DESIGN CONSIDERATIONS FOR RESIDENTIAL SOLAR HEATING AND COOLING SYSTEMS UTILIZING EVACUATED TUBE SOLAR COLLECTORS

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ABSTRACT

As solar heating systems become a commercial reality, greater efforts are now being employed to incorporate solar cooling components in order to obtain a complete solar heating and cooling system, and thus take advantage of the cost-effectiveness of year-round use of the solar equipment. The solar heating and cooling system design presented in this paper incorporates design considerations which have been obtained from previous experimental efforts utilizing evacuated tube solar collectors. These advanced collectors are capable of significantly higher efficiencies, even at the higher temperatures required for solar cooling operation. Most of the considerations presented here are based on the experience gained in the design and performance of the solar heating and cooling systems for CSU Solar Houses I through IV.

INTRODUCTION

The addition of a solar cooling capability to a solar space and domestic hot water (DHW) heating system allows for a substantially improved usage of the solar collector array and the associated components of the solar system. Year-round usage gives greater cost-effectiveness of the solar equipment because of greater cost savings on conventional fuels in return for a relatively small additional initial capital investment. An integrated solar heating and cooling system (including DHW heating) is expected to achieve a much lower cost per unit energy than a solar space and DHW heating system in most areas of the continental United States. [1] There is, however, the possibility of technical complications which must be resolved before any cost advantages can be realized. A simple example is the inability (without significant and costly complications) to provide for an optimum tilt of the collector for both summer and winter solar angles. (Generally, it is necessary to choose either the winter heating or the summer cooling for optimum tilt of the collector, because a compromise angle between the two alternatives would actually optimize the collector for spring and fall solar radiation.) In addition, there are substantially different operating characteristics of the various systems. Solar cooling would typically provide about 80 percent of the summer cooling load on a daily basis (with auxiliary providing 20 percent each day), whereas heating is usually 100 percent solar on most days (no auxiliary use), and 60 to 80 percent auxiliary on 20 to 30 days during the heating season. [2] In addition, sizing of DHW, space heating, and space cooling systems will vary considerably. (See, for example, reference 3.)

The performance of solar heating and cooling systems is strongly dependent upon the operating characteristics of the various subsystems and, in particular, upon the characteristics of the cooling method. At present, most residential solar cooling systems utilize absorption cooling units (usually lithium-bromide). Other possibilities include: (1) the use of night air cooled to the wet bulb temperature to chill a pebble-bed rock storage unit for daytime use (generally limited to arid and semi-arid regions); (2) radiative-evaporative cooling; (3) solar assisted desiccant dehumidification; (4) Rankine-vapor compression (these are generally only available in very large units); (5) solar assisted heat pumps; (6) solar Rankine heat pumps; (7) heat pump cooling of rock beds at night; and (8) roof water ponds. However, because of the commercial availability of lithium-bromide absorption cooling units, these cooling systems and their operating characteristics will be used as the basis for discussion of incorporating evacuated tube solar collectors into solar heating and cooling systems.

The particular characteristics of the evacuated tube solar collectors also play an important part in overall system design. For example, because of the geometrical shape of evacuated glass tube solar collectors, snow removal can be expected to require longer melting periods than conventional flat-plate solar collectors, because snow slide-off is generally impeded by the tubular design. On the other hand, wind and hail will be much less of a threat to the structural integrity of an evacuated glass tube.

EVACUATED TUBE SOLAR COLLECTORS

Evacuated tube solar collectors permit the use of a vacuum of sufficient magnitude to virtually eliminate convection and conduction heat transfer losses. In addition, these collectors generally require a minimum amount of material per square foot of collector and thus provide for the possibility of lower costs (under conditions of large scale
manufacturing processes). Finally, the vacuum may help to protect a selective surface used on the absorber (for reduction of long-wave radiation heat losses) against performance degradation over the life of the collector.

The performance of these evacuated glass tube solar collectors can be represented (for steady-state conditions) by an equation of the form:

\[ n = n_0 - U \left( \frac{T_f - T_a}{S} \right) \tag{1} \]

where \( n \) is the solar collector efficiency (dimensionless), \( n_0 \) is the solar collector efficiency when \( T_f - T_a = 0 \) (dimensionless), \( U \) is a function of the various heat transfer coefficients applicable to a particular collector (\( \text{kj}/(\text{hr})(\text{m}^2)(\circ C) \)), \( T_f \) is the outlet fluid temperature (\( ^\circ C \)), \( T_a \) is the outdoor air temperature (\( ^\circ C \)), and \( S \) is the solar radiation on the collector (\( \text{kj}/(\text{hr})(\text{m}^2) \)).

For comparison purposes, value of \( n_0 \) and \( U \) are given in Table 1 for several different kinds of solar collectors. Equation (1) can also be used to estimate temperature that may be reached during stagnation conditions (no flow of coolant fluid through collector loop). Under these conditions, \( n = 0 \) and Equation (1) reduces to:

\[ T_f - T_a = \frac{n_0}{U} \tag{2} \]

It will be noted that the values of \( n_0 \) (column 5 of Table 1) are higher and the values of \( U \) (column 6 of Table 1) are lower for the evacuated glass tube solar collectors compared to the other more conventional (concentration = 1) flat-plate solar collectors listed, but their stagnation temperatures are also much higher (column 7 of Table 1). For values of \( (T_f - T_a)/S > 0.033 \) (\( \text{hr})(\text{m}^2)(\circ C)/\text{kj} \), both evacuated glass tube solar collectors outperform concentrating (concentration = 10) solar collectors that track the sun.

The superior performance of evacuated glass tube solar collectors provides for substantially higher efficiencies (as much as double that of conventional flat-plate solar collectors) under the same conditions of high outlet fluid temperatures (\( T_f \)). This is especially important because of the high temperature requirements of solar cooling equipment. The input temperature to the generator of a LiBr absorption unit should be (for cooling water temperatures of 30\( ^\circ C \)) between 80 and 100\( ^\circ C \). For lower cooling water temperatures (24\( ^\circ C \)), the range is 70 to 100\( ^\circ C \). Under these conditions, the Coefficient of Performance (COP) is in the range of 0.65 to 0.70. While even lower generator inlet temperatures can be tolerated for absorption units equipped with an internal pump, the resulting decrease in COP makes lower operating temperatures infeasible.

The temperatures required to effectively operate absorption cooling units emphasize the necessity of using high performance collectors, capable of high efficiencies even under conditions of high collector fluid temperatures and high temperatures in thermal storage. For example, if the minimum realistic input temperature to the generator of the absorption machine is 80\( ^\circ C \), the thermal storage temperature must exceed 80\( ^\circ C \). If a heat exchanger is located between the collector and storage, then the collector is operating at minimum temperatures of 85 to 90\( ^\circ C \). For ambient temperatures of 30\( ^\circ C \), and good solar conditions of 900 \( \text{W/m}^2 \), a typical flat-plate collector may achieve peak efficiency of only 40 percent, whereas an evacuated tube solar collector could operate at a peak efficiency of 60 percent. For daily efficiencies the evacuated tube can be expected to collect twice as much useful energy as the flat-plate collector. This is due, in part, to the fact that the minimum intensity of solar radiation necessary for the collection of useful heat is lower for an evacuated tube solar collectors, and thus for a given system and identical climatic conditions, the evacuated glass tube solar collectors will have longer daily operating intervals (i.e., they begin collecting useful energy earlier in the day and continue longer in the afternoon before ceasing operation.)

The present major disadvantage of evacuated tubular collectors is the high cost (ranging from 300 to 500 \$/m\(^2\)). However, the material costs are only

<table>
<thead>
<tr>
<th>Manufacturer (1)</th>
<th>Type of Solar Collector (2)</th>
<th>Fluid (3)</th>
<th>Concentration (4)</th>
<th>( n_0 ) (5)</th>
<th>( kJ/(hr)(m^2)(°C) ) (6)</th>
<th>( T_f - T_a, °C ) for ( n_0 = 0^* ) (7)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Northrup</td>
<td>Fresnel Lens (tracking mode)</td>
<td>Liquid</td>
<td>10</td>
<td>0.845</td>
<td>6.16</td>
<td>466</td>
</tr>
<tr>
<td>General Electric</td>
<td>Fresnel Lens (non-tracking mode)</td>
<td>Liquid</td>
<td>4</td>
<td>0.859</td>
<td>9.67</td>
<td>302</td>
</tr>
<tr>
<td>CSU (Solar House I)</td>
<td>Observed ** Selective surface</td>
<td>Liquid</td>
<td>1</td>
<td>0.751</td>
<td>13.10</td>
<td>194</td>
</tr>
<tr>
<td>Solaron (Solar Houses II and IV)</td>
<td>Theoretical</td>
<td>Liquid</td>
<td>1</td>
<td>0.66</td>
<td>18.3</td>
<td>122</td>
</tr>
<tr>
<td>Amtek</td>
<td>Selective surface, 2 glass cover plates</td>
<td>Liquid</td>
<td>1</td>
<td>0.74</td>
<td>12.9</td>
<td>208</td>
</tr>
<tr>
<td>Corning (Solar House I)</td>
<td>Evacuated glass tube</td>
<td>Liquid</td>
<td>1</td>
<td>0.74</td>
<td>12.9</td>
<td>208</td>
</tr>
<tr>
<td>Owens-Illinois (Solar House III)</td>
<td>Evacuated glass tube</td>
<td>Liquid</td>
<td>1</td>
<td>0.791</td>
<td>3.65</td>
<td>350</td>
</tr>
</tbody>
</table>

*Calculated using equation (2) assuming that \( S = 3.406 \text{kJ/(hr)(m}^2) \) = 946 \text{W/m}^2

**Based on an experimental test module whose sides were not insulated
about $30 to $40/m², so that on a mass production basis, it is anticipated that the costs could be reduced to about $150 to $200/m², which is comparable to the costs of conventional flat-plate solar collectors.

One evacuated glass tube solar collector design also has a significant heat capacity (cw, kJ/(°C)(m²)). For a weight per unit area (W) of 19.6 kg/m² when dry and 34.2 kg/m² when filled with water [specific heats (c) of water and glass are, respectively, 4.18 and 0.836 kJ/(kg)(°C)], the value of cw for the Owens-Illinois collector is 77.6 kJ/(°C)(m²).

We may note the effects of a large heat capacity by the following. Let Q represent the heat delivered by the collector in kJ/(m²)(hr). Then:

\[ Q = n_0 - U \left( \frac{T_f - T_a}{S} \right) - cW \left( \frac{\Delta T_f}{\Delta t} \right) \]  

(3)

where \( \Delta T_f \) is the change in collector outlet fluid temperature during the time interval \( \Delta t \) (°C), and \( \Delta t \) is the time interval (hr).

For the Owens-Illinois collector, equation (3) becomes:

\[ Q = 0.791 - 3.59 \left( \frac{T_f - T_a}{S} \right) - 77.6 \left( \frac{\Delta T_f}{\Delta t} \right) \]  

(4)

There are several variations in the design of evacuated tube solar collectors; in addition to the Corning and Owens-Illinois collectors, both General Electric and Philips (Germany) have done considerable research into particular designs. While the differences in design are significant, there are also several noteworthy similarities. All utilize a liquid (usually water or water-ethylene glycol) as a coolant fluid, with the inlet and outlet of the liquid at the same end of the tube (the latter aspect simplifies the problem of differences in the thermal expansion of the outer glass tube and the absorber surface). In addition, numerous tubes (6 to 24 per module) are piped in series to ensure sufficient flow through each collector tube (modules are usually in parallel). This provides for the possibility of substantial pressure drop through the collector modules and, consequently, significant pumping power.

For example, the pressure drop through the General Electric vacuum tube solar collector will be between 0.35 and 0.70 kJ/cm² at 82°C. For the Owens-Illinois collector, the pressure drop ranges from about 0.07 kg/cm² at a flow rate of 0.38 t/min up to 0.88 kg/cm² at a flow rate of 1.5 t/min. Of course, some pressure drop is necessary in the collector module if uniform flow distribution is to be maintained with practical size headers. On the other hand, the viscous contribution to the total pressure drop is almost always substantial in solar collector modules and, consequently, use of a liquid other than water (such as an ethylene glycol aqueous solution) with significantly higher viscosity may greatly increase the pressure drops per collector module.

In one case a 1.5 horsepower pump was required for an evacuated tube solar collector array, supplying a residential-size solar heating and cooling system. While both collection and cooling equipment were in operation, power requirements totaled 53.5 amps at 115 volts, or 6.152 kilowatts. A conventional vapor compression machine (3-ton unit) might require only 4 to 5 kilowatts!

In general, arranging the collector modules in parallel flow patterns (rather than series flow patterns) will reduce the total pressure drop through the collector array. However, these parallel flow patterns do require careful design if uniform flow distribution is to be achieved. Experience indicates that attempting to obtain uniform flow distribution by trial-and-error adjustments of either valves or dampers is essentially impossible in solar collector arrays and, consequently, uniform flow distribution must be designed into the array from the outset.

Other difficulties associated with the use of water and water-glycol in evacuated tube solar collectors are the potential for freezing and boiling. The freezing problem is relatively easy to avoid by using a liquid with a sufficiently low freezing point. If an ethylene glycol water mixture is used sufficient ethylene glycol must be added to permit flow under all conditions. It is possible to use lower concentrations of ethylene glycol than that specified by the manufacturer to prevent damage to the collector, but flow will not be possible. This, however, leads to the extremely dangerous situation where the liquid in the collector reaches the boiling point, but the lines leading to the collector are still frozen sufficiently to prevent flow. Experience dictates that solar collector systems should not be used in a freezing climate with any liquid whose freezing point is above any air temperatures that might be encountered.

While the use of ethylene glycol or other non-freezing liquid may solve the freezing problem, the potential for power failures over the life of the system virtually assure numerous opportunities for boil-off of the collector liquid. In many cases the violent boiling conditions may damage or destroy the solar collector (thermal shock, excessive vibration of the absorber within the glass tube, etc.), but in all cases the liquid must be replaced. If provisions are not made to capture the boiling liquid, this reoccurring cost penalty can be an additional disadvantage of liquid-heating evacuated tube solar collectors.

Finally, the stagnation temperatures of almost any reasonably well designed solar collector will exceed the boiling point of water and water-ethylene glycol mixtures. In addition, the stagnation temperatures of evacuated glass tube solar collectors (see column 7 of Table 1) will exceed the boiling points of almost any reasonable heat transfer liquid, so provision for boiling must be provided, regardless of the heat transfer liquid used. One of the potential disadvantages of evacuated tube solar collectors is that they may not be able to withstand the high stagnation temperatures they
The use of non-aqueous liquids (by eliminating corrosion problems) permits the use of materials considerably less expensive than copper (even copper is not immune to corrosion in an aqueous environment). But since the only practical liquid for heat storage is water, the use of non-aqueous liquids requires a heat exchanger between the collector liquid circuit and the hot water storage tank, a factor which reduces the overall solar system efficiency. However, many non-aqueous liquids have boiling points well above those of water or water-ethylene glycol mixtures. This is an important consideration in the case of evacuated tube solar collectors because of the possibility of unusually high temperatures. The disadvantages of using a conventional heat exchanger (with some temperature difference across the exchanger, e.g., 5-10°C), can be offset by using a direct contact, liquid-liquid heat exchanger (DCLLHE). The DCLLHE uses a liquid immiscible with water as the collector liquid and is circulated through the hot water storage tank, providing the same contact that would be obtained if water were used in the collector loop.

An obvious conclusion to the foregoing is the advantages associated with using air as the collector fluid. Freezing, boiling, and corrosion difficulties are eliminated. In addition, maintenance costs of air type solar systems are shown to be of the order of one percent per year of the initial installed cost. [4] The annual maintenance costs of liquid type solar systems are not likely to be any lower and, in fact, boil-off, annual replacement of antifreeze, corrosion, etc., may indicate higher maintenance costs. Failure to perform any required maintenance may result in a solar system performance decline substantially greater than one percent per year.

To avoid the difficulties of excessive pressure drops and subsequent high blower power requirements, air heating evacuated tube solar collectors may require substantial modifications in design. One possibility is the use of a straight line flow pattern where the inlet and outlet ducts of the collector are at opposite ends of the collector tube. The problem of differential thermal expansion between the absorber and glass tube could be accounted for by the use of a flexible duct portion of the absorber. To maintain a vacuum, the flexible portion would have to be a metal section, sufficiently thin-walled to flex easily and without allowing significant forces to be applied to the metal-glass seal at each end of the collector tube.

**SOLAR HEATING AND COOLING SYSTEM DESIGN (LIQUID SYSTEM)**

Figure 1 is a design schematic for an integrated solar heating and cooling system. It incorporates evacuated tubular solar collectors and a liquid-liquid, direct contact heat exchanger as the thermal storage unit. In addition, the system comprises an auxiliary boiler for providing conventional heat to the heating coils or to the absorption chiller, heating coils, and cooling subsystem (see below), automatically actuated 3-way valves, and a DHW preheating system. The various operating modes of the system are shown in Table 2.

<table>
<thead>
<tr>
<th>Auto Valves</th>
<th>Coll Pump</th>
<th>Circ Pump</th>
<th>DHW Pump</th>
<th>Fan</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Solar collector storage heat</td>
<td>--</td>
<td>--</td>
<td>On</td>
<td>--</td>
</tr>
<tr>
<td>2. Space heating from storage</td>
<td>A</td>
<td>A</td>
<td>A</td>
<td>--</td>
</tr>
<tr>
<td>3. Space heating from auxiliary</td>
<td>B</td>
<td>A</td>
<td>A</td>
<td>--</td>
</tr>
<tr>
<td>4. DHW heating from storage</td>
<td>--</td>
<td>--</td>
<td>B</td>
<td>A</td>
</tr>
<tr>
<td>5. Storage to chiller</td>
<td>A</td>
<td>B</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td>6. Auxiliary to chiller</td>
<td>B</td>
<td>B</td>
<td>A</td>
<td>B</td>
</tr>
</tbody>
</table>

*May provide DHW heating in addition to space heating and cooling in these modes

Part of the design rationale of the system shown in Figure 1 is the minimizing of the number of pumps, e.g., the use of a single pump to operate the chiller and at the same time provide heat to the DHW system. Note that the higher temperature water is delivered to the chiller (e.g., 85°C) and a slightly lower temperature then reaches the DHW double-walled heat exchanger (~80°C). The utilization of numerous automatically actuated 3-way valves can be easily incorporated into solid state control schemes and constitute less of a capital investment than a multitude of pumps.

**COOLING SUBSYSTEM DESIGN**

The design of the cooling subsystem must consider not only the operating characteristics of the absorption cooling unit, but the other components of the solar system as well. For example, if the thermal storage unit is located within the conditioned space of the building, then any heat losses from the storage will add to the cooling load and thus degrade the performance of the total system. Heat losses from a liquid thermal storage unit can be very substantial and can, in fact, mean the difference between a successful and inadequate solar cooling design. Not only do heat losses from storage add to the cooling load, but they also detract from the ability of the solar storage to provide heat at sufficiently high temperatures to operate the cooling unit. While the additional cooling load due to these heat losses can be eliminated by relocating the thermal storage unit exterior to the conditioned space, such losses still reduce the availability of solar heat to operate the absorption unit. Another difficulty of an exterior installation is the possibility of freezing during the winter heating season, the fact that heat losses to the ambient will be higher because of higher temperature differences, and the heat losses will not be able to contribute to the building’s heating load. Because of these considerations, the choice of interior or exterior installations can profoundly affect the overall performance of the solar heating and cooling system.

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The ability of the thermal storage to provide heat to operate the absorption cooling unit is, as mentioned previously, heavily dependent on temperature. As an example, an inlet temperature to the generator of the absorption unit of 88°C may provide for a COP of 0.6 (for cooling water temperatures of 30°C). If the inlet temperature is reduced to 80°C, the COP may be reduced to 0.5. On the other hand, if the inlet temperature is increased to 96°C, the COP will again be reduced because there is no additional cooling with additional input heat to the generator. In this latter case, the COP might be only 0.45.

This example emphasizes the importance of providing design conditions for the operation of the absorption cooling unit. Departures from design conditions can severely penalize the performance of the unit.

Such penalties can also occur in the inlet temperatures of the fluid to be cooled, i.e., the building air (or in cases of an absorption water chiller, the cool water). For a typical lithium bromide absorption water chiller, the inlet temperature of the water to be chilled is 14°C and the outlet temperature is 9°C, yielding a COP of 0.55. However, if the inlet temperature is 11°C, the COP drops to 0.53 (see reference 3 for more details).

A cooling subsystem design that incorporates these considerations is shown in Figure 2. This design utilizes two identically-sized water storage tanks (designated as "warm storage" and "cool storage"), with the total volume of water in the cooling subsystem equal to the volume of one tank. Each of the storage tanks acts as a variable level tank. Operation of the chiller lowers the water level in warm storage and raises the amount of water in cool storage. During the delivery of cool water to the cooling coils (load), the reverse occurs. When both the chiller is operating and cool water is delivered to the cooling load, the levels in the storage units remain constant.

The use of two variable-level cool storage tanks insures the all-important temperature stratification between the inlet and outlet sides of the water chiller. In addition, the chiller can be run in a continuous fashion and avoid cycling of the absorption unit under conditions of intermittent load. Finally, the proper use of automatically actuated 3-way valves allows the use of a single pump for the chiller and cooling load loops. The position of these valves and status of the cooling subsystem pumps are detailed in Table 3.

<table>
<thead>
<tr>
<th>Mode</th>
<th>3-way valve positions</th>
<th>Pumps</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Chiller operating, storing cool</td>
<td>A B A</td>
<td>On On</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Chiller operating, water to cooling coils</td>
<td>A A A</td>
<td>On On</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cool storage providing water to cooling coils</td>
<td>B A B</td>
<td>On Off</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

It is noteworthy that a cool storage will undergo some heat gains from the ambient and that, if the cool storage units are located within the conditioned space, this will constitute a heat removal method and contribute toward meeting the cooling load. In this respect it is similar to the concept of heat losses from a thermal hot storage unit reducing the winter heating load. In order to take advantage of these two aspects, an alternative storage system could be comprised of three water storage tanks. The first tank would be located exterior to the conditioned space and would be twice the volume of the other two, identically-sized tanks, both of which would be located within the conditioned space. During the winter heating season, the exterior tank would be empty and the two interior tanks would provide a stratified thermal storage unit for the solar heating system. During the summer cooling season, the exterior tank would be used for the thermal hot storage subsystem and the interior tanks would be used in the cooling subsystem (as described in Figure 2). Thus winter heat losses of the interior tanks would aid in meeting the heating load and summer heat gains aid in meeting the cooling load. The heat losses of the exterior tank in summer would be less (due to a lower temperature difference, i.e., higher ambient temperature) and would not add to the cooling load.

CONCLUSIONS

Evacuated tube solar collectors have the same inherent problems associated with liquid-heating solar collectors: Freezing, boiling, corrosion, etc. However, for the evacuated collectors, the problems are usually more severe; it is painfully easy to cause boiling in an evacuated tube collector which utilizes aqueous collector fluids. It would appear that an air-heating evacuated tube solar collector design could be of substantially greater value. And with the utilization of pebble-bed storage at higher temperature levels (e.g., 60 to 90°C instead of 20 to 60°C), could perform better than liquid systems. And at the higher temperature levels, absorption chiller units could utilize solar heated air via a heat exchanger.

It will, of course, be necessary to ensure that an air-heating evacuated tube collector not incorporate the large pressure drops associated with present designs. But by using straight line flow patterns, this problem can be overcome. An example would be a double-walled pyrex tube with vacuum between the concentric tubes (the coefficient of expansion of pyrex is low enough to prevent damage due to the differential thermal expansion of the two concentric tubes).

REFERENCES


Figure 1. Solar Heating and Cooling System Design Schematic

Figure 2. Cooling Subsystem Design Schematic
TWENTY MONTHS OF OPERATING EXPERIENCE WITH A SOLAR HEATED AND COOLED OFFICE BUILDING

Dr. Robert L. San Martin
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The New Mexico Department of Agriculture building has been in continuous operation since October 1975. This large solar heated and cooled building has had extensive reports in public circulation which have documented the system design, system studies, initial cost, installation and initial start-up, and even the lessons learned during its first few months of operation. An instrumentation* system was developed and installed to monitor the operation of the solar and mechanical system and sub-systems. It has been in continuous operation, logging historical data on the building, since November 1976.

Several small but important changes have been made to the building's solar system which are improving the performance.

* - This project has been supported by the State of New Mexico, Energy Research and Development Program and the Solar Energy Division of the Energy Research and Development Administration.

This sophisticated system has had anomalies observed with its piping, collectors, controls and even sub-component operation consideration. They will be analyzed and explained in this paper.

The historical data which has been collected allows for a detailed analysis of the building's performance and the isolation of specific areas which have appeared as sources of anomalies. Over one dozen of these have been observed and they are documented as information for others who will undertake projects of this type and magnitude.

The system has been found to operate very well within the limits that were anticipated in its design. The solar system has been capable of providing almost one hundred percent of the building's heating requirement and over sixty percent of the cooling requirements. Operating expenses in addition to thermal performance of this system are documented.