15286	4702	3925	119	0	659	290	804	83.4	30.0
May	79	15117	5306	4236	96	0	973	326	795	84.5	14.0
Jun	79	15166	2864	2156	116	1595	593	460	85	96.4	30.0
Jul	79	14285	3143	2562	102	1822	479	503	158	94.4	31.0
Aug	79	14558	3358	2757	123	1965	478	506	232	92.6	31.0

system was first started, and Sept 78 and May 79 when the system was switched to the other mode of operating (heating or cooling). Very little cooling was accomplished in May 79 and the data have not been reduced. A plot of collector input and output is shown in Fig. 5 for the 22 month period. The solar and auxiliary energy provided to the building or chiller, along with the cooling accomplished by the chiller, is shown in Fig. 6. The plot shows the increased amount of auxiliary energy used in the heating season due to the more severe winter.

The monthly system performance parameters are given in Table II. Each of these parameters are defined by the following:

ξ = Factor for converting the electrical energy usage to its thermal energy equivalent, dimensionless.

β = Factor to account for the degree of non-usefulness of heat losses to the interior of the conditioned space, dimensionless.

$\overline{Q_u}^*$ = Average daily solar heat delivered to load over a specified period (typically one month).

$\overline{Q}_{HC}$ = Average daily savings in non-renewable energy resources over a specified period (typically one month) by the solar heating or cooling system.

SCOP' = Solar coefficient of performance, dimensionless.

$$= \frac{\text{Total useful heating and/or cooling by solar (daily average).}}{\text{Total electrical power usage to operate the solar system (daily average).}}$$

SCOP = Solar controlled coefficient of performance, dimensionless.

$$= \frac{\text{Total, controlled, useful heating and/or cooling by solar (daily average).}}{\text{Total electrical power usage to operate the solar system (daily average).}}$$

η_C = Average daily collector efficiency over a specified period (typically one month), dimensionless.

η_S = Average daily solar heating and cooling system efficiency over a specified period (typically one month), dimensionless.

η_{ST} = Average daily solar heating and cooling system thermal efficiency over a specified period (typically one month), dimensionless.

The Study Center is unique because the equipment room that encloses the entire solar energy system is heated but not cooled. Hence the system heat loss is useful in the winter but is not detrimental in the summer. The losses are interior losses in the heating season and exterior losses in the cooling season. The interior losses were considered useful whenever the mean daily ambient temperature was below 10°C (50°F). The term β (the fraction of losses that are not useful) varies from 0.0 in winter to 1.0 in summer. The amount of useful daily energy collected ($\overline{Q_u}^*$) varies from a monthly average high of 5312 MJ/Day (5.04 million Btu) in heating to a low of 2109 MJ/Day (2.0 million Btu) in cooling. The amount of non-renewable energy sources saved ($\overline{Q}_{HC}$) varies from 4543 MJ/Day (4.3 million Btu) to 669 MJ/Day (0.6 million Btu) due to large amounts of electrical energy required to run the system in the cooling mode.

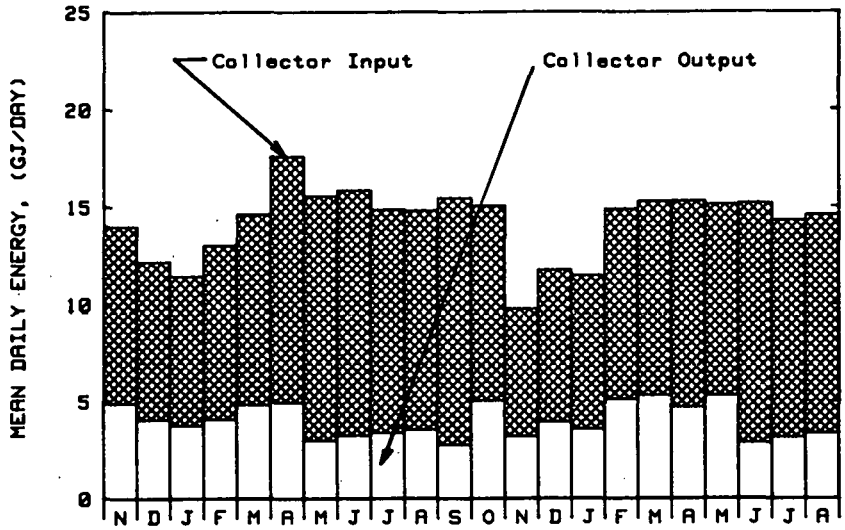


Fig. 5. Monthly average collector input and output from November, 1977, through August, 1979.

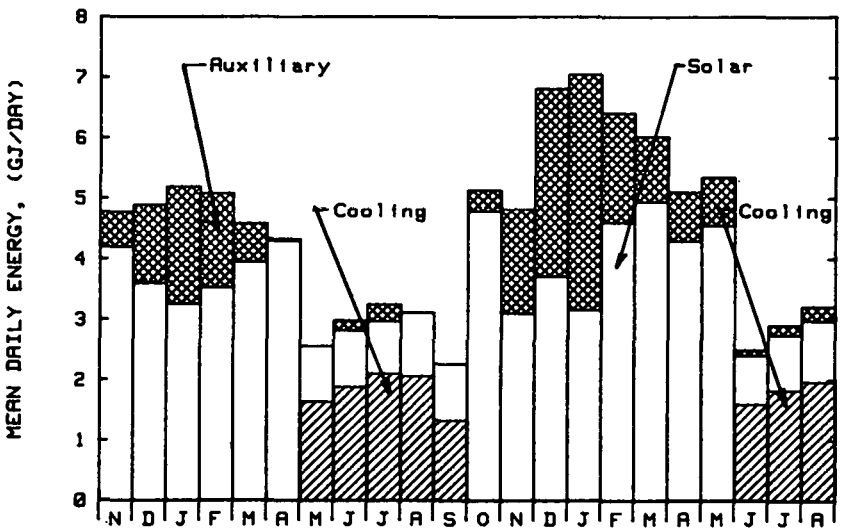


Fig. 6. Monthly average solar delivered auxiliary used and cooling delivered from November, 1977, through August, 1979

Seasonal System Performance

Table III gives a seasonal system performance summary for two heating seasons and two cooling seasons. The daily average values in comparing seasons are almost identical except for the larger amount of auxiliary energy needed this past 78-79 heating season. Seasonal Performance Parameters are given in Table IV. Again, the comparison of seasons is almost identical.

YEARLY PERFORMANCE

The total amount of solar energy incident on the collector, the collector output, the useful energy utilized, and the non-renewable energy resources saved are given in Table V for 77-78 and 78-79. Since at this time only 22 months of [illegible] are available, September and October of 78 [illegible] common to both. This table is intended to show the total amount of energy produced over a year.

Chiller Performance

The performance of the absorption unit and the Rankine unit has been determined separately. The Rankine unit was operated from June 1 to July 14, 1979, and the absorption unit from July 15 to August 31, 1979. The results for these two periods are given in Table VI. These values include the time when the system was operating on auxiliary hot water. The electrical energy consumption used by each component in the cooling system is also listed in Table VI. Both chillers performed nearly the same with the performance slightly lower than last year (6).

The big energy consumers are the collector pump, the cooling tower pump and the cooling tower, which take about 67% of the total. By changing the collector fluid from paraffinic oil to glycol-water and by using a cooling tower with a propeller-type fan instead of squirrel-type fan and placing it outside at ground level, the power of these three items could be reduced at least

TABLE II

MONTHLY SYSTEM PERFORMANCE
PARAMETERS (ξ = 2.5)

Month	Year	β	COP	$\overline{Q}_U$* (MJ/DAY)	$\overline{Q}_{HC}$ (MJ/DAY)	SCOP'	SCOP	η_C %	η_{ST} %	η_S %
Nov	77	.14	1.00	4740	4036	16.83	13.38	35.1	33.9	28.9
Dec	77	0.00	1.00	4070	3391	14.99	13.13	33.4	33.4	27.8
Jan	78	0.00	1.00	3780	3162	15.30	13.14	33.0	33.0	27.6
Feb	78	0.00	1.00	4079	3405	15.13	13.18	31.3	31.3	26.1
Mar	78	.16	1.00	4724	3986	16.00	13.98	33.1	32.3	27.3
Apr	78	.47	1.00	4706	3942	15.41	14.48	28.2	26.8	22.4
May	78	1.00	.71	2425	933	3.87	3.87	19.2	11.1	6.0
Jun	78	1.00	.69	2689	1015	3.81	3.81	20.5	11.6	6.4
Jul	78	1.00	.69	2851	1043	3.67	3.67	22.8	13.3	7.0
Aug	78	1.00	.70	3005	1171	3.94	3.94	24.0	14.3	7.9
Sep	78	1.00	.66	2109	699	3.17	3.17	17.9	9.0	4.3
Oct	78	.61	1.00	4780	4020	15.72	15.17	33.5	31.8	26.7
Nov	78	.10	1.00	3165	2619	14.50	13.21	32.7	32.4	26.8
Dec	78	0.00	1.00	3941	3361	16.98	14.76	33.5	33.5	28.6
Jan	79	0.00	1.00	3566	3040	16.94	13.59	31.1	31.1	26.5
Feb	79	0.00	1.00	5104	4395	18.00	15.29	34.3	34.3	29.6
Mar	79	0.00	1.00	5312	4543	17.28	15.26	34.8	34.8	29.8
Apr	79	.60	1.00	4307	3581	14.84	13.93	30.8	28.2	23.4
May	79	.43	1.00	4889	4073	14.97	13.27	35.1	32.3	26.9
Jun	79	1.00	.71	2272	798	3.51	3.51	18.9	10.7	5.3
Jul	79	1.00	.67	2664	942	3.54	3.54	22.0	12.5	6.6
Aug	79	1.00	.66	2880	1062	3.74	3.74	23.1	13.0	7.3

TABLE III

SEASONAL SYSTEM
PERFORMANCE SUMMARY (MJ/DAY)

	Season	Incident Solar	Collector Output	Storage to H-C Load	Storage to DHW	Cooling Output	Storage and Piping Losses	Total Operating Electricity	Auxiliary	Percent Solar	Days of Data
Heating	77-78	13793	4433	3698	83	0	652	278	1004	79.0	179
	78-79	13446	4459	3799	65	0	398	267	1762	68.7	236
Cooling	78	15288	3200	2528	98	1809	574	489	88	96.8	151
	79	14665	3125	2495	115	1796	516	490	159	94.3	92

TABLE IV

SEASONAL SYSTEM
PERFORMANCE PARAMETERS

Season	Year	β	COP	$\overline{Q}_U$* (MJ/DAY)	$\overline{Q}_{HC}$ (MJ/DAY)	SCOP'	SCOP	η_C %	η_{ST} %	η_S %
Heating	77-78	.13	1.0	4346	3650	15.6	13.6	32.1	31.5	26.5
	78-79	.20	1.0	4341	3672	16.2	14.5	33.2	32.3	27.3
Cooling	78	-	.69	2625	972	3.72	3.72	20.9	11.9	6.4
	79	-	.68	2609	935	3.60	3.60	21.3	12.0	6.4

50%. The system COP would then be increased to over 5.

TABLE V

YEARLY PERFORMANCE

	77-78 361 days GJ (10^6Btu)	78-79 346 days GJ (10^6Btu)
QINC	5244 (4975)	4820 (4573)
QCOUT	1433 (1360)	1373 (1303)
QUSE	1322 (1254)	1280 (1214)
QSAVED	925 (878)	935 (887)

TABLE VI

CHILLER PERFORMANCE AND PARASITIC POWER REQUIREMENTS

1979 COOLING SEASON

		Absorption Unit	Rankine Unit
1.	Hot Water Input	132.6 GJ	112.5 GJ
2.	Chilled Water Output	86.6	78.6
3.	Collector Pump	7.51	7.56
4.	Heat Exchanger Pump	0.67	0.69
5.	Chiller Hot Water Pump	3.27	3.09
6.	Chilled Water Pump	1.29	1.15
7.	Cooling Tower Pump	3.30	3.34
8	Cooling Tower Fan	5.04	4.55
9.	Chiller Power	2.73	2.37
10.	Total Power	23.82	22.76
	COP	0.658	0.699
	SCOP	3.635	3.455
	SCOP - Modified (Reduce Item 3,7,8 by 50%)	5.35	5.23

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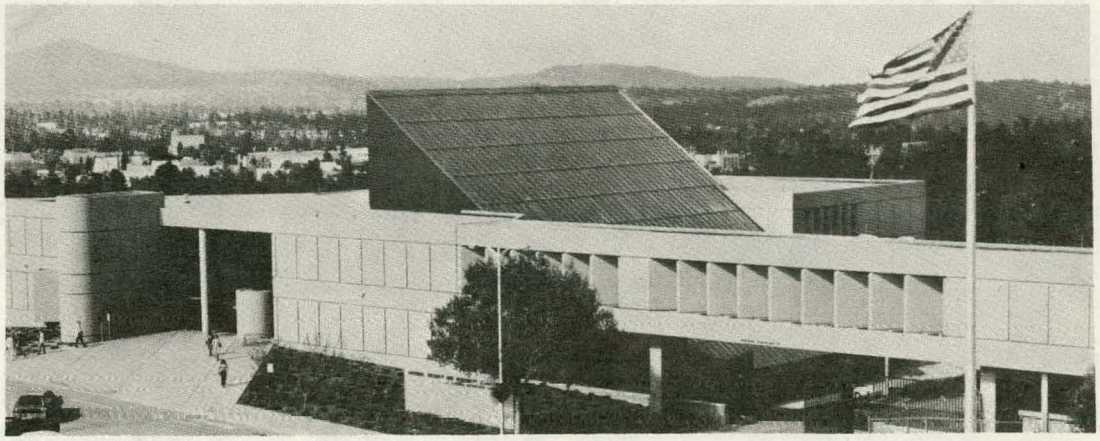
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NOTES

IMPROVING PERFORMANCE OF SOLAR COOLING SYSTEMS

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ABSTRACT

During the summer of 1978, Colorado State University Solar House III utilized a Yazaki 2-ton lithium bromide absorption chiller, two 500 gallon cool storage tanks, one 1200 gallon hot storage tank (located within the conditioned space), 631 square feet of Chamberlain single cover selective surface flat-plate solar collectors, and an electric auxiliary boiler. After a year of heating and cooling data were obtained, the system was redesigned in the spring of 1979. Major modifications included reduced piping runs, reduced electrical power requirements, elimination of cool storage, improved insulation, and a relocation of the hot storage to the exterior of the conditioned space. The performance results of these two distinct installations are shown in Table 1, columns 3 and 5. Also included are design predictions and selected data presentations which provide useful analysis of the variation in system performance. It is noteworthy that the solar collector array installation is the same for both system designs.

Table 1 also lists the resulting values of those component and system parameters which are necessary for a realistic evaluation of any solar heating and cooling system.

The paper includes an analysis of the two consecutive years of data acquisition on the two systems for selected months in the cooling season.

INTRODUCTION

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. 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 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 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 pumped to a cooling coil for space cooling. Auxiliary heating and cooling is supplied by an air-to-air heat pump. An automatic control system provides for all functions of the system operation.

The house 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 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, system design variables, etc. can also be investigated.

SYSTEM MODIFICATIONS

In the spring of 1979, the solar heating and cooling system of Solar House III was redesigned and extensively modified. The purpose of the modification was to increase the overall performance of the system. Data analyzed for the 1978 cooling season provided ample evidence that, for solar cooling to be competitive with conventional methods, significant system changes would have to be implemented. Based on this analysis and other experience gained in operating an absorption cooling system, the following recommendations were made:

(1) Reduce parasitic power consumption
(2) Reduce length of piping runs and piping heat losses
(3) Decrease the cooling load from uncontrolled heat losses by relocating the thermal storage tank exterior to the conditioned space
(4) Install an air-to-air heat pump as an auxiliary source of heating and cooling (replacing an electric boiler)
(5) Reduce the complexity of the system design and controls

The first two objectives were accomplished by optimizing the placement of the system components, thereby minimizing the length of interconnecting piping. Table 2 shows that the total length of piping was reduced by over 400 feet. In addition, the number of valves, elbows, and other flow restrictions were reduced. The combination of these

improvements with the use of higher efficiency pumps allowed the total pumping horsepower to be reduced from 2.31 to 0.74.

The analysis of cooling data from 1978 showed that approximately 25% of the total cooling load was the result of uncontrolled heat losses from thermal storage. To prevent this from recurring the thermal storage tank was moved to an attached tank room and insulated with four inches of polyurethane foam. While the use of an attached tank room does not completely eliminate the negative effects of internal heat loss during the cooling season (because of conduction and infiltration from the tank room), it does allow these losses to be utilized during the winter through controlled venting of heated air. The result of these modifications on the location and amount of heat loss can be seen in Table 1, lines 6 through 9b. It should be noted that the heat losses have a more pronounced effect during the cooling season since twice the amount of heat lost to the conditioned space must be supplied to the chiller to restore the space to its original condition (assuming $COP_{chiller} = 0.5$).

The installation of the air-to-air heat pump was made primarily to replace the electrical boiler that had previously been used to supply auxiliary heat to the absorption chiller generator. Since the heat pump has an approximate COP of 2, as compared to 0.55 for the chiller, some savings in electrical consumption do accrue through its use. This does not imply that the heat pump is necessarily the best auxiliary source for solar heating and cooling systems. The additional cost of a heat pump as compared to other auxiliary forms offsets some of its advantages. Further comparisons and studies will have to be made before final determination of the most attractive auxiliary source for a particular installation.

A significant fact is that the principal system components (such as collectors, chiller, storage tank, etc.) remain the same for the two-year period. Therefore this study was directed primarily toward improving system performance through design improvements.

EFFECTS OF MODIFICATION ON SYSTEM PERFORMANCE

A comprehensive analysis of the effects of system changes on performance would require substantially more data than were collected during the summer of 1979. Further, the atypical weather experienced during this period in Fort Collins compounded this problem. This is evidenced by the fact that, during 11 days of August 1979, no cooling was required by the house. Despite these limitations, a performance comparison for August 1978 and 1979 yields some interesting results.

The results discussed are summarized in Table 1. This table lists a number of system performance parameter and their value for August 1978 and 1979. In addition, it includes columns for an "ideal" five day cooling period in August 1979 and an estimation of the 1979 performance without DHW (note: the DHW system was not in operation during August 1978). Finally, figures are shown for the performance predictions attained from methods described in the next section.

The greatest improvement was found to be in the reduction of electrical power usage, which declined from 59.0 MJ/day to 22.5 MJ/day. This 61% reduction, coupled with the increase in useful heat delivered to load (Q_u^A) of 50 MJ/day and a reduction in thermal losses, caused the solar coefficient of performance (SCOP) to increase from 1.59 to 6.1.*

The full effect of reducing electrical power consumption cannot be appreciated until the electrical savings are converted to their equivalent thermal energy. For this analysis, the conversion of one kilojoule electrical power is approximately 2.4 KJ thermal power. Applying this conversion shows that piping modifications account for an average savings of 87 KJ/day of thermal energy over the previous system. This result shows the need for great care in piping design and pump selection.

The thermal losses from storage and piping, as shown in Table 1, were somewhat reduced from 1978 yet were still significantly greater than predicted. An analysis of these losses shows that approximately 25% of the tank losses occur by conduction through the tank legs. Additionally, the possibility of an optimistic R value for the tank insulation and unexpected thermosyphoning in the heat exchanger and domestic hot water loops also contribute to the loss.

The system thermal efficiency, defined as

$$\eta_{ST} = \left(\frac{Q_u^A - Q_L^{Ext} - Q_L^{Int}}{A_c I_T}\right) COP_{Chiller}$$

increased from 8.7% to 13.1%. This increase can be primarily attributed to the thermal of the thermal storage from the conditioned space. The overall system efficiency increased from 0.4% to 10.2% and this increased degree of improvement is due to the threefold increase in the system SCOP.

Another important parameter to consider is Q_c, the actual savings of non-renewable energy. It is to be noted that, by optimially designing the system, the "real" energy savings (electrical and thermal) were increased from a low of 4.4 MJ/day in August 1978 to 108 MJ/day in 1979.

COMPARISON OF ACTUAL TO PREDICTED PERFORMANCE

During the summer of 1979 an attempt was made to model the performance of the two systems previously discussed (see Figure 1 for system schematic). Two methods were used to arrive at these predictions. The first, using the $\bar{\phi}$-f-chart formulas by Klein and Beckman [2], was done for both system utilizing data inputs from August 1978 and 1979. These

* $$SCOP = \left(\frac{Q_u^A - Q_{Loss}}{\text{Electrical use}}\right) COP_{Chiller}$$

results are shown in column 2 of Table 1. Next, a detailed analysis (1979 only) was made which optimized the predictions to the specific system by considering additional inputs such as heat losses, their location, and parasitic power use.

An examination of Table 1 shows a fair degree of agreement between actual performance and the $\bar{\phi}$-f-chart prediction in the collection related categories which is, of course, the essential function of $\bar{\phi}$-f-chart. However, the closeness of the agreement for the 1979 season is suspect because of DHW use, which was not accounted for in the $\bar{\phi}$-f-chart calculation. The discrepancy between $\bar{\phi}$ and actual performance for 1978 does not have the DHW complication and is therefore a better indication of the accuracy of $\bar{\phi}$-f-chart. Here the error was an overprediction of 16%.

$\bar{\phi}$-f-chart does not consider system heat losses or electrical usage, however, and thus correlation between other parameter predictions and actual performance is limited. Comparison of valuves of Q_u shows a discrepancy of 211 MJ/day (130%) in 1978 and 145 MJ/day (70%) for 1979. This discrepancy is almost entirely due to lack of consideration of heat losses which can be devastating in a cooling system. Examination of the prediction which includes heat losses shows a reduction of the Q_u error to an acceptable level (in this case, 21%).

A factor which even more severely limits $\bar{\phi}$-f-chart's usefulness as a design tool is its inability to differentiate between systems whose thermal and electrical use characteristics, and thus performance, vastly differ. For example, $\bar{\phi}$-f-chart predicts that η_S for both August 1978 and 1979 will be approximately 20%. In actuality, the 1978 figure was 0.4% and increased to 10.2% in 1979. The $\bar{\phi}$-f-chart model not only predicts a system efficiency twice the maximum achieved, but fails to predict the significant difference in efficiencies for the two systems. Once again, the detailed model shows much better correlation. These results clearly show the need for consideration of additional system related variables in forming accurate performance modeling.

CONCLUSIONS

The design, operation, and performance analysis of two solar cooling systems has led to several conclusions and recommendations. They are as follows:

(1) First, a solar cooling system should be designed to minimize the parasitic power use. This can be accomplished through careful selection and placement of system components to reduce the length and complexity of piping connections. Additionally, appropriately sized, high efficiency pumps and/or blowers should be used.
(2) Careful consideration must be given to heat losses and their location. An externally located thermal storage tank and R-30 insulation on the tank is extremely desirable and perhaps essential for a successful system. Other less obvious means of heat loss, such as conduction through the tank support structure and thermosyphoning in load as well as collector loops, should be identified and eliminated if possible.
(3) Solar cooling feasibility depends on the specific design and installation variables, such as parasitic power consumption, thermal losses, domestic hot water use, control strategies, building configuration, and many other factors. Therefore an accurate estimation of the performance of a solar cooling system must necessarily include a careful calculation/consideration of these variables.

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ACKNOWLEDGEMENT

This work has been supported by the U.S. Department of Energy, Office of Solar Applications, Office of the Assistant Secretary of Conservation and Solar Energy.

TABLE I. CSU SOLAR HOUSE III PERFORMANCE DATA/PREDICTIONS

Parameter (column 1)	$\bar{\phi}$-f-chart Prediction 1978 (column 2)	$\bar{\phi}$-f-chart Prediction 1979 (column 2)	Experimental Performance Aug 1978 (column 3)	Detailed Model Prediction Aug 1979 (column 4)	Experimental Performance 2-26 Aug 79 (column 5)	Experimental Performance 2-7 Aug 79 (column 6)	Estimated Cooling Only (without DHW) Performance 2-26 Aug 79 (column 7)
1. $\bar{I}_T$ MJ/day·m²	18.4	18.0 (a)	18.4	18.0 (a)	18.0	25.2	18.0
2. $\bar{I}_T$ (during operations)	---		11.3	11.1 (c)	13.9	21.4	13.9
3. $\bar{Q}_u^A$ MJ/day	365.7	350.2 (c)	315.5	309.1 (c)	357.9	562.6	357.9
4. η_c (during operations)	---		47.6%	47.6%(a)	43.9%	44.9%	43.9%
5. η_c (daily)	33.9%	33.2% (c)	29.3%	29.3%(a)	33.9%	38.1%	33.9%
6. $\bar{Q}_L^{Ext}$ (piping) MJ/day	0	0	14.7	15.2 (t)	17.8	20.9	17.8
7. $\bar{Q}_L^{Ext}$ (storage)	0	0	19.1	13.9 (t)	35.9	31.1	35.9
8. $\bar{Q}_L^{Int}$ (piping)	0	0	9.8	7.4 (t)	12.7	19.4	12.7
9a. $\bar{Q}_L^{Int}$ (storage)	0	0	38.1	4.6 (t)	9.6	10.4	9.6
9b. $\bar{Q}_L^{Int}$ (storage + preheat)	0	0	38.1*	20.1 (t)	21.4	16.1	9.6
10a. $\bar{Q}_{Load}$ (total)	365.7	350.2	233.8*	268.0 (c)	300.2 †	451.2 ††	300.2
10b. $\bar{Q}_{Load}$ (chiller)	365.7	287.9	233.8	205.7 (c)	237.9	422.7	300.2
11. $\bar{Q}_{Cooling}$	219.4	172.7	141.5	123.4 (c)	125.4	239.6	158.2
12. $(COP)_c$	0.600 (a)		0.605	0.600 (a)	0.527	0.567	0.527
13. Total solar system electrical power	---		59.0	18.3 + 5.1(DHW)	22.5	25.7	18.7
Component Parameters 14. $\bar{I}_c$ MJ/hr·m²	1.12	1.12	1.16 (e)	1.03 (a)	0.98 (e)	1.18 (e)	0.98 (e)
Component Parameters 15. $\bar{\phi}$	0.424	0.415	0.406 (e)	0.448	0.471 (e)	0.529 (e)	0.471 (e)
Component Parameters 16. $(COP)_c$	0.600		0.605	0.600 (a)	0.527	0.567	0.527
Component Parameters 17. β	0		2.65	2.67	2.90	2.76	2.90
System Parameters 18. $\bar{Q}_u$* MJ/day	365.7	350.2	154.8	206.6	205.3	412.6	239.5
System Parameters 19. η_{ST}	20.3	22.3	8.7%	14.1% (with DHW)	11.8% (no DHW)	13.1%	16.7%	12.0%
System Parameters 20. SCOP = SCOP'	∞		1.59	6.4 (with DHW)	5.3 (no DHW)	6.1	9.6	6.8
System Parameters 21. $\bar{Q}_C$ MJ/day	219.4	205.0	4.4	113.8 (with DHW)	88.9 (no DHW)	108.0	201.2	101.6
System Parameters 22. η_S	20.3	22.3	0.4%	10.8 (with DHW)	8.4 (no DHW)	10.2	13.6%	9.6

(a) Assumed
(c) Calculated
(e) Calculated from experimental data
(t) Theoretical
* No DHW in 1978
† 5.1% error in storage heat balance
†† 3.0% error in storage heat balance

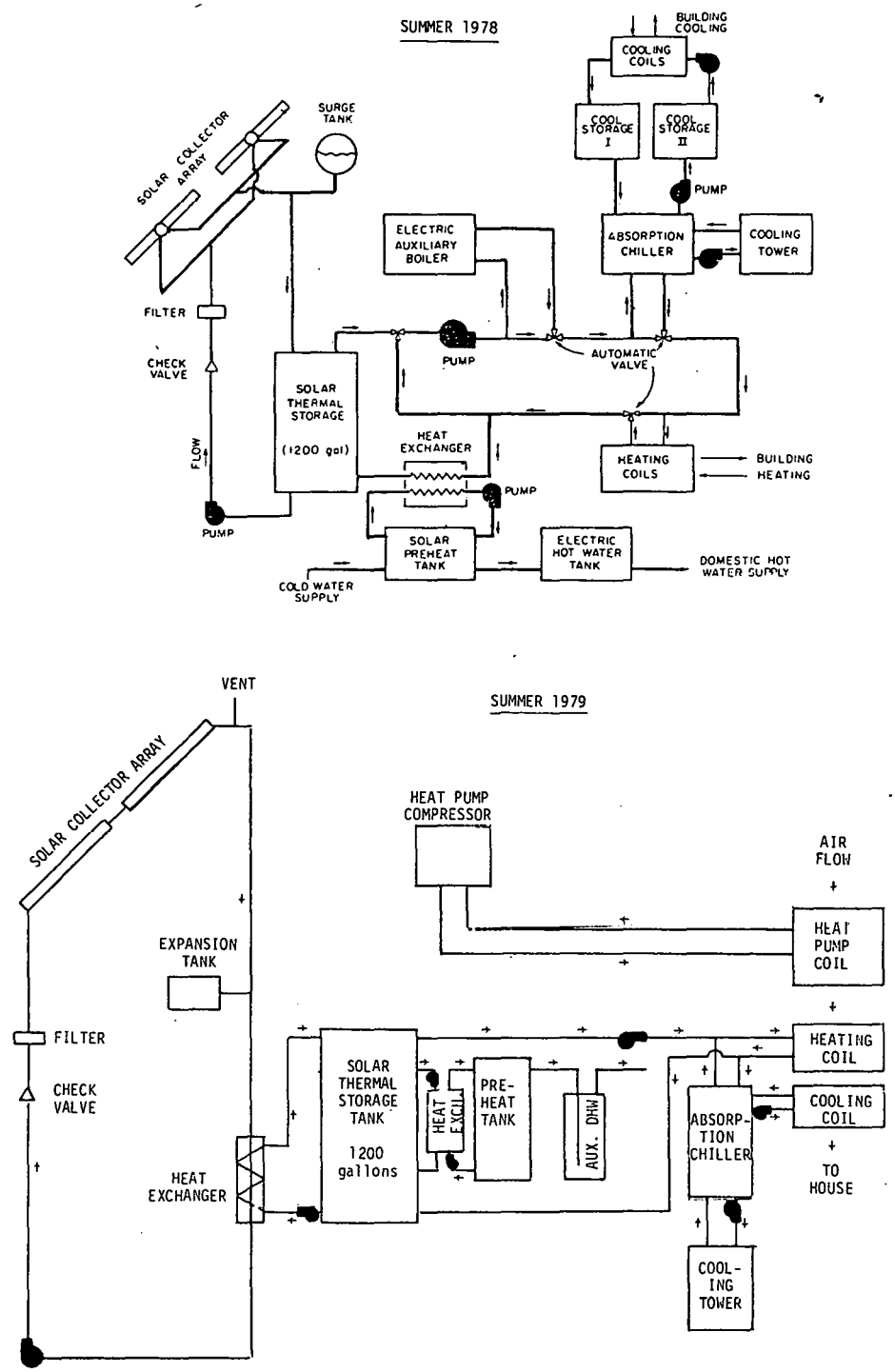


Figure 1. Solar System Schematics

TABLE 2. SYSTEM MODIFICATIONS

Loop	Length of Pipe (feet)		Pressure Drop (psi)		Pump Size (hp)	
	1978	1979	1978	1979	1978	1979
Collector	190	99	15	2.5	3/4	1/8
Exchanger	46	7	4.1	2.0	1/6	1/12
DHW preheat	24	17	17.0	9.0	3/4	1/8
DHW storage	--	13	--	3.0	--	1/20
Cooling tower	40	24	7.1	7.8	1/6	1/4
Circulating	110 + 90 aux. heat	37	9.7	1.6	1/4	1/25
Chilled water	62	4	5.8	5.0	1/6	1/12
Cool storage	40	--	2.7	--	1/12	--
Totals	602	201	61.4	30.9	2.31	0.74

RESULTS OF IMPROVEMENT IN THE DECADE 80 SOLAR HOUSE COOLING SYSTEM

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ABSTRACT

This paper compares the seasonal performance of the original and the improved versions of the Arkla solar-fired cooling system in the Decade 80 Solar House in Tucson, Arizona. Data have been taken continuously on the performance of the system from October 1975 to the present. Data from the peak cooling months of July and August of 1976, 1977 and 1979 are compared. The air conditioning system was extensively modified during the summer of 1978 when two Arkla 501-WF direct expansion machines were replaced with two Arkla WF-36 water chillers. The lower operating temperature of the new units allowed the collector to collect up to 75% more energy. The lower temperatures yielded lower heat losses, as did reduced piping. Electric power consumption did not increase in spite of adding an additional pump. The fraction of the cooling provided by solar increased about threefold, and non-renewable fuel savings about sixfold.

THE HOUSE

The Decade 80 Solar House (Fig. 1) is a single-story building of wood-frame construction, with a stone-on plywood skin. It is insulated with R-11 glass fiber batts in the walls and R-19 batts in the ceiling. There are essentially no windows in the south- and east-facing walls, and one small window on the west wall. The entire north wall is glazed with a tinted, dual pane glass.

The floor area of the cooled space is about 300m^2 (3200 ft^2). The mechanical equipment is housed in a small basement room, except for one air-handler in a closet off the garage. The house is otherwise built on a slab foundation.

The storage tank is buried outside immediately north of the mechanical room wall. It is a vertical cylinder with a capacity of 3000 gal, filled to 2800 gal. The tank is insulated with sprayed-on urethane foam on the top and sides and an unknown amount of "Cilsulate" granular poured-in-place insulation beneath it.

A cooling tower is located beyond the north-west corner of the house, about 30 m (100 ft) from the mechanical room.

The solar collector is the Revere laminated-panel combination roof/collector system, assembled in place by the construction crew. It is double-glazed with Polar Window" and was painted flat black, on site, with 3M "Black Velvet". It is plumbed with 23 sections of serpentine-flow-path tubes connected in parallel to feed and return headers. Its area is 178.65 m^2 (1923 ft^2). Its working fluid is a 30% solution of propylene glycol in water circulated at 25 gpm.

MECHANICAL SYSTEMS

The Old System

The original air conditioning system designed and built in 1973 - 74, consisted of two three-ton Arkla model 501-WF direct expansion air conditioners. Each separately served one of the two zones (east and west) of the solar house. Each unit cycled on and off according to the demand from its zone thermostat.

The New System

The new cooling system, installed during the summer of 1978, replaced the Arkla 501-WF units with two three-ton Arkla model WF-36 water chilers. The two units are connected into a single chilled water loop. During periods of low cooling demand one chiller will feed chilled water into the loop, supplying either or both zones. As the demand for cooling increases, the second chiller starts, supplying more chilled water to the loop.

This piping arrangement assures that the first chiller will operate as steadily as possible although the individual zone fan-coils may be cycling on and off. The chillers coefficient of performance is maximized by operating it as continuously as possible. Cool storage is not used.

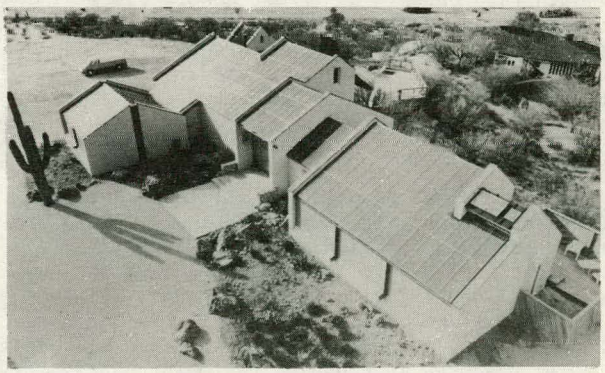


Fig. 1. The Decade 80 Solar House in Tucson, Arizona.

Table 1. Cooling Performance Data for the CDA Decade 80 Solar House, Data are for an average day of each month. Data rounded for clarity. All calculations were done before rounding.

	OLD SYSTEM				NEW SYSTEM	
	1976		1977		1979	
	July	August	July	August	July	August
$\bar{I}_T$ – Total insolation, kJ/m^{2}day	18,629	22,046	22,445	22,114	22,750	22,625
$\bar{I}'_T$ – On-time insolation, kJ/m^{2}day	11,398	13,612	12,385	14,100	15,905	14,735
$\bar{T}_a$ – Outdoor drybulb temp, C	29.9	30.7	30.8	30.6	31.2	28.3
Collector operating efficiency, %	13.2	15.7	10.1	10.5	18.4	17.7
Collector daily efficiency, %	21.6	25.4	18.2	17.1	26.3	27.1
$\bar{Q}_u^A$ – Collected energy, kJ/day	439,200	617,700	403,300	431,900	748,000	713,800
$\bar{Q}_L^{ext}$ – Storage loss to ground, kJ/day	-----	96,500 (calculated)	-----		66,250 (calculated)	
$\bar{Q}_L^{ext}$ – Piping loss, kJ/day	-----	57,500 (calculated)	-----		25,900 (calculated)	
$\bar{Q}_L^{int}$ – Loss to house, kJ/day	-----	8,130 (calculated)	----------			
$\bar{Q}_{load}^S$ – Solar energy to Arklas, kJ/day	286,100	346,300	236,500	364,300	633,000	623,000
$\bar{Q}_{load}^{Aux}$ – Aux. energy to Arklas, kJ/day	1,369,400	1,208,000	908,400	616,100	864,000	320,700
$\bar{Q}_{load}^T$ – Total energy to Arklas, kJ/day	1,606,000	1,492,000	1,101,300	938,700	1,366,200	926,000
$\bar{Q}_{cooling}$ – Chilled air/water, kJ/day	1,151,000	1,200,600	762,800	625,600	722,700	508,500
$COP_{cooling}$ – (total)	.72	.81	.69	.67	.53	.55
f–Fraction of Arkla load by solar	.15	.19	.18	.34	.37	.65
Basic electric power usage, (total) kJ/day	220,700	235,800	183,600	153,000	188,300	146,200
(solar) kJ/day	76,900	109,400	89,300	67,800	88,600	97,300
Total system operating power (total) kJ/day	408,000	436,000	340,000	283,000	347,700	270,600
(solar) kJ/day	125,000	185,000	149,000	109,000	148,500	167,900
(less blowers) kJ/day	92,800	136,000	114,700	80,800	108,900	115,700
Days of useful data	30	22	31	25	29	30

Economizer

An outdoor-air intake system with an enthalpy-sensing control was installed in the west zone of the house when the new chillers were installed. Its operation is monitored but those data are not included in this paper. No "free" cooling is included in the data that follow. During mid-summer, nighttime temperatures seldom drop low enough to make use of the free cooling, although the economizer is quite valuable in early and late summer.

Occupancy

The solar house was occupied at all times during 1976 by two persons, the host/caretakers. They preferred their air conditioning set to 76 F (24 C). The present owners were out of town during the summers of 1977 and 1979, but the air conditioning was maintained at 78 to 80 F (26 - 27 C). The house was entered and "occupied" several times a week and hot water was used in cleaning and maintenance. Thus the cooling load was lower during those two summers, and the infiltration was less because doors were opened less.

DATA SYSTEM

The computer data-taking system used in this project is designed around a Tektronix 31/53 programmable calculator system [1]. Temperatures are measured with YSI Series 400 thermistor probes. Flows are measured with Ellison Annubar sensors and meter. The pyranometer is an Eppley 8-48. It is periodically calibrated by DSET Inc., at an angle equal to that of the collector, to minimize the cosine error this model is known to have.

Each data point is read once every 2.2 minutes throughout the day, with the computer doing all data interpretation, reduction, integration and summing between scans. Data are printed on paper tape immediately, hourly and daily. Heat balances are taken across each component of the system.

Absolute accuracy of temperature measurements is $\pm$0.2 C; flow measurements are $\pm$5%. Heat balances normally agree within about 10%. Heat balances temperature intercomparisons are relied upon to check accuracy, in lieu of periodic recalibration of sensors.

The following sections discuss the data shown in Table 1.

Weather Data

Insolation is measured by a pyranometer mounted above the collector and in the same plane, 26½° south-facing. Table 1 shows that except for July 1976 average daily insolation was remarkably constant at slightly over 22 MJ/day.

Outdoor drybulb temperature is measured in the shade immediately north of the house. The sensor is exposed to breezes. The data indicate a relatively constant average temperature from month to month, except for a slightly cooler August 1979. Note that these temperatures are 24-hr averages, not daytime averages.

Indoor Temperature

The controlled temperatures inside the house were necessarily slightly different during the course of the testing, as the occupancy pattern changed. During the first year the temperatures were maintained at about 75 to 76 F (24 C) at the occupants' preference. During 1977 and 1979 test periods the thermostats were generally 78 to 80 F (26 - 27 C). Thus the cooling load was slightly less for the last two summers. Bear in mind, however, that the daytime temperature outdoors normally exceeds 100 F (38 C) in July and August; thus there was a significant cooling load on the air conditioning, even at an 80 F set-point.

Collected Energy

Table 1 lists the measured amount of energy, $\bar{Q}_u A$, collected and delivered to storage for the average day of each month. We note a significant decrease both in collected energy and in collector efficiency between 1976 and 1977. The probable cause was air entrapped in the collector tubing (thus blocking some flow passages and reducing effective collector area).

Before the 1979 test period an attempt was made to balance flow through the collector. Surface temperatures across the collector were measured with an infrared thermometer. It was found that fluid flow to the east and west ends of the collector was lower than to the central portion. Balancing cocks, installed when the system was built, were adjusted to somewhat equalize the fluid flow. This probably accounts for some of the increase in collector efficiency during 1979.

At the end of the 1976 test period a modification was made to the collector heat exchanger. A second, spare, heat exchanger was installed in series with the first, in a counterflow arrangement. This lowered the collector's operating temperature by an average of 5 to 7 F (3 to 4 C).

We see in Table 1 that the collected energy increased s ficantly in 1979 compared to the previous years; I in fact about 75% greater. The primary reason for this is certainly the lower operating temperature of the entire system. The original Arkla air conditioners required hot water at 190 to 210 F (87 to 99 C). The new chillers operate cooler, requiring a generator input temperature of only 170 F (77 C). The lower storage temperature permits the collector to operate more efficiently throughout the day. Typical noontime efficiency is about 30 to 34%.

Heat Losses and Gains

Heat losses from storage and piping to the outdoor environment, collectively called $\bar{Q}_L^{ext}$, are listed in Table 1.

Loss from storage tank to ground, part of $\bar{Q}_L^{ext}$, was measured by closing off all input and output lines to the tank for a 24 hour period and measuring the temperature decrease at the top, middle and bottom of the tank. At a 180 F starting temperature the tank lost 2.99 F, indicating a loss of 69,700 Btu/day (73,500 kJ/day).

Using this figure the tank loss from 170 F (80 C) would be 62,800 Btu/day (66,250 kJ/day). The loss from a 210 F (99 C) tank would be 90,600 Btu/day (95,600 kJ/day).

Loss from piping between the storage tank and the Arklas arises from 70 ft of 1½-in. Type L copper tube, insulated with 1-in. of jacketed glass fiber pipe insulation and 40 ft of 1-in. tube insulated with 3/4-in. closed-cell expanded elastomeric insulation. Total heat loss, $\bar{Q}_L^{ext}$, from the tubing at 175 F to the mechanical room at 105 F is calculated to be 1,360 Btu/hr. (1,440 kJ/hr.). Over an average operating day of 18 hrs, heat loss to the room is 24,600 Btu/day (25,900 kJ/day). Loss from the longer piping of the old system, at 210 F, is estimated to be 54,500 Btu/day (57,500 kJ/day).

Loss into the house, $\bar{Q}_L^{int}$, occurs only via the ceiling of the mechanical room which is insulated with two batts of R-11 glass fiber insulation so heat loss to the cooled space is low. Heat gain by the house, at 80 F from the mechanical room, at 105 F, is calculated to be 7,700 Btu/day (8,130 kJ/day).

Heat contribution by blower motors in each of the two air handler units which are mounted in the air streams subtract from the cooling available from the air conditioners. This heat gain is not considered in the heat balance calculations, but it has been measured. At high fan speed the ½-hp motor in the east air handler raises the air temperature by 3 to 3.3 F (1.7 to 1.8 C). The 3/4-hp motor in the west unit (at high fan) contributes 1.3 to 2.3 F (0.7 to 1.3 C) to the air stream. (Its air flow rate is higher.).

It would be useful to have the blower motors mounted outside of the air stream, since both air handlers are in uncooled spaces.

Domestic hot water heat consumption cannot be measured directly, and is not shown in Table 1. The dhw circulator pump, P-8, has a typical cycle time of less than 2.2 min, the computer scan interval. (It was instrumented in the old system for approxi-

mate measurements of operating time, however.) Some inferences about its heat consumption can be made.

Note in Table 1 the data for measurements of $\overline{Q}_u^A$ and $\overline{Q}^S_{load}$. Their difference is losses plus the domestic water heat consumption. There was a significant increase in the fraction of collected solar energy delivered to the cooling load during 1979 as compared to the previous years. The old dhw system used a tube-in-shell heat exchanger, with a pump on each side. A large mass of water sat in the heat exchanger and tubing, cooling between pump cycles. The new domestic hot water system uses as its heat exchanger a 50-ft coil of 1-in. copper tube submerged in the main storage tank. A 1/20-hp pump draws water from the domestic tank, sends it thru the submerged coil, and returns it to its storage tank. A much smaller mass of water is left to cool outside the tank between cycles.

Another improvement was also made. The new system is plumbed in pre-heater fashion; cold water entering the system passes through the submerged heat exchange coil on its way to the domestic water tank.

There is a continuously-circulating dhw pump, P-9. It was shut off at the end of July 1977, reducing the hot water load on the storage tank. Also, during August 1977 the hot water thermostat was set down from 135 F to 115 F in the owners' absence. (It was reset to 135 F in 1979.) The reduction in heat loss plus dhw heat consumption for that month ($\overline{Q}_u^A - \overline{Q}^S$) is evident in Table 1. At no time during the test periods was any auxiliary electric energy used to heat domestic water.

Cooling Performance

The total measured cooling provided to the house on an average day of each month is shown in Table 1 as $\overline{Q}_{cooling}$. The cooling load of the occupied house was 1,100 to 1,200 MJ/day while the cooling required by the (nearly) unoccupied house was about half to two-thirds that amount, 500 to 800 MJ/day. This reflects the higher thermostat settings, the lower infiltration and the lower internal heat generation. The decrease of about 200 MJ/day in the last month is apparently due to the 3 C decrease in outdoor temperature.

Solar fraction, the solar energy delivered to the air conditioners, $\overline{Q}^S_{load}$ is of more significance. With the new system about twice as much solar energy was delivered to the chillers as with the old. The solar fraction,

$$f = \frac{\overline{Q}^T_{load} - \overline{Q}^{Aux}_{load}}{\overline{Q}^T_{load}}$$

has leaped from 0.15 to 0.19 for the old system, to 0.37 to 0.65 for the new. (The balance of the required thermal energy was provided by a gas-fired boiler for both old and new systems.)

Coefficient of performance showed a disconcerting decline through the three years of testing. The high figures shown for the first year are believed to be in error, perhaps because of uncertainty in measuring air flow through the Arklas. The COP figures of the second year of testing seem to reflect more nearly the true performance of the direct-expansion Arklas.

The coefficients of performance for the most recent two months are surprisingly low at 0.53 to 0.55. A temperature profile made on the chillers in late July revealed some non-condensible gas accumulation in the chillers. They were vacuum pumped in August, and their coil temperatures were observed to drop by 10 to 15 F while their COP's increased somewhat.

In addition, it was discovered during August that the chillers were inadvertantly being over-fired. That is, their generators were receiving at 180 to 185 F (and up to 190 F from the auxiliary boiler), when they should have been fired at 170 F. Excess heat input to an absorption chiller is simply carried over to the condenser without providing further cooling. The chillers have a modified refrigerant charge to match the Tucson location. Their lithium-bromide solution has been diluted to permit them to operate at 20 to 25 F cooler generator input temperatures. This is possible when condenser water is available at 71 to 75 F, rather than the standard 80 to 85 F. A penalty is that their COP drops off rather sharply when the generator temperature exceeds 175 F.

Steady-state measurements made in late August indicate that a COP of 0.75 is regularly attainable when the proper input temperatures are maintained.

Electric Power Consumption

Table 1 shows several figures for electric power usage in order to accurately characterize the system and analyze its performance. Basic electric operating power includes only pumps P-1, P-2, P-3, P-7 and the cooling tower fan (and P-6 for the new system). The other devices, such as the boiler pump, blower motors, motorized valves and controls were not measured in the old system. Table 2 lists individually metered power consumers within the system.

When the new system was installed, watthour meters on the remaining motors and controls were added and a master meter was installed on the breaker panel input to monitor the total house cooling system. Thus measured watthour figures are given for the total electric consumption by the new system.

To calculate a total wattage figure for the old system for each test month, it was assumed that the unmeasured power in the old system was the same fraction of total power (46%) as with the new system. The figures given for total system operating power for the old system are calculated on this basis.

Differences between the old and new systems include the addition of the chilled water circulator, P-6; each new Arkla's solution circulating pump, (about 100 W each); and four fewer motorized valves in the new system.

As can be seen from the total system operating power figures, and from Table 2, this is indeed a large

system, consuming typically 75 to 100 kWh/day. This high figure primarily results from the physical layout, which was designed to meet neighborhood zoning requirements and for maximum consumer comfort and appeal in order to demonstrate solar heating and cooling in an attractive setting. As a result, air ducts tend to be long, convoluted and cramped; condenser water must be pumped to a cooling tower 100 ft away from the mechanical system; and the collector headers are necessarily circuitous to fit the multiply-sectored attic. In another house in the same geographic location with the same cooling requirements -- but designed around the cooling system -- every pump and fan motor would certainly need only be about 1/3 to 1/4 its present size.

Energy Efficiency Ratio and a Seasonal EER for the new system can be calculated. For a fully loaded system at steady-state operation, we define

EER = (Btu cooling output) / (watts input)
SEER = (Btu cooling/day) / watthours/day)

Using the rated output of 72,000 Btuh (6 tons, or 75,960 kJ) of cooling, and the wattages from Table 2 with the control system at 590 W, the basic air conditioning system (less collector and blowers) draws 3,213 watts. With the collector and blowers included the wattage drawn is 6021 W. (We quote the figures with and without the collector because their operating times do not match; further, manufacturers of split air conditioning systems often quote EER without the air handler included, as it is specified separately.)

Thus we find that

$$\text{EER (basic, steady-state)} = \frac{72000 \text{ Btuh}}{3213 \text{ W}} = 22.4$$

SEER (full system operation) = 12.0

For the average July 1979 day, measured SEER = 7.5. For the average August 1979 day, measured SEER = 6.8.

Had the chillers not been over-fired the operating time to provide the same cooling would have been shorter, and the measured figures would have been slightly higher. The system parameters discussed below show efficiency from a different viewpoint.

PERFORMANCE PARAMETERS

Component Parameters

Performance parameters are listed by month in Table 3. Their derivations are discussed below.

Critical insolation level, $\bar{I}_c$, is the minimum level of solar radiation required to maintain the collector plate at the temperature of the entering collector fluid [2]. It is calculated from

$$\bar{I}_c = \frac{F_R U_L}{F_R(\overline{\tau\alpha})} (T_{f,i} - T_a + \Delta T_{control})$$

where U_L = 0.77 Btu/ft^2-hr-$^\circ$F
(0.62 cover loss + 0.15 back/edge loss)

= 4.37 W/m^2-$^\circ$C = 15.7 kJ/m^2-hr-$^\circ$C

τ = 0.69 from PPG data.

α = 0.96 by recent measurement of absorber plate sample.

From a recent printout of data averaged over an operating day;

$T_{f,i}$ = 82.3 C

T_a = 36.3 C

ΔT control = 1.7 C

From Klein's assumption $(\overline{\tau\alpha})$ = 95% of $(\tau\alpha)$

= .95(.69) (.96) = 0.629.

Thus, $\bar{I}_c$ = 331 W/m^2 = 1,193 kJ/m^{2}hr

= 105 Btu/ft^{2}hr.

Uncertainties in the various parameters in the calculations above indicate that $\bar{I}_c$ could be about 4 to 20% higher. The most reasonable figures were used above.

Collector area, A_c, is 178.65 m^2. This is gross area.

Coefficients of performance and constants ξ, the electrical/thermal energy conversion factor, is taken as 2.50 as a typical electric air conditioner COP, while β, the heat loss "non-usefullness" factor is $(1 + \frac{1}{(COP)})$, and is calculated for each month.

COPs are listed in Table 1.

Monthly average daily utilizability, $\bar{\phi}$, is calculated for each month according to the method of Klein [2]. It is found by interpolation between Klein's Figs. 6, 7 and 8, and from calculation of the parameters X_c and $\bar{R}/R_n$. Klein's article should be referred to for a detailed discussion of the parameters relating to $\bar{\phi}$ and their derivations.

$\bar{R}$ is based on $\bar{K}_T$, the ratio of monthly average extraterrestrial radiation. Extraterrestrial radiation is inferred from Klein to be 39.5 MJ/m^2 day. $\bar{K}_T$ varies according to monthly insolation from 0.56 to 0.63.

R_n is the ratio of radiation on a tilted surface at noon to that on a horizontal surface at noon. This figure was derived from interpolation of the July 21 and August 21 data for "Solar Position and Insolation Values for 32 Degrees North Latitude" from Morrison and Farber's ASHRAE paper [3]. For July R_n = 0.974. For August R_n = 1.036 at our collector angle of 26½°.

Table 2. Electric power consumers, CDA Decade 80 Solar House

Device	Function/purpose	Rated Horsepower	Wattage	Kilowatt-hours per day*
P – 1	Collector pump	3/4	762	4.3
P – 2	Collector/storage pump	1/4	406	2.2
P – 3	Hot water from storage to Arklas	1	892	16.4
P – 4	Hot water to fan coils for heating	1/4	(300) est.	(winter only)
P – 5	Boiler circulator	1/4	450	3.2
P – 6	Chilled water circulator	1/4	280	5.2
P – 7	Cooling tower circulator	1	1017	18.3
P – 8	DHW heat-exchanger pump	1/20		0.05
P – 9	DHW loop circulator	1/20		2.1 (when used)
Cooling tower fan		1/2	346	6.0
System controls and other **				10.6
East zone air handler		1/2	646 (high)	
			573 (low)	10.0
West zone air handler		3/4	1155 (high)	
			1040 (low)	17.9

*Typical, based on July 1979 data.
**Including Arklas, valves, relays, differential controls, boiler controls.

Table 3. Performance Parameters, CDA Decade 80 Solar House.

	OLD SYSTEM				NEW SYSTEM	
	1976		1977		1979	
	July	August	July	August	July	August
Component Parameters						
$\bar{T}_c$ – Ave. daily solar threshold	---------- 1,193 kJ/m^2 hr = 331 w/m^2 ----------					
A_c – Collector area (gross)	---------- 178.65 m^2 ----------					
ξ – Conv. factor, electric to thermal	---------- 2.50 ----------					
β – Heat loss factor	2.39	2.23	2.45	2.40	2.89	2.82
$\bar{\phi}$ – Ave. daily utilizability	0.37	0.46	0.43	0.47	0.47	0.47
COP_c – Coef. of performance, cooling	0.72	0.81	0.69	0.67	0.53	0.55
System Parameters						
$\bar{Q}_u^*$ – Ave. daily solar heat to load, kJ/day	265,800	445,600	229,400	257,700	632,400	598,700
$\bar{Q}_c$ – Ave. daily savings, kJ/day	–34,100	–17,400	–99,100	–9,200	138,500	98,400
with air handlers excluded), kJ/day	24,400	85,400	–39,600	37,300	190,900	170,300
η_s – Ave. daily system efficiency, %	–1.0	–0.4	–2.5	–0.2	3.4	2.4
(with air handlers excluded) %	0.7	2.2	–1.0	0.9	4.7	4.2
η_{ST} – Ave. daily Solar efficiency, %	8.0	11.3	5.7	6.5	15.6	14.8
SCOP – Solar coef. of performance	1.53	1.95	1.06	1.59	2.26	1.96
(with air handlers excluded)	2.06	2.65	1.38	2.14	3.08	2.86

$\bar{R}/R_n$ varies by month from 0.88 to 0.94. (In all months but one it exceeds 0.9.)

Critical level X_c is the ratio of $\bar{I}_c$ to the daily average radiation at noon. It is calculated from $\bar{I}_c$ = 1114 kJ/m^{2}hr, calculated above, from the ratio $r_{T,n}$ of radiation at noon to daily radiation, and from average daily insolation on the collector surface for each month. Ratio $r_{T,n}$ = 0.130 for July and 0.134 for August, from the ASHRAE paper mentioned above. X_c varies by month from 0.39 to 0.49.

We finally find, by interpolation among Klein's graphs, that $\bar{\phi}$ ranges from 0.37 to 0.47, as listed in Table 3.

System Parameters

Solar Heat Delivered

The heat collected, $\bar{Q}_u^A$, is the actual, measured heat collected, as listed in Table 1. It is the basis of the calculated figures of Table 3.

$\bar{Q}^*_u$ is the calculated daily solar heat delivered to the load for the average day of each month. It is the measured heat collected less the losses from storage and piping to the outdoors and from the mechanical room to the house interior,

$$\bar{Q}_u^* = \bar{Q}_u^A - \bar{Q}_L^{Ext} - \beta \bar{Q}_L^{Int.}$$

It is listed in Table 3 by months. It compares approximately with the measured $\bar{Q}^S$ of Table 1.

Energy Savings

$\bar{Q}_c$ is the calculated daily savings in renewable energy resources for the average day of each month. It consists of the useful energy $\bar{Q}^*_u$, less the amount of cooling that could have been provided had the electric power used by the solar cooling system been used instead to operate compression air conditioning (at a COP of 2.50). $\bar{Q}_c = (\bar{Q}^*_u - \bar{E})(COP)_c$

There is some ambiguity in the definition of $\bar{Q}_c$ with respect to the role of the air handlers in the house cooling system. The operation of the air handlers, which are separate from the solar energy system and the Arklas, will be exactly the same whether the cooling is done by a solar cooling system or by a mechanical air conditioner. Accordingly, two calculations of $\bar{Q}_c$ are shown in Table 3, one which charges the energy cost of running the air handlers to the solar cooling system and the other, with the air handlers excluded, which does not. (The corresponding measured electric operating power requirements are shown as the last two lines of Table 1. Excluding the air handlers from the solar cooling system yields an average daily savings for the old system ranging from negative 39,600 kJ/day to a plus 85,600 kJ/day. The new system on the other hand saved 190,900 kJ/day in July and 170,300 kJ/day in August.

The parameter η_s, the average daily cooling system efficiency reflects these savings ranging from a - 1.0 in July of 1977 for the old system to + 4.7 for the new system in July of 1979. Parameter η_{st}, the average daily thermal efficiency $\bar{Q}^*_u/A_c\bar{I}_T$, is a measure of how efficiently the solar energy is used. It shows the effects of the system re-design which cause the efficiency to rise from about 8% to about 15% for the new system.

The solar coefficient of performance,

$$SCOP = (\bar{Q}^*_u/\bar{E})(COP)_c$$

improves from about 2 in the old system to about 3 in the new, with the air handlers excluded.

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NOTES

PERFORMANCE OF CONTROLLERS IN SOLAR HEATING SYSTEMS

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ABSTRACT

Inspections of more than one hundred solar heating installations have been conducted by technical personnel from Solar Control Corporation and Solar Environmental Engineering Company, Inc. In addition, a survey of more than two hundred manufacturers, architects, engineers, solar system dealers and solar system installers has been performed by Solar Control Corporation for the purpose of determining perceptions regarding controllers. Results from the system inspections and the survey point to several problems in the area of controllers. These problems, the survey and results from the inspections are presented in this paper.

INTRODUCTION

Solar Control Corporation's technical personnel have been hired to consult and troubleshoot more than 100 solar systems. In addition, under contract to DOE, they conducted a survey among a combination of more than 200 solar system manufacturers, architects, engineers, dealers and installers to determine their perceptions of the desired modifications and needs for new controllers.

Solar Environmental Engineering Company, Inc., Fort Collins, Colorado, has been designing and troubleshooting installations for several years and has been performing in a Reliability Assessment Team (RAT) program for SERI to report on the operating problems of many installed solar systems. SEEC has completed twelve detailed mechanical inspections of residential solar heating and domestic hot water (SH and DHW) systems in the State of Colorado. Most of the systems have been installed with financial assistance of the HUD Solar Demonstration Program and are less than one and one-half years old. The sample represents a random selection of air and hydronic heating and domestic hot water (DHW) systems using a variety of available equipment. Of the twelve systems studied, only one system operates at its expected mechanical, thermal and economic performance level. Four other systems perform well mechanically, but only one of these has thermal performance meeting design expectations. None of these four systems have a satisfactory economic return as anticipated by the system owners. None of the remaining systems work well mechanically or thermally. One system significantly raised the electricity bill of the house in which it was installed. SEEC engineers performing the system inspections have no evidence to indicate that the systems selected for inspection are lower in caliber than "average" state-of-the-art systems. Although the sample is relatively small, many other such inspections by SEEC engineers confirm the disappointing performance of the majority of present residential SH systems. Such failures should not be construed to mean that SH and DHW systems cannot or do not work. On the contrary, some solar system suppliers have excellent systems, which, as installation experience grows, promise very effective performance.

The survey and the field results showed agreement in the performance of controls and pointed to remedial actions. The results obtained from both the survey and the RAT Patrol are presented in the next sections.

RESULTS FROM THE SURVEY

The survey results are presented first for domestic hot water systems and then for space heating and/or cooling systems.

Domestic Hot Water Systems

The overwhelming majority (>78%) of responders indicated satisfaction with the state-of-the-art and performance of controllers for this function. Packaged control modules (i.e., pumps, etc.) seemed particularly trouble-free. Leading sources of trouble were at the installer level:

1. Installer induced failures of triac output devices.

2. Mismatch of controller logic to system freeze protection requirements.

At the large project (e.g., New England Electric, etc.) and OEM level, it was evident that usually no technical specifications establishing controller performance existed. This problem still is inherent in the controls provided by many leading manufacturers in that these firms are still unable to help these dealers' customers solve transient cycling, freeze and similar problems.

Installer skills, product familiarity and numeric temperature read-outs have virtually eliminated the installer check-out problems that were chronic 2-3 years ago. MTBF of available equipment is now in excess of 70,000 hours. Little quantitive energy reduction is cited by field users and OEMs in their choice of controls. Probe accuracy, tracking accuracy, optimum differential offset and hysteresis are quite secondary to the "if the sun shines, does it click" concern.

Space Heating and Cooling Controls

As compared to domestic water heating, very few (<20%) were satisfied with the controls for these functions. All field reports substantiated that this is a major area of concern. Field reports tended to incriminate the installer and/or the control manufacturers. To the extent that installation instructions weren't followed, this was true. To the extent that the equipment reliability has reflected product newness, this is also true. However, the major source of problems arose from attempts to interface the controls with the various components without the benefit of thorough system design. The lack of design information manifested itself in two ways: 1) a control strategy with no un-priortized contradicting conditions and 2) the lack of a clear wiring schematic. The study of the various manufacturers pointed out how little technical help they were prepared to give to the field. Few had the HVAC know-how and/or electrical expertise within their company. Generally, they were simply a component manufacturer (in many cases the collector manufacturers, with a few sketches in their promotional literature showing a simplistic strategy, i.e., a differential thermostat for collection and a two-stage thermostat for distribution. Little or no provision was made for an optimized strategy of the several hundred circuit permutations that exist with various furnaces, heat pumps and thermostats. Installers with limited design skills have been trying to address these problems. The generally depressed and disgruntled space heating market is largely a result of this benign neglect.

RESULTS FROM THE RAT PATROL

The following material has been abstracted from "Investigation of the Performance of Some Residential Solar Energy Systems in the State of Colorado," which has been submitted to Mr. Cordon Cross of the Solar Energy Research Institute as part of the reliability assessment work contracted to SEEC by SERI.

A listing of the significant findings of each system's inspection is given in Table 1. A tabular summary of the problems which were observed is shown in Table 2. Within each category of problems, the problems are listed in order of decreasing frequency of occurrence. The causes of the problems are also listed. The cumulative problems for each category are shown opposite the category heading. The first column of the table shows a contrived

TABLE 1. SUMMARY OF SIGNIFICANT FINDINGS OF SYSTEM INSPECTIONS

System #	Significant Conclusions of Inspection
1	Very poor system design. Poor installation quality.
2	Significant installation deviation from design.
3	Demand rate electric structure greatly diminishes economic solar savings.
4	"Homemade" collector virtually nonfunctional. Equipment located improperly by installer.
5	Very poor system design.
6	A workable air heating system after a few startup problems dealt with by the owner.
7	An air heating system with significant leakage problems as its only detriment.
8	An undeveloped drain down prototype system which failed.
9	A workable liquid heating system with a few design deficiencies in need of correction.
10	A system containing poorly developed solar components contributing to low performance, noise and odor problems.
11	A workable air heating system with diminished performance due to installer modifications of specified design.
12	A good example of a reliable solar DHW system which has been well developed and packaged and requires a minimal installation effort.

measure of the severity of the problem. Each "potential" or "minor" problem is assigned one problem unit while each "major" problem is assigned three problem units. A "major" problem is one which seriously degrades the performance of the system or prevents its operation entirely. Failures of components which can be considered replaceable as normal repair parts, such as motors or sensors, are not included. The total problem units for each problem and for each category of problems are shown in the first column. A pictorial representation of system problems is also shown in Figs. 1 and 2.

Three controller failures were noted. These problems can be attributed to the components. The problem of mislocation or omission of sensors is caused by installation errors, which may be due to inadequate instructions or supervision. Most systems were built by experienced conventional system installers who were inexperienced with solar system installations.

Solar contribution fractions were found to be overpredicted by system suppliers in all twelve systems inspected; although in a few cases the overprediction was minor. A graphical representation of the estimated error in these predictions is shown in Fig. 3. A variety of techniques was used to arrive at the expected contribution by the system supliers and SEEC engineers have found them all to be idealistic to some degree, ignoring real system losses. Even the well known FCHART program, as it is commonly used, ignores transport losses, heat exchanger degradations, effects of low flow rates in liquid systems and other deficiencies unless otherwise accounted for by the user.

The problems which were corrected and for which dates and costs were available are shown in Table 3. In some cases, the costs are estimates by SEEC engineers since the repairs were made under warranty by the system installer at no cost to the homeowner. The cost is included in the summary of maintenance costs since it must ultimately be born by the customer.

The Time Between Failures is shown in Table 3. For components which failed only once, the Time Between Failures is the time between the installation date and the failure. Twelve failures occurred, excluding installation errors, which were repaired before the Reliability Assessment inspection. The twelve systems inspected have been operating for approximately one year. Abnormal maintenance costs were $51.67 per system for the first year. This figure does not include the cost of normal maintenance such as filters, lubrication, etc. or correction of installation errors. It is evident that, at present, solar system maintenance costs are unreasonably high.

The Mean Time to Repair, at 26 days, is also much greater than desirable for a reputable solar energy industry. The length of this period has two primary causes. First, most installers lack specific solar heating system repair training and are, therefore, somewhat unwilling to attempt repairs. Also, the responsibility for repair of the system is often poorly defined and disputes over responsibility for repair result between the system buyer, seller and installer.

CONCLUSIONS

Controller problems continue to be significant with respect to solar heating and cooling systems but are not as significant in the case of solar domestic hot water systems. There is a definite need for clear specifications regarding the installation of controllers. Triacs continue to be a serious problem source in solid state controllers.

TABLE 2. SUMMARY OF PROBLEMS

	Total Problem units	Potential	Minor	Major	Number (cause)
A. GENERAL SYSTEM	33	2	4	9	2(1), 9(2), 4(3)
1. Poor performance	17		2	5	4(2), 3(3)
2. Freezing problem	5	2		1	3(2)
3. Leakage	7		1	2	1(1), 1(2), 1(3)
4. Noise	4		1	1	1(1), 1(2)
B. COLLECTORS	37	2	8	9	8(1), 4(2), 7(3)
1. Leakage in manifolds	13	1	3	3	1(1), 1(2), 5(3)
2. Leakage within collector	6	1	2	1	3(1), 1(2)
3. Slope	2		2		1(2), 1(3)
4. Broken or damaged covers due to:					
a. wind	3			1	1(1)
b. unknown causes	3			1	1(2)
5. Insulation degradation	3			1	1(1)
6. Serviceability	3			1	1(1)
7. Poor design				1	1(1)
8. Sealing			1		1(3)
C. PIPING OR DUCTING	24	1	14	3	18(3)
1. Leakage	8		5	1	6(3)
2. Poorly insulated	5		5		5(3)
3. Sized improperly	4		1	1	2(3)
4. Noise	1		1		1(3)
5. Freezing	1	1			1(3)
6. Installation quality	1		1		1(3)
7. Excessive heat loss	3			1	1(3)
8. Aesthetics	1		1		1(3)
D. VALVING OR DAMPERS	19	1	3	5	2(1), 4(2), 3(3)
1. Omission or Mislocation	11		2	3	2(2), 3(3)
2. Leakage	4		1	1	1(1), 1(2)
3. Sticking	4	1		1	1(1), 1(2)
E. STORAGE	10	1	3	2	2(2), 4(3)
1. Leakage	3		3		3(3)
2. Excessive pressure drop	3			1	1(2)
3. Excessive heat loss/freezing	3			1	1(3)
4. Overheating/boiling	1	1			1(2)
F. PUMP OR BLOWER	17		6	4	3(1), 2(2), 1(3)
1. Motor burnout	3		3		3(1)
2. Noise	4		1	1	1(1), 1(3)
3. Sized improperly	6			2	2(2)
4. Leakage	1		1		1(1)
5. Serviceability	3		1		1(1)
G. CONTROLLER	17		3	5	3(1), 1(2), 3(3), 1(4)
1. Sensor mislocation	4		1	1	2(3)
2. Inadequate room temperature control	3			1	1(4)
3. Component failure	5		2	1	3(1)
4. Sensor missing	3			1	1(3)
5. Improper application	3			1	1(2)

TABLE 3. SUMMARY OF MAINTENANCE COSTS

	Time to Repair	Cost to Repair	Time Between Failures
Blower Motor	30 days	$40	150 days
Blower Motor	30 days	$40	90 days
Blower Motor	21 days	$70	13 days
Blower Pulley	2 days	$20	120 days
Blower Pulley	2 days	$20	60 days
Controller Failure	14 days	$100	210 days
Controller Failure	60 days	$100	28 days
Blower Noise	120 days	$50	30 days
Damper Motor	2 days	$60	200 days
Pump Motor	14 days	$50	480 days
Pump Leak	14 days	$50	210 days
Collector Leak	17 days	$20	30 days
	316 days	$620	

MTTR = 26 days

MCTR = $51.67

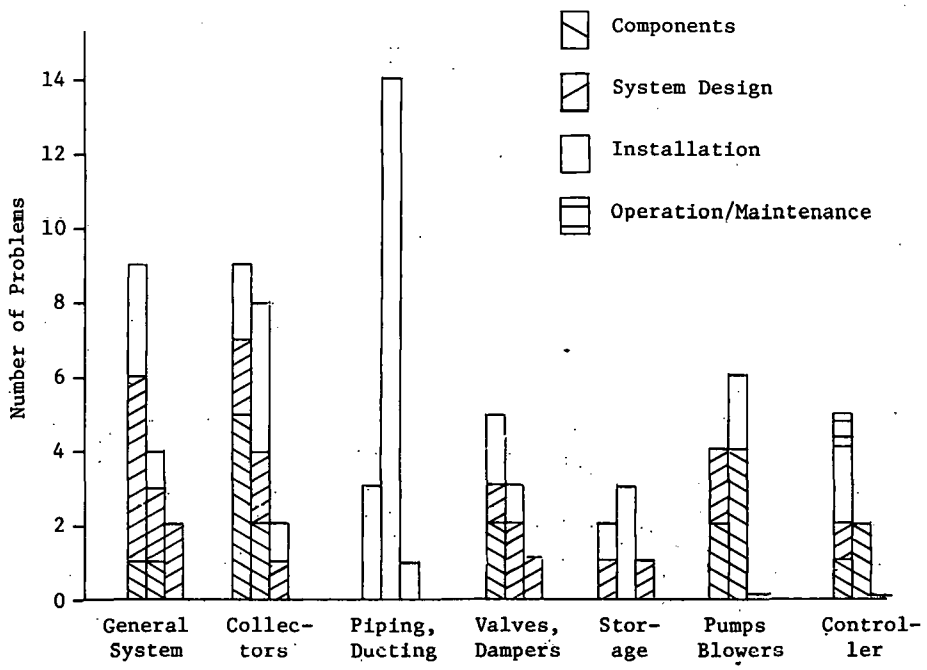


Fig. 1. Summary of Problems for Twelve Investigations

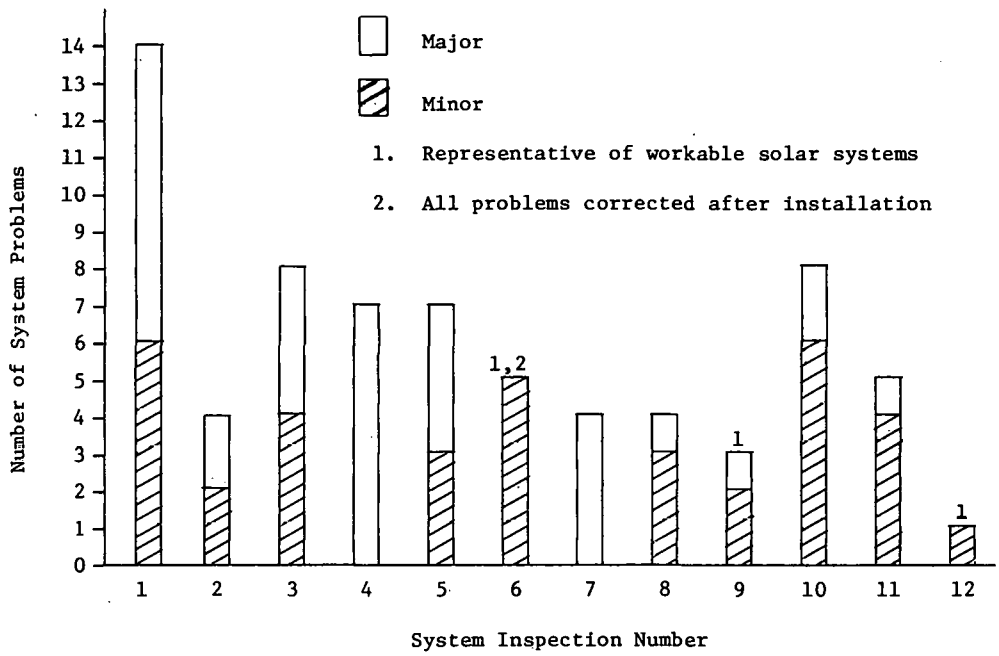


Fig. 2. Number of Problems per System

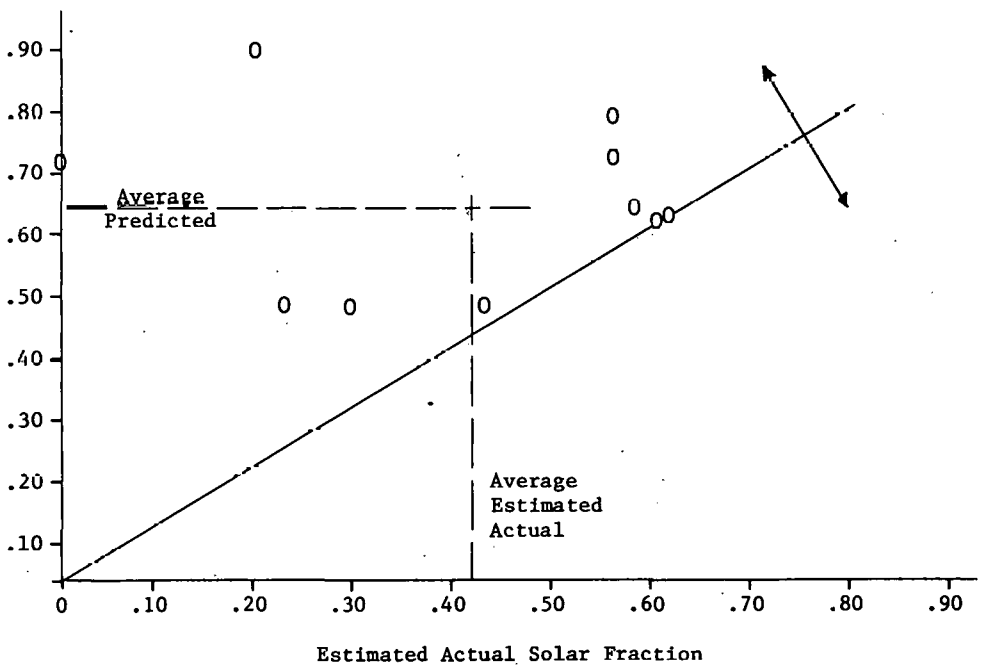


Fig. 3. Predicted Solar Contribution Fractions

OPERATIONAL IMPROVEMENTS IN THE CSU SOLAR HOUSE I SYSTEM SUPPLIED WITH HEAT FROM EVACUATED COLLECTORS

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ABSTRACT

During the months of May and June 1979, the CSU Solar House I operating system was completely re-piped, rewired and reinstrumented, new pumps were installed, a new high R-value modular heat storage tank with stratification enhancing manifolding was installed, and the system was modified so as to reduce pumping power and achieve improved performance. The new system became operational in mid-June and data have been recorded continuously since that time.

Energy collection and utilization were increased by these design changes as well as by modified operating procedures. The location and extent of heat losses were determined by making progressive additions of insulation to system piping, heat exchangers and valves, and measuring the resulting changes in solar heat collection and use. The evacuated collector cover glass with 0.82 transmittance was replaced with a cover having solar transmittance of 0.91.

SYSTEM CHANGES

During May and June 1979, extensive changes were made in the CSU Solar House I solar heating and cooling system that is shown in Fig. 1. The objectives of this redesign were to shorten piping lengths, reduce pumping power, and increase storage efficiency. The collector, absorption chiller, cooling tower, and heat exchangers were not altered. Attention was given to design details rather than to the use of major new components in the system.

The pumping power required in Solar House I has been substantially reduced from that used in past years, ough it is still higher a reasonable minimum requirement. A flow rate of 18 gpm through 480 ft^2 of collectors is provided by a 1/3 hp (250 watt) collector pump. The storage/heat exchanger pump is 1/12 hp (60 watt) and provides a flow of 7 gpm. This low flow rate to the storage tank is desirable so that stratification is enhanced. Hot water is supplied from storage to the chiller and air heating coil by a 1/3 hp (250 watt) pump, and domestic hot water supply is preheated in a loop requiring 2, 40-watt pumps. One horsepower (750 watts) is required for circulating cooling water, chilled water and the absorber solution in the chiller. The total pumping power for the system is approximately 1.9 hp (1400 watts) for cooling and 1/2 hp (390 watts) for solar supply to heating. This demand can be compared with approximately 3/4 hp (560 watts) used in Solar House III in which similar heating and cooling system components are used but where a principal objective is to minimize pumping requirements.

The only major system component replaced this year was the poorly insulated, cylindrical steel solar heat storage tank. A new 1100-gallon rectangular storage tank consists of interlocking panels containing 4 inches of urethane foam insulation sand-

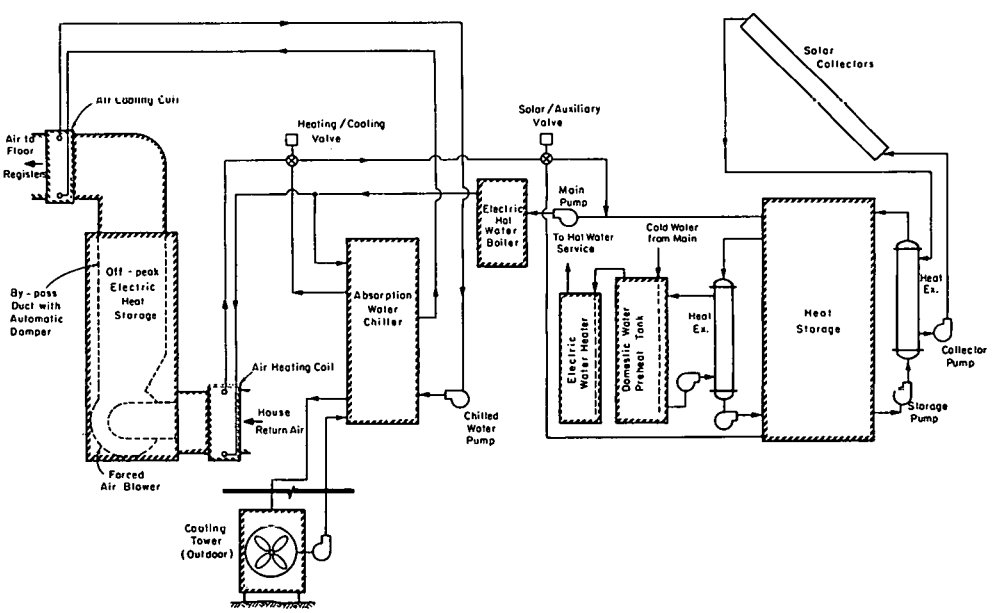


Fig. 1. Solar House I System Schematic, 1979

wiched between two galvanized steel sheets. A theoretical R-factor of 34 $ft^2 \cdot °F \cdot hr/Btu$ (6 $m^2 \cdot °C/W$) is reported.* Flat sides, bottoms, and top provide compact heat storage in a 6-foot cube. The panels were easily carried into the basement of the house, and their erection was completed in a half-day. A vinyl liner serves as the watertight interior of the tank, and a vinyl wrapping on the top panel prevents moisture problems.

Prior to the changeover, a two month experiment with a new stratified storage concept was conducted. This experiment has been continued on a permanent basis. Fluid flows into and out of the tank are through concentric tubes slightly shorter than the tank depth, where the outer tube is perforated with holes along its entire length. The flow is to or from the layer in the tank having the appropriate density level. This stratification approach was found to produce results that are as good as some more elaborate schemes and yet is inexpensive. The stratification manifolds were installed for the hot water preheat, heating or cooling, and collector inlets and outlets to the thermal storage and the flow rates were designed to be appropriately low level. Conventional inlets and outlets were also included so that mixing can be achieved.

Domestic hot water is now preheated in an exchanger supplied from the main storage tank. This exchanger had previously been located in the collector loop or in the loop supplying hot water from storage to space heating or cooling, so that energy could be transferred to the preheat tank only during operation of the collector or load pump, respectively. The auxiliary gas water heater has now been replaced with a conventional 40-gallon electric water heater.

Auxiliary energy is now supplied to the chiller by an electric boiler. The boiler is located in the load loop so that water being pumped to the chiller always passes through it. When solar storage temperature is below the set point (80° on, 70° off), an automatic valve isolates the solar storage tank, electricity is supplied to the boiler, and water circulates through the boiler and chiller in a closed loop.

Throughout the first part of July, pipes and other components were insulated, operating set points were established, and several operating and control problems were solved. A significant heat loss from storage early in the month was found to occur by thermosyphoning in all the tank inlet/outlet loops. Check valves were installed, alleviating the problem.

After installation of the check valves, insulation of piping and other components was started. By 19 July, most of the piping (but not the heat exchangers, valves and pumps) was insulated, and an overall storage heat loss coefficient of 0.029 $MJ/hr \cdot °C$ was measured with the tank vent open and no fluid flows in or out of the tank. By the end of July, with the heat exchangers and piping insulated (but not most of the valves, sensor fittings and unions), the loss coefficient was 0.022 $MJ/hr \cdot °C$, corresponding to an overall R factor of 14 $ft^2 \cdot °F \cdot Btu$ (2.4 $m^2 \cdot °C/W$). Heat loss attributed to the tank consisted not only of small direct losses through the tank walls, but also some vaporization loss, conduction through piping from the tank, and other sources. There are 13 penetrations through the top of the tank, at least twice as many as in a normal installation, to accommodate both stratified and mixed tank operation as well as a two-inch copper vent pipe.

Through the first half of July, the chiller cycled frequently. This condition resulted in low overall COP's from disproportionately frequent transient chiller operation. The problem was due to an excessively narrow, 0.2°F, deadband in the room thermostat. The fault was corrected by 20 July by disabling an anticipator in the thermostat cooling circuitry.

The electric auxiliary boiler has four stages that are powered sequentially at 20 second intervals. When the high set point is reached, all stages turn off and the turn-on sequence is repeated as soon as the low set point is reached. The oscillating temperatures reduced chiller performance. By disabling the fourth stage of the boiler, continuous operation at about a 74°C supply temperature was achieved. At this temperature, COP is at maximum and cooling capacity is decreased about 20 percent.

Maximum COP's indicated by the manufacturer's performance curves† and achieved two years ago in Solar House I operation were not reached during the 1979 summer. Possible causes are non-condensable build-up and scale deposits in the chiller. These possibilities are being investigated.

Tank temperature stratification was normally about 4°C to 6°C during combined collector/chiller operation. To stabilize tank temperature profiles and also to provide a more uniform temperature to the chiller generator, a portion of the return flow from the generator was added (by manual valve adjustment) to the supply flow. The results were a steady 74°C heat supply and greatly increased stratification during collector/chiller operation. Temperature differences between the bottom and top of the storage tank were then usually 12°C to 15°C. An automatic tempering valve circuit was therefore designed and installed in September.

A new cover glass having 91 percent solar transmission was installed on the Philips collector on 5 and 6 July by Philips and CSU personnel. The new glazing replaced the original glass that had experienced a reduction in transmittance to about 0.82 due to darkening by exposure to ultraviolet radiation in sunlight.

RESULTS

Table 1 is a summary of average daily performance of the solar cooling system in Solar House I. Jul 1979 is divided into two periods, the first during

*Bally Case and Cooler, Inc., Bally, Pennsylvania

†Arkla Industries, Evansville, Indiana, WF-36, 3 ton lithium bromide absorption chiller

which pipes were being insulated and most of the operating adjustments were being made, and the second during which the system was operating under more normal conditions. Also included is the September 1978 period of operation before the tank, plumbing, and instrumentation changeover.

TABLE 1. SOLAR HOUSE I PERFORMANCE SUMMARY

Measurement (all data in MJ/day)	Sept 1978	July 1-16 1979	July 17-31 1979	Aug 1979
S45 incident radiation (1)	961.1	941.5	865.5	842.3
S450 incident, collector on (2)	815.0	743.0	674.5	646.1
Heat delivered to storage	308.2	308.5	285.3	284.4
Collection efficiency, on (3)	.378	.415	.423	.440
Collection efficiency (4)	.321	.328	.330	.338
Storage loss to interior	94.0	99.8	61.8	64.8
Solar energy to chiller	178.1	152.9	187.2	187.5
Auxiliary energy to chiller	263.3	0	132.0	33.7
Cooling supplied, total	166.3	66.1	158.1	99.5
Cooling supplied by solar	70.6	66.1	89.5	81.8
COP	.377	.432	.495	.450
Solar energy to hot water	13.7	43.1	39.3	41.3
Auxiliary energy to hot water	24.8	7.8	13.6	6.8
Number of days of data	30	16	15	28
Heat balance error, %	1.6	12.5	8.1	6.0

Notes:
(1) Total daily radiation on 45° south-facing surface
(2) Solar radiation on 45° south-facing surface during period of collector pump operation
(3) Heat delivered to storage divided by solar radiation during collector pump operation
(4) Heat delivered to storage divided by total daily solar radiation

As can be seen from the table, the solar heat loss into the building interior has been substantially reduced, both from the old tank and plumbing of 1978 and by insulation on the pipes after July 16. Improvements can still be made in these loss values by insulating valves, pumps, and instrument fittings.

Longer running times for the chiller have been achieved due to the increased tank stratification. Analysis is presently underway to determine the extent to which the lengthened times exceed those of a mixed tank.

The periods shown in the table do not have equal insolation levels and comparable tank-to-ambient temperature differences, so collector efficiency
arisons should be based on Fig. 2 and not on
table. Figure 2 indicates that the improvement in instantaneous system efficiency achieved by use of the new cover glass is on the order of

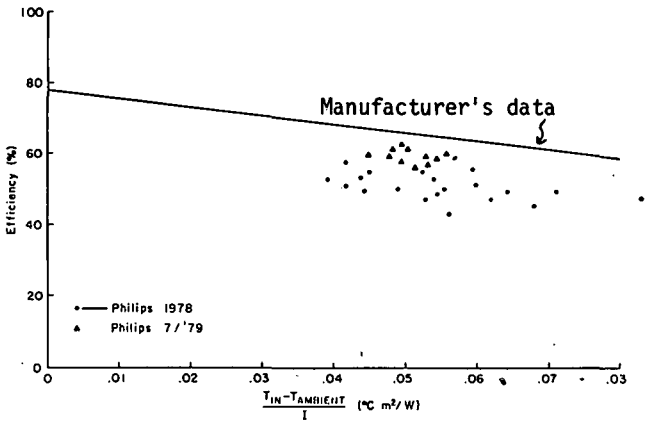


Fig. 2. Selected Hourly System Solar Collection Efficiencies Based on Absorber Area

eight percentage points based on selected midday hourly readings. Figure 3 shows that the increase in energy collected averages about 70 MJ when operating at average daily temperature differences between storage and ambient of about 45°C.

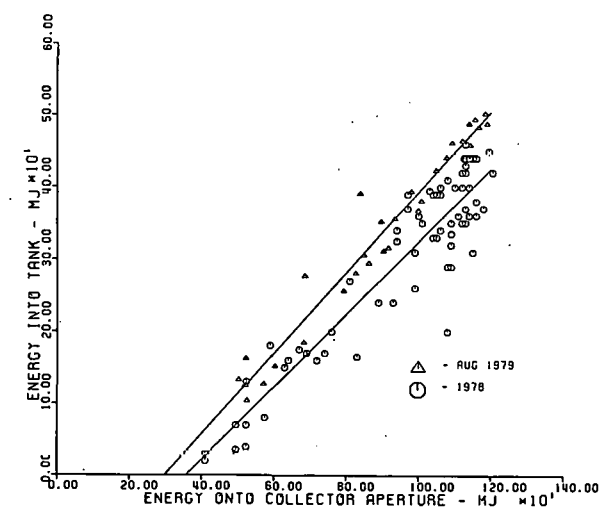


Fig. 3. Daily Solar Availability and Collection (Average daily temperature of storage approximately 45°C above ambient)

Chiller COP's varied because of manual tempering experiments, control changes, and leakage of hot water from the storage tank to the duct heating coil (during the 1978 period). Chiller performance trends are therefore inconclusive.

It is seen from Table 1 that both the solar energy to hot water and the total energy to hot water are greater after the changes. The increases are due to a more typical hot water usage in the building and to relocating the service hot water preheat heat exchanger so that the preheat tank can draw on solar any time solar storage can provide it.

Data reliablity has considerably improved, as

evidenced by fewer missing data and substantial reduction in errors encountered in processing raw data.

CONCLUSIONS

Tank, plumbing, and instrumentation changes in Solar House I have resulted in substantial improvements in performance and reliability. Additional performance improvement is expected in the experiment itself and in the design and operation of systems for general use.

ACKNOWLEDGEMENT

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SOLAR SYSTEMS THAT WORK WELL

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ABSTRACT

A great deal has been learned from the HUD Solar Program regarding problems and errors that occur in solar systems, either because of faulty design or because of defects in installation, operation and maintenance. The lessons learned can be used for determining what makes solar systems work well, and in suggesting designs and procedures for solar systems that will be free of the familiar defects. In this paper, we examine selected solar systems in the HUD Solar Residential Demonstration Program, to illustrate what makes solar systems work well.

INTRODUCTION

The projects in the HUD Solar Program have given us a great deal of information about the different kinds of problems that can occur in solar systems. Some of these problems arise from defective design, while others arise from defects in installation, operation and maintenance. The lessons learned from the operational results in these projects can be used for determining what is needed to make solar systems work well. In this paper, we examine selected solar systems in the HUD Program to see to what extent they have the desired characteristics.

The main constraints in selecting the projects were that they should be instrumented projects for which reliable operational results are available for at least a few months, and that the system performance should be reasonably good and free of major problems. These constraints greatly limit the choice of the systems available for discussion. In particular, we are unable to find instrumented air systems that satisfy the above requirements.

A large number of factors determine whether a solar system will work well. These will be outlined in the next section. We shall summarize the overall performance of the solar systems in terms of the following performance factors (where no ordering is implied):
(a) Monthly Solar Energy Conversion Efficiency, which is the ratio of the Useful Solar Energy delivered to the load to the Solar Energy Incident on the collecting surface.
(b) Monthly Solar Fraction, which is the ratio [Solar Energy Used - Operating Energy Used]/Load. Care is needed in defining each term, especially for passive systems.
Savings of conventional energy each month.
(d) Solar System C.O.P., which is the ratio of the solar energy used to the operating energy.
(e) Durability and reliability of the system.
(f) Comfort level (for a space heating system).

These performance factors will give an overview of the overall system performance. They depend on proper design, installation, and operation and maintenance. Proper system design depends on a large number of factors, which will be classified and listed in the next section.

DETAILED DESIGN CRITERIA FOR JUDGING A SOLAR SYSTEM

A solar system must be examined as a whole to be able to judge its performance properly. For a residential solar space heating (or space cooling) system, the building and site characteristics should also be taken into account. The detailed design criteria for judging a solar system will be grouped as follows:

A) Energy conservation characteristics
B) Passive solar characteristics
C) Mechanical (or active) system characteristics
D) Overall system characteristics

These characteristics, along with proper installation and proper operation and maintenance, determine how well the system will work. The detailed criteria are classified and listed as follows:

A. Energy Conservation Characteristics

1. Proper selection of design temperatures and comfort factors.

2. Protection from outside temperature variations.

This will include features such as

(a) a compact building form, with minimum surface exposure, in a cold climate;
(b) the use of earth berming/undergound structures;
(c) the use of utility spaces as buffers;
(d) proper insulation of the building envelope;
(e) reducing heat loss through glazing, by using double panes, storm windows, movable insulation, etc.
(f) use of massive walls for thermal inertia.

3. Protection from Infiltration and Wind

(a) use of site and vegetation as wind breaks;
(b) use of building orientation and form;
(c) proper weatherstripping and caulking;
(d) allowing for summer ventilation while cutting down the effects of the winter wind.

4. Protection From Overheating

(a) proper siting and orientation to minimize exposure to the summer sun, and reducing the east and west glass areas;
(b) use of natural shading devices;
(c) use of overhangs, and recessed windows/doors;
(d) use of natural and induced ventilation;
(e) taking care to allow solar access in winter, while providing for summer shading.

B. Passive Solar Heating & DHW Characteristics

1. Proper siting and orientation. Adequate exposure and absence of winter shading.

2. Adequate south-facing glazing area, with heat loss minimized.

3. Selection of interior surfaces for absorbing sunlight. Cutting down glare problems.

4. Passive Storage Characteristics
(a) Sufficient storage capacity
(b) Exposure of storage mass to direct sunlight in the winter
(c) Proper thickness of massive walls and floors
(d) Proper location of insulation
(e) Making water walls leakproof

5. Heat Distribution System Characteristics
(a) If the system depends on natural convection and radiation, will these be adequate? Will the air flow along the designed paths?
(b) For thermosiphoning, is the storage higher than the collecting surface by the proper amount?
(c) If blowers are used, will the operating energy usage be small?

6. Passive System Controls

Are the controls properly designed, simple and reliable? How much of the control system depends on manual operation? What is the degree of occupant participation required?

7. Protection from Overheating

Has this been ensured by proper storage sizing, and use of venting and shading devices?

8. Other Considerations
(a) Will the system be easy to install, operate and maintain? Are the different parts of the system easily accessible for maintenance?
(b) Have proper building materials and sealants been used? Are questions of weather proofing, moisture proofing, and freeze protection addressed?
(c) Has structural integrity been ensured?
(d) Are safety features included, esp. fire safety?

C. Active System Characteristics

1. Adequate collector area
2. Choice of collector materials
3. Collector tilt and orientation
4. Absence of winter shading
5. Adequate freeze protection
6. Protection from overheating
7. Selection of heat transfer fluids
8. Heat exchanger design and sizing
9. Storage material and size
10. Adequate controls, including off normal operation
11. Proper sizing of pumps, fans, pipes and ducts
12. Proper valving
13. Potable/Non-potable water separation
14. Installation considerations
15. Cutting down operating energy use

D. Overall System Characteristics

1. Adequate back up system
2. Are the parts of the system well integrated?
3. Absence of major design/installation defects
4. Is the overall system simple and reliable?

EXAMPLES OF SYSTEMS

In this section, we shall discuss selected examples of operating solar systems, to see to what extent they have the characteristics desired of solar systems. The selection here is severely limited by the space available, and by the constraints discussed in the Introduction.

Example 1 A-Frame Industries Site, Kaneohe, Hawaii

System Description: This is a simple active solar system, used for heating DHW in a single-family residence. The system uses 77 sq. ft. of flat plate collectors, with a 21° tilt, facing 10° east of south. The heat transfer medium used is potable city water, in an open system. The solar-heated water is stored in a 120 gallon tank. Auxiliary energy is provided by an electrical heating element. The control system is based on a simple differential controller. No freeze protection is provided for, because it is not required in this warm climate. Table 1 summarizes the system performance.

Discussion: This is an example of a simple system, which seems to be well-designed and installed properly, and has been free of any serious problems. The climate in that location has enabled a very simple system to be used. The main problem reported is that the measured collector efficiency was lower than the predicted value (by a factor of 0.6). Also the measured collector array efficiency was 28% as compared to the predicted 58%. The discrepancy was traced to the flow rate assumed in the analysis. The only other problem reported is that in some months the heat loss from the DHW system has been large. e.g. In February 1979, the large heat loss of 0.29 MBTU from the DHW system reduced the solar fraction from a possible 83% to an actual 62%.

____e 1 Quantitative Information and Performance Factors for The A-Frame System
(All Energy Values are in MBTU = A Million BTU)

	Feb. 1979	March 1979	May 1979
Monthly Incident Solar Energy	2.05	3.873	4.048
Monthly Collected Solar Energy	0.533	0.968	0.978
Operational Collector Efficiency	0.33	0.29	0.285
Collector Efficincy based on Total Incident Energy	0.26	0.25	0.24
Average Daytime Ambient Temp.	72°F	75°F	78°F
Average Hot Water Temp.	116°F	140°F	136°F
Daily Hot Water Usage (Gallons/Day)	51	42	53
Monthly DHW Load	0.57	0.743	0.809
Monthly Solar Energy Used	0.519	0.914	0.933
Monthly Operating Energy for Solar DHW	0.052	0.07	0.074
Energy Loss Between Collector and DHW Subsystem	0.014	0.54	0.045
Monthly Energy Loss from DHW Subsystem	0.29	0.054	No Info.
Monthly Auxiliary Energy Used for DHW	0.323	0	0
Solar Fraction for DHW System	62%	100%	100%
Solar Conversion Efficiency	0.26	0.236	0.23
Solar System C.O.P.	10	13.1	12.6
Monthly Savings of Conventional Energy:			
(a) Electric	0.519	0.914	0.933
(b) Fossil Fuel	-	-	-

Example 2 Facilities Development Gas Company Site
San Diego, California

Description of System: A 3-story, multifamily condominium with 31 units in San Diego, California. Solar energy is used for preheating DHW for the complex, in an active system with 520 sq. ft. of flat plate collectors. Potable water is the heat transfer medium. The storage tank is a 1000-gallon, underground tank. Back up is provided by electrical heating. The control system is very simple. Freeze protection is provided by warm water circulation through the collectors.

Table 2. Quantitative Information and Performance Factors for the Facilities Development System
(All Energy Values are in MBTU = A Million BTU)

	Jan. 1979	March 1979	June 1979
Monthly Incident Solar Energy	18.43	25.7	21.2
Monthly Collected Solar Energy	7.52	12.2	9.1
Operational Collector Efficiency	0.48	0.53	0.45
Collector Efficiency Based on Total Incident Energy	0.41	0.433	0.43
Average Daytime Ambient Temp.	60°F	63°F	71°F
Average Hot Water Temp. Delivered		No Information	
Daily Hot Water Usage (gallons/day)	749	1189	979
Monthly DHW Load		Not Determined	
Monthly Solar Energy Used	6.4	11.1	7.8
Monthly Operating Energy Used in Collector Loop	0.18	0.23	0.30
Monthly Energy Loss Between Collector & Storage	0.66	1.3	0.32
Monthly Energy Loss from Storage	0.9	-	1.1
Monthly Average Storage Temp.	90°F	89°F	98°F
Monthly Auxiliary Energy Used Towards DHW Load	42.4	45.8	33.2
Solar Fraction of Total Energy Used	13%	20%	19%
Solar Conversion Efficiency	0.35	0.43	0.37
Solar System C.O.P.	35.6	48.4	26
Monthly Savings of Conventional Energy:			
(a) Electric	6.4	10.9	7.8
(b) Fossil Fuel	-	-	-

Discussion: The measurements made on this system were incomplete; there were also a few minor discrepancies. The hot water load and the temperature of the hot water supplied were undetermined because of inadequate instrumentation. The system performed approximately as expected from a modified f-Chart ysis. Between September 1978 and March 1979, ough the measured solar energy delivered to the load was about 12% less than the expected value, solar energy satisfied 23% of the DHW requirements (as compared to an expected 22%). Although the solar fraction during the months shown is relatively small, this is not due to system inefficiencies; the expected solar energy used is quite close to the measured quantity. The measured solar fractions during September, October and November 1978 were 51%, 45% and 37% respectively.

In summary, the system appears to be properly designed and installed, and free from any serious problems. The collector area is undersized, considering the total energy required by the system. The losses are within about 10% in the winter months. Improved instrumentation will free the gaps in the solar system information.

Example 3 Homes By Marilyn Site, Albuquerque, N.M.

System Description: A single family house, with an active solar energy system, for both space heating and DHW. 335 sq. ft. of flat plate collector. The heat transfer fluid is ethylene glycol. Storage is in an 1000-gallon underground tank. Solar heat from the collectors is delivered to a liquid-to-liquid heat exchanger in the storage loop, and to a liquid-to-air heat exchanger in the space-heating subsystem to heat outside air for the heat pump source. City water is preheated by circulating it through a heat exchanger in the storage tank. Back up for space heating is from a heat pump and electrical heating.

Table 3 Quantitative Information and Performance Factors for Homes By Marilyn System
(All Energy Values are in MBTU = A Million BTU)

	March 1979	April 1979	May 1979
Monthly Incident Solar Energy	16.3	16.5	14.7
Monthly Collected Solar Energy	4.1	3.73	2.4
Operational Collector Efficiency	0.33	0.33	0.31
Collector Efficiency Based on Total Incident Energy	0.256	0.23	0.16
Average Daytime Ambient Temp.	56°F	66°F	75°F
Monthly Space Heating Load	5.35	1.65	0.21
Operating Energy Used in Space Heating	0.75	0.215	0.024
Monthly Solar Energy Used for Space Heating	2.73	0.875	0.14
Auxiliary Energy Used for Space Heating	0.98	0.29	0.05
Average Solar Storage Temp.	114°F	135°F	130°F
Solar Fraction for Space Heating	51%	53%	64%
Monthly DHW Load	1.85	0.99	1.35
Daily Hot Water Usage (Gallons/Day)	103	63	92
Average Hot Water Temp. Delivered	132°F	133°F	127°F
Operating Energy Used for DHW	0.23	0.23	0.235
Monthly Solar Energy Used for DHW	0.49	0.80	1.17
Losses from DHW Subsystem	0.02	0.40	0.39
Auxiliary Energy Used for DHW	2.76	0.58	1.13
Solar Fraction for DHW	50%	70%	66%
Monthly Total Load (Heating & DHW)	7.2	2.64	1.56
Monthly Solar Energy Used by System	3.51	1.68	1.31
Monthly Operating Energy for System	1.66	0.987	0.81
Monthly Auxiliary Energy Used by System	2.36	0.787	1.19
Monthly Storage Losses	1.5	2.2	1.4
Solar Fraction for System	51%	59%	66%
Overall Solar Conversion Efficiency	0.22	0.1	0.09
Solar System C.O.P.	2.1	1.68	1.61
Monthly Savings of Conventional Energy:			
(a) Electric	1.25	0.88	0.253
(b) Fossil Fuel	-	-	-

Discussion: There have been a few errors in the measurements; their sources are not known. Potential sources are detective flow sensors, actual flow variance caused by the collector pump, and improper operation of the valves. A problem during some months is that because of the large hot water usage, and the small (30-gallon) DHW preheat tank, water replenishing this tank did not have sufficient time to be heated by solar energy before being delivered. Therefore electrical energy was frequently required to heat the water in the DHW tank to the thermostat setting. Another problem was that although the space-heating subsystem was designed to use the storage when the storage temperature was higher than 85°F, in some months it could not do so when the storage temperature went below 105°F. This was probably because the cycling DHW preheat loop continued to extract energy from storage as long as there was a DHW demand. The storage needed to be reheated to replace the storage losses and the energy used by the DHW preheat, before solar heat could be used for space heating. Apart from the above problems, the system appears to have functioned well.

›le 4 Hullco Construction System, Prescott,Az.

System Description: A passive solar space-heating system for a single-family residence, with 1056 sq. ft. of living space. Solar collection is through 400 sq. ft. of double glazed Kalwall panels on a greenhouse on the south side. There is a sliding glass door in the massive wall between the greenhouse and the house. The storage consists of 3 parts - the concrete floor of the building, the wall between the greenhouse and the building, and two rock beds under the floor on the north side of the house. Fans help move the heat from the greenhouse area to the rock bed, and from the rock bed through the vents to the north side of the building. The house heat losses are reduced by heavy insulation, and earth berming against the north wall. The greenhouse is vented to avoid overheating. Back up heating is provided by electrical radiant panels in each room and by a wood stove.

Discussion: This was, on the whole,a very satisfactory solar system. Most of the load was supplied by passive solar heat; during the winter of 1978 - 1979, solar heat provided about 75% of the heating. The back up wood stove has also worked satisfactorily. The system performance level during the year was slightly below design specifications, but this was accounted for by both the severe winter and some features of the owner's operation of the system. The comfort levels were reasonable most of the year. There were some problems with summer overheating and winter interior humidity levels; also, sometimes there was a significant difference between the comfort levels between different zones. These problems can be corrected by proper operation and proper use of the system controls.

For instance, the actual infiltration rate (of between 1 and 1 1/2 air changes per hour) was higher than the design value of half an air change per hour. The reason was that when the relative humidity inside increased in the winter, the occupants opened the windows a little to allow the dry outside air to relieve the humidity. The increase in the humidity seems to arise mainly from the humid greenhouse air blown through the rock bed storage and into the house. Shutting off the blowers when the humidity increased helped to relieve the problem. Apart from these problems, the system has worked well. It has also not given any maintenance problems until now.

Table 4 Quantitative Information and Performance Factors for the Hullco System
(All Energy Values are in MBTU = A Million BTU)

	Dec 1978	Feb. 1979	March 1979
Monthly Incident Solar Energy	20.0	20.2	20.6
Monthly Collected Solar Energy	10.4	9.6	8.3
Monthly Space Heating Load	10.5	9.2	8.4
Average Daytime Ambient Temperature	42°F	48°F	52°F
Average 24-Hour Ambient Temp.	31°F	35°F	41°F
24-Hour Ambient Temp. Swing (Max. - Min.)	39°F	19°F	15°F
24-Hour Average Building Interior Temp.	72°F	72°F	71°F
24-Hour Building Temp Swing (Max. - Min.)	8°F	8°F	7°F
Average Storage Temperature	71°F	70°F	70°F
Monthly Solar Energy Used (Space Heating)	10.5	9.2	8.35
Operating Energy Used for Space Heating	0.08	0.085	0.1
Auxiliary Energy Used for Space Heating	0.03	0.008	0.006
Solar Fraction for Space Heating	100%	100%	100%
Overall Solar Conversion Efficiency	52%	48%	41%
Solar System C.O.P.	131	108	83.5
Monthly Savings of Conventional Energy:			
(a) Electric	10.43	9.1	8.25
(b) Fossil Fuel	-	-	-

Example 5 Colorado Sunworks System, Longmont, Co.

System Description: This is a passive solar energy system, used for both space heating and DHW preheating, in a 3 bedroom, single story house with about 1800 sq. ft. of living space. The passive space-heating system is a combination of a direct gain system with 300 sq. ft. of glazing, and a drumwall system with 54 water-filled drums (each 55 gallon). There is also additional storage in the 6 inch thick concrete floor and the exterior masonry walls. Heat loss through the glazing is reduced by using moveable beadwall insulation. The heat dist tion is by convection and radiation. Heated a s moved to the north side of the house by a thermosiphon flow through a plenum. The building uses several energy conserving features, including proper insulation, earth berming, an entry vestibule as an airlock and a garage on the northwest as a windbreak. Back up heating is provided by either natural-gas-fired hydronic baseboard units or a wood burning fireplace. The passive solar DHW system consists of two 30-gallon uninsulated tanks, painted black and with moveable insulation in front of them. Summer overheat protection is provided by overhangs, moveable insulation, and provision for cooling by night time ventilation, enhanced by the use of wind turbines.

Table 5 Quantitative Information and Performance Factors for the Colorado Sunworks System
(All Energy Values are in MBTU = A Million BTU)

	Dec. 1978	Jan. 1979	March 1979
Monthly Incident Solar Energy	14.9	13.9	10.5
Monthly Collected Solar Energy	8.8	7.9	4.77
Monthly Space Heating Load	8.25	7.35	4.4
Average Daytime Ambient Temperature	30°F	28°F	49°F
Average 24-Hour Ambient Temperature	20°F	16°F	39°F
24-Hour Ambient Temperature Swing (Max. - Min.)	41°F	35°F	24°F
24-Hour Average Building Interior Temperature	71°F	66°F	70°F
Storage Average Temperature	75°F	70°F	74°
Monthly Solar Energy Used for Space Heating	8.25	7.35	4.37
Operating Energy Used for Space Heating	--	--	--
Auxiliary Energy Used for Space Heating	0.07	0	0
Energy Losses in Space Heating Subsystem	--	--	--
Solar Fraction for Space Heating	100%	100%	100%
Monthly DHW Load	2.2	2.14	1.66
Daily Hot Water Usage (Gallon/Day)	95	91	73
Average Hot Water Temperature Delivered	141	136	131
Operating Energy Used in DHW Subsystem	--	--	--
Monthly Solar Energy Used for DHW	0.56	0.55	0.51
Auxiliary Energy Used for DHW	6.2	6.05	4.69
Energy Losses in DHW Subsystem	--	--	--
Solar Fraction for DHW	21%	21%	25%
Monthly Total Load (Heating + DHW)	10.5	9.5	6.0
Monthly Solar Energy Used by Systen	8.8	7.9	4.87
Monthly Operating Energy for System	0.175	0.18	0.121
Monthly Auxiliary Energy Used by System	6.3	6.05	4.69
Monthly Energy Losses in System	--	--	--
Solar Fraction for System	83%	82%	79%
Overall Solar Conversion Efficiency	0.59	0.57	0.42
Solar System C.O.P.	--	--	--
Monthly Savings of Conventional Energy:			
(a) Electric	--	--	-0.12
(b) Fossil Fuel	17.5	12.3	8.1

Discussion: On the whole, the system performance has been very satisfactory. During the winter of 1978 - 1979 the passive solar heating system satisfied almost 100% of the space-heating demand. It also satisfied 25% of the hot water demand.

Comfort levels inside the building were acceptable to the occupants over most of the winter. During certain months, there were minor difficulties. e.g. In January 1979, the daily average temperature ranged between 61°F and 71°F; however, the occupants did not turn on the auxiliary heating system. The radiant heating from the fireplace helped increase the comfort level on cold days.

An important feature of this house was the awareness of the system operation by the occupants, and their willingness to adjust. For instance in using the solar-preheated water, the timing of the water usage is important, since the water in the uninsulated preheat tanks would cool rapidly unless used soon after being heated by the sun. The occupants adjusted their habits by shifting their hot water usage to the afternoon, which minimized the auxiliary energy used. Similarly, the occupants' willingness to accept slightly cooler temperatures without turning on the auxiliary space heat led to additional energy savings. Difference in the comfort level between the two building zones were generally quite small, showing that the system could effectively transfer heat from the south rooms to the north rooms. Earth berming of most of the north side of the building helped to keep the heat losses small. The beadwall operation ran into difficulty once, but this was corrected.

CONCLUSIONS

In this paper, we have discussed selected HUD Solar Projects that have been relatively trouble-free, and which indicate what helps make a solar system work well. Each of these projects has been examined according to the design criteria listed earlier, and has been found to be satisfactory according to most of them. Most of the systems have had problems, but it is now clear how they can be resolved. In a few cases, the problems have led to a lower system performance which is, however, still acceptable. Undersizing of the solar collection area has sometimes led to a small solar fraction. Since this is largely decided by economics, it has not been considered a serious defect, provided the system has otherwise worked well. Among the instrumented HUD solar projects, the air-based solar systems have shown up more problems in operation; however, it is now clear what needs to be done to avoid them.

SUMMARY REVIEW: SOLAR COOLING SYSTEMS

Leonard Hall
Carrier Corporation

The integration of components into a solar assist cooling system, regardless of capacity, approaches the complexity of larger commercial air conditioning systems. Beeause of this, a system must be engineered with the understanding of the interaction of each individual component on the total system. This is not unlike the earlier years of electric air conditioning when systems were built up on site using separate components. With the advent of packaged condensing sections with matched evaporator coils and tubing kits, the engineering requirements almost completely disappeared. Today, there are few bad electric cooling applications that are directly attributed to the installation of the basic hardware.

Since solar cooling has not reached the packaged equipment maturity level as yet, we need to learn the pitfalls which have caused difficulties. Likewise, we must learn from the successful jobs and take advantage of the willingness of others to share their knowledge.

Although each subsystem is important, the single most cost influenced element is the solar collector. Not only is the collector the primary energy source but if the field is oversized, it has a major influence on system cost. The temperature level which must be provided for presently known cooling equipment is relatively high and not all collector designs can work efficiently at these temperatures. However, in making the final selection single point efficiency is not the only criterion to consider. What does it cost per BTU for the entire cooling season which must include the number of hours per time period this temperature level energy can be supplied. Secondly, what pump power is required for the fluid flow required to maintain collector efficiency and equipment performance. Another power factor to overcome pressure drop attributed to fluid viscosity has to be evaluated.

Each passing season has seen more cooling installations and hence the collector manufacturers have gained additional experience. These manufacturers are not only interested in selling their goods, but recognize the influence poor jobs have on future usage of their collectors. Assistance not only in sizing but piping recommendations are normally available for the asking.

Most heat operated cooling machines have internal controllers which operate independent of external system controls. It is most important to understand this control logic and how to interrelate it with the external logic. Using the example cited about condenser water low temperature which resulted in poor equipment COP has a relatively simple connection. The use of two condenser water pumps, in parallel, of different sizes operated off entering condenser water temperature will maintain the temperature above minimum. An important additional benefit is the reduction in pump power because a relatively high percentage of the time the wet bulb temperature is well below design. This operating condition is more significant on commercial or industrial loads which have high internal load factors which result in operation when outdoor ambients are relatively low.

There are two installations operating using the packaged concept which are presently under evaluation. On a single base is the cooling machine; pumps for solar hot water, condenser water and cooling load; controller valves; interconnectiong piping; system centralized control box. Built into the controls are the interaction with the auxiliary energy source. On one site the auxiliary energy is waste process steam. The steam to water heat exchanger and condensate sump and pump have been included in the package. The other system uses an air to water heat pump. Theoretically, all you need to do is connect the insulated pipes to the other subsystems. This concept is designed to eliminate much of the field labor. It also allows in-house inspection of much of the entire system.

Both packages were shipped on standard over-the-road trucks and delivered successfully to the job sites. Setting into place caused no difficulties. Start-up was relatively straightforward with minimal problems. Sufficient performance data have not been obtained as yet to make positive observations.

A successful installation does not only relate to thermal performance but also to long term maintenance free performance. Certain types of valves are fine when used where maintenance is always available such as in an industrial plant, but are not advisable for unattended systems. Specialized pumps should be avoided because of availability of service parts locally.

If there is one goal that should be the objective for each installation, it should be "simplicity". The simpler the design normally the less costly the installation and the greatest chance for success.

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Program

SECOND ANNUAL
SOLAR HEATING AND COOLING SYSTEMS
OPERATIONAL RESULTS CONFERENCE
November 27-30, 1979
BROADMOOR Hotel
Colorado Springs, Colorado

November 27, 1979

6:00 - 9:00 p.m.	Registration	Mezzanine Lounge - Main

November 28, 1979

7:00 - 8:30 a.m.	Registration	International Center
	SESSION I	International Center
	OVERVIEW	
	Dan S. Ward, Chairman Colorado State University	
8:30 - 8:45 a.m.	Conference Introduction Welcome: The What and Why of the Conference	Dan S. Ward, Conference Chairman and Research Associate Professor, Colorado State University
8:45 - 9:30 a.m.	Keynote Speaker	Dr. Fred Morse, Director, Office of Solar Applications, Office of Assistant Secretary of Conservation and Solar Energy, Department of Energy
9:30 - 10:00 a.m.	Audio-Visual Presentation "The Solar Revolution"	H. Jackson Hale, Acting Chief, Active Building Systems Branch, Office of Assistant Secretary of Conservation and Solar Energy, Department of Energy
10:00 - 10:30 a.m.	BREAK	
	SESSION II	International Center
	SOLAR DOMESTIC HOT WATER "DHW" SYSTEMS	
	Jack Meeker, Chairman Northeast Solar Energy Center	
10:30 - 11:00 a.m.	Thermal Performance of DHW Solar Energy Systems in the National Solar Data Network During the 1978-1979 Heating Season	Jonathan M. Nash IBM Corporation
11:00 - 11:15 a.m.	The Performance of a Daystar HW2/F-B Solar Water Heater in Southern New Jersey	Harold E. Taylor Stockton State College

11:15 - 11:30 a.m.	Thermal Performance of a Solar Heated Hot Water System	Harry A. Rockefeller Oral Roberts University
11:30 - 11:50 a.m.	Performance of Six Solar Domestic Hot Water Systems in the Mid-Atlantic Region	Robert Dikkers National Bureau of Standards
11:50 - 1:15 p.m.	LUNCHEON	International Center
1:30 - 1:45 p.m.	The Northeast States Solar Water Heater Performance Experience	Paul Erhartic Northeast Solar Energy Center
1:45 - 2:00 p.m.	Preliminary Results - Solar Domestic Hot Water Monitoring in Pennsylvania	William K Aungst Pennsylvania State University
2:00 - 2:15 p.m.	Performance Data on Solar-Assisted Hot-Water Systems in Five Tucson Residences	Robert B. Kinney University of Arizona
2:15 - 2:30 p.m.	Design and Operation of Solar DHW Systems	N. B. Ostrye General Development & Supply
2:30 - 2:45 p.m.	System Description and Early Performance of Pool and Domestic Water Heating Installation, Willmar State Hospital, Willmar Minnesota	Paul F. Cummings State of Minnesota
2:45 - 3:00 p.m.	Operational Results for a Commercial Office Building Solar DHW System in Burlington, Massachusetts	John Zvara Aerospace
3:00 - 3:30 p.m.	BREAK	
3:30 - 4:00 p.m.	Collector Array Performance of Instrumented Sites in the National Solar Data Network	William H. McCumber IBM Corporation
4:00 - 4:15 p.m.	Performance of a Solar Hot Water System Installed in an Interstate Highway Visitor Center	David R. Jackson University of Connecticut
4:15 - 4:30 p.m.	Solar Water Heater Demonstration	Gerald R. Guinn Alabama Solar Energy Center
4:30 - 5:00 p.m.	Summary Review: Solar DHW Heating Systems	Jack Meeker Northeast Solar Energy Center
5:30 - 7:30 p.m.	Time is available for authors to set up their poster displays	Mezzanine Main

November 29, 1979 — SESSION III — International Center

SOLAR SPACE HEATING SYSTEMS

Session III A - Passive Systems
J. Douglas Balcomb, Chairman
Los Alamos Scientific Lab

8:30 - 8:50 a.m.	Results of Performance Analysis of Passive Systems in the National Solar Data Network	Michael W. Weston IBM Corporation
8:50 - 9:10 a.m.	Performance Data Evaluation of the Balcomb Solar Home	J. D. Balcomb Los Alamos Scientific Lab
9:10 - 9:25 a.m.	The Solar Classroom at Hamilton College	J. W. Ring Hamilton College (NY)
9:25 - 9:40 a.m.	Results from a Passive Thermal Storage Roof on a Mobile/Modular Home in Los Alamos	J. C. Hedstrom Los Alamos Scientific Lab
9:40 - 10:00 a.m.	"Did/Didn't--Should'a/Shouldn't'a" Recommended Architectural and Engineering Practices based on Experiences from Operational Solar Energy Systems	S. Weinstein Ehrenkrantz Group
10:00 - 10:30 a.m.	BREAK	

Session III B - Active Systems
Robert G. Mills, Chairman
The Mills Company

10:30 - 11:00 a.m.	Thermal Performances of Solar Energy Space Heating Subsystems	R. S. Skidmore IBM Corporation
11:00 - 11:20 a.m.	Thermal Performance of a Commercially Available State-of-the-Art Solar Air-Heating System	S. Karaki Colorado State University
11:20 - 11:35 a.m.	Air Leakage Effects on Active Air-Heating Solar Collector System Performance	Dennis E. Jones Solaron
11:35 - 11:50 a.m.	The Performance of Solar Energy Systems in the Residential Solar Demonstration Program	William E. Freeborne Housing and Urban Development
12:00 - 1:15 p.m.	LUNCHEON	International Center

1:30 - 3:00 p.m.		Session III - Poster Session	Main Mezzanine (3 rooms)
	1.	Solar Heating Results for the Nambe Community Center	H. S. Murray Los Alamos Scientific Lat
	2.	Thermal Characteristics of a Rock Storage Unit for Solar Energy	Russell B. Brinsfield University of Maryland
	3.	Performance of the SunEarth House	Pamm McFadden Colorado Sunworks
	4.	Hail Resistance of Solar Collectors with Tempered Glass Covers	G. O. G. Löf Colorado State University
	5.	Energy Usage of an Energy Efficient, Solar Heated House	Albert Kent Southern Illinois University
	6.	Combined Solar Thermal/Photovoltaic System - Cooling Season	Ghazi Darkazalli University of Texas (Arlington)
	7.	Solar Economics	Vern J. Hellenbrand Research Products Corporation
	8.	Heat Extraction from a Large, Salt-Gradient Solar Pond	L. J. Wittenberg Monsanto Research Corporation
	9.	Lumsdaine Solar House	Monika Lumsdaine New Mexico State University
	10.	A Low Cost Retrofit Solar Home for Demonstration and Consumer Acceptance Experiments	D. Y. Goswami North Carolina A&T State Univer
	11.	Economics of Heating Maryland Broiler Houses with Solar Energy	Jarvis L. Cain University of Maryland
	12.	The Use of Operational Results to Identify Potential Improvements in the Thermal Performance of Air Solar Heating Systems and to Establish Performance Criteria	David E. Cassel Mueller Associates
	13.	Overview of Field Experience with Three Active Solar Commercial Buildings	C. W. Rouzer Bickle/CM Inc.
	14.	Performance of a Space Heating System	E. M. Swindle Aircraftsman
	15.	Passive Energy Design and Habitability Aspects of Earth Sheltered Housing in Oklahoma	Lester L. Boyer Oklahoma State University
	16.	Solar Analysis Based on Utility Bills	H. R. Sparkes Dubin-Bloome Associates
	17.	Problems and Questions in Solar Analysis Methodology	K. Raman Dubin-Bloome Associates

3:00 - 3:30 p.m.	BREAK	
	Session III (Continued)	International Center
3:30 - 3:45 p.m.	Performance of the Solar-Assisted HVAC System for Terraset Elementary School with Passive Earth Cover	Robert L. Whitelaw Virginia Polytechnic Institute and State University
3:45 - 4:10 p.m.	Cost Effectiveness - An Assessment Based on Commercial Demonstration Projects	Thomas A. King Mueller Associates
4:10 - 4:35 p.m.	Summary Review: Solar Active Space Heating Systems	Robert G. Mills Mills Company
4:35 - 5:00 p.m.	Summary Review: Solar Passive Space Heating Systems	J. Douglas Balcomb Los Alamos Scientific Lab
6:00 - 7:30 p.m.	Reception (Cash Bar)	Mezzanine - Main

November 30, 1979 — SESSION IV — International Center

SOLAR COOLING SYSTEMS

Leonard Hall, Chairman
Carrier Corporation

8:30 - 9:00 a.m.	Thermal Performance of Space Cooling Solar Energy Systems in the National Solar Data Network	C. T. Wallace IBM Corporation
9:00 - 9:20 a.m.	Solar Heating and Cooling Results for the Los Alamos Study Center	J. C. Hedstrom Los Alamos Scientific Lab
9:20 - 9:40 a.m.	Performance of the CSU Solar House III Heating and Cooling System	H. S. Oberoi Colorado State University
9:40 - 10:00 a.m.	Results of Improvements in the Decade 80 Solar House Cooling System	Russell K. Johnson Copper Development Association
10:00 - 10:30 a.m.	BREAK	
10:30 - 10:50 a.m.	Performance and Controls in Solar Heating Systems	Tom Kent Solar Controls
10:50 - 11:05 a.m.	Operational Improvements in the CSU Solar House I System Supplied with Heat from Evacuated Collectors	W. S. Duff Colorado State University
11:05 - 11:20 a.m.	Solar Systems that Work Well	K. Raman Dubin-Bloome Associates
11:__ - 11:50 a.m.	Summary Review: Solar Cooling Systems	Leonard Hall Carrier Corporation
12:00 - 1:15 p.m.	LUNCHEON	International Center

1:30 - 2:15 p.m.	Department of Energy Solar Technology Programs	Maurice Katz, Deputy Director, Distributed Solar Technology Division, Department of rgy
2:15 - 2:30 p.m.	Conference Summary and Closing Remarks	Dan S. Ward Colorado State University

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