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ABSTRACT

Results and conclusions to date of a program to design, erect, and test a 5,000-square-foot solar energy system are presented in this report. The program described demonstrates the ability of solar collectors to supplement the heating and hot water requirements of North View Junior High School in suburban Minneapolis. The report discusses in detail the collector design, system design, system operation, and system performance. The basic rationale for the program is the necessity of obtaining firm answers in three areas: (1) validation of system performance, (2) determination of overall system costs, and (3) acquisition of data to determine the benefits of such a system. The program is obtaining engineering data that may be used to improve collector performance and system performance or design. In addition, data are being compiled that may be used to define investment requirements for similar installations. The program is also helping to determine community acceptance of solar heated school buildings. Testing at the site is continuing on a day-by-day basis to obtain additional data on system performance and benefits. (Author/MLF)

NSF-RA-N-7

SOLAR HEATING PROOF-OF-CONCEPT EXPERIMENT FOR A PUBLIC SCHOOL BUILDING

Prepared for:

NATIONAL SCIENCE FOUNDATION

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Office of Public Technology Projects

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SOLAR HEATING PROOF-OF-CONCEPT EXPERIMENT FOR A PUBLIC SCHOOL BUILDING

Sponsored by:
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SECTION I
INTRODUCTION

The application of solar energy for heating a building is becoming increasingly more attractive as fuel prices increase and the supplies diminish. During cold weather, buildings in the Northern states which are heated by gas on a demand basis must shift to hard-to-obtain alternate fuels or pay a high gas rate. These buildings are ideal candidates for supplemental heat supplied by solar powered systems.

The program described in this report demonstrates the ability of solar collectors to supplement the heating and hot water requirements of North View Junior High School in suburban Minneapolis. The program is obtaining engineering data which may be used to improve collector performance and system performance or design. In addition, data are being compiled which may be used to define investment requirements for similar installations. The program is also helping to determine community acceptance of solar heated school buildings.

The basic rationale for the program is the necessity of obtaining firm answers in three areas: 1) validation of system performance, 2) determination of overall system costs, and 3) acquisition of data to determine the benefits of such a system.

Only by actually constructing a solar heating system and full scale testing can good engineering data be obtained to compare with analytical predictions. Further, testing the system in conjunction with its actual use in a functioning building will provide real-time inputs not envisioned in simulation or other artificial test environments. The testing location provides a long, severe heating season with a wide variation of insolation and other environmental factors.

Fabrication and installation of a solar collector system and its associated heating system will provide improved cost and technical information on construction techniques and installation problems not encountered in fabricating a few units used only for engineering test purposes. Even more important is the ability to define the costs and problems associated with the operation and maintenance of the system. In addition, the nonfinancial costs of space use and impact on building and community esthetics can be determined.

The most comprehensive and important benefit of the program is the experience obtained in actually heating and operating a large-scale facility and in obtaining good engineering data. Financially the school will benefit in fuel savings principally in the winter from space heating and in the summer from domestic water heating and swimming pool water heating.

In January 1974 the Systems and Research Center of Honeywell Inc., under contract to the National Science Foundation, embarked on a program to design, erect, and test a 5000-square-foot solar-energy system. Results and conclusions to date are presented in this report. The report discusses in detail the collector design, system design, system operation, and system performance. Testing at the site is continuing on a day-by-day basis to obtain additional data on system performance and benefits.

SECTION II
SITE SELECTION

The solar energy system is located in Minnesota Independent School District No. 279 at North View Junior High School, 5869 69th Avenue North, Brooklyn Center, Minnesota, a suburb of Minneapolis. The building is a modern educational facility of about 165,000 square feet of floor area with swimming facilities, gymnasium, cafeteria, and an array of various classrooms.

The building statistics for North View Junior High are:

• Total building area	165,583 square feet
• Total building volume	2,326,140 square feet
• Total awarded prime and equipment contract	$3,114,120.00
• Cost per square foot	$18.80
• Site area	Approximately 30 acres
• Construction date	1968-69
• Architects	Matson and Wegleitner
• Student capacity	Approximately 1300

Present heating system used in the building is a low-pressure steam system fired by two gas-fired (with oil standby) combination burners in the mechanical equipment room on the first floor in the southeast corner of the building. The ventilating and the heating of the building is handled primarily through a central ventilating system housed in the equipment rooms--one located above the two-story classroom wing and one located between the gymnasium and the pool on the roof of the locker room.

The mechanical equipment located on the roof of the two-story classroom wing houses two central ventilating units through which all the outside air moves via duct work in the ceilings of the first and second floors of the building. Two large ventilating units in the mechanical room on the roof have steam coils fed from the steam-fired boilers to temper the outside air to 60 degrees. Reheat coils thermostatically operated at the point of entry into the various spaces, such as the classrooms, provides the final comfortable temperature for these spaces.

The equipment located above the locker room consists of heating and ventilation units for the pool, gymnasium, stage area and several smaller athletic rooms. These units are similar to the classroom units and provide tempering of outside air and a thermostatically operated final temperature modulating coil.

The heating and ventilating control system is pneumatic with the controls operating steam valves and/or outside air dampers.

The school consumes an average of 21,000 million BTU's annually with usage covering the summer months. In addition, the facility is used as a community recreation area with emphasis on swimming pool and gymnasium use during off-school hours. This large consumption of energy in a building with such diverse usage enables a solar collector system to provide almost year-around usage.

Factors which favored selecting the North View site as the preferred site are as follows:

1) The heating/cooling demand was large enough to require several thousand square feet of collector. The facility usage was diverse enough to use hot water during the summer, thus giving almost year-round use of the solar power.

2) The daily load for the school reasonably matches the availability of solar flux; it, therefore, requires a minimum of heat storage capacity.

3) Siting is relatively easy because of the abundance of open areas (ball diamonds, etc.) which assure unobstructed solar input. The North View site by placing the collectors in full view of a major freeway provides high visibility and is easily found by visitors.

4) A school system has strong educational and socio/political perspectives which are seldom found in other institutions.

5) A public school system should be an acceptable institution to be the recipient of a benefit from a government-funded program.

6) The school is a modern energy-conserving building but nevertheless, a large energy user. It demonstrates a critical need with high visibility to the community.

7) The proposed system does not block school views and is not attached to the building; therefore, it is not subject to restrictive building codes and the system can also be enlarged if desired.

8) The school system is expected to be relatively independent and unbiased in assessing system performance.

9) The relative similarity of schools and their usage provides a commonality such that data collected would have broad application.

Minnesota was chosen as a good proving ground because:

1) It is representative of a low solar insolation particularly during the heating season.

2) The characteristically clear skies accompanying cold snaps promotes the use of solar heating.

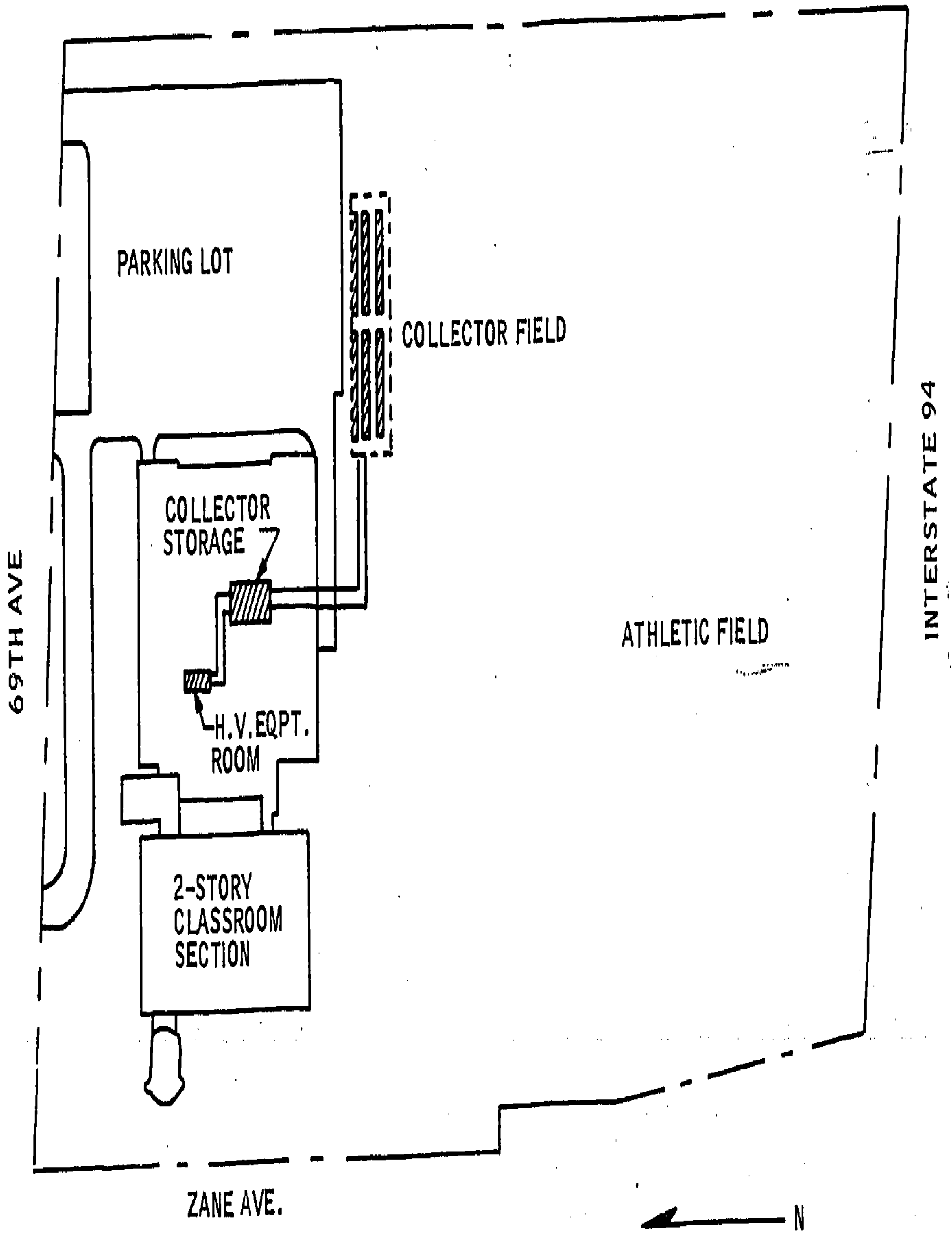
3) The heating season is unusually long extending into the spring; therefore, it provides adequate test time.

4) Fuel costs on a yearly basis are high in Minnesota.

5) Minnesota has a wide annual variation in temperature (-30 to +95°F) and many daily variations of 40 to 50°F, thus providing a wide range of technical data.

School personnel are exceptionally cooperative and enthusiastic about a solar collector system from an energy saving standpoint as well as the educational benefits of such a system. Energy conserving measures are a common practice among the custodians in this school district.

Integrating the solar system components with the existing heating and ventilating system was a reasonable design goal. Ample space was available for storage and piping and heat exchangers were easily installed. Figure 2-1 is a drawing of the overall school area.



NORTHVIEW JUNIOR HIGH SCHOOL

Figure 2-1. North View Junior High School Site

SELECTION OF SOLAR HEATING AREAS

Upon finalization of the contract, Honeywell personnel along with representatives of Michaud, Cooley, Hallberg, Erickson and Associates, the consulting engineering firm who had designed the existing heating and ventilating system, visited Northview Junior High School for a preliminary survey of the school's heating and ventilation system.

Three possible locations for air heat exchangers were found. The first was the heating and ventilation units for the two-story classroom wing which consisted of two units moving 36,000 cfm of air each with a minimum of 25 percent outside air. The second was the gymnasium unit which handles 20,000 cfm with a minimum of 25 percent outside air. The third was the pool unit which handles 12,000 cfm with the outside air intake controlled by a humidistat with no less than 25 percent fresh air.

The classroom units were attractive from a load standpoint supplying 9000 cfm of fresh air each, but necessitated running about 300 feet of pipe to these units outside of the building. This would have required penetrating the building exterior three times and running pipe vertically up an outside wall. The gymnasium unit was complicated by a zone-controlled output and mechanical problems in installing heat exchangers. The pool unit provided the best choice because of ease of installation and the fact that monitoring the fresh air intake over several weeks revealed that the fresh air intake rarely was less than 6000 cfm. In addition, the pool air is heated to a temperature of 80 to 82°F. This represents a load almost equivalent to the classroom heat exchangers (360,000 BTU/hr with 22° outside air). With 100 percent outside air this unit alone would represent a reasonable winter load for the solar system.

In addition, monitoring of hot water usage revealed a daily consumption of 2000 to 3000 gallons. This water enters the building at 50 to 60°F and is heated to 140°F. This represents an energy need of about 1.2 to 2 million BTUs. It was decided to install a water heat exchanger ahead of and in series with the existing water tank to preheat the incoming domestic water.

SECTION III

SYSTEM DESIGN

COLLECTOR SIZING AND DESIGN

The general design of the collectors consists of liquid-cooled steel plates coated with selective absorbing coatings and enclosed in an insulated steel housing which is covered by two transparent windows, one glass and one Tedlar. Figure 3-1 is a corner section of a typical collector. Shown are the glass cover, Tedlar sheet, absorber plate, insulation, and the steel housing with the various structural assemblies.

The outer housing is composed of 18-gauge, cold-rolled steel 36 inches wide, 96 inches long and 6 inches deep. The housing is formed from a single sheet of steel in a press brake and welded at the corners. Internal brackets are welded to the housing for absorber mounting supports. In addition, channels are spot welded to the housing sides for support of the Tedlar sheet. Four access holes are provided for the plumbing and six additional holes are for maintenance and installation.

Hat sections of steel are formed and welded to the back of the housing for stiffening and to provide support for collector suspension on the support structure. Figures 3-2 and 3-3 show the housing design details.

The absorber panel itself is pictured in Figure 3-4. The unit consists of two 25-gauge, 1020 cold-rolled steel sheets 31.75 inches wide by 46.5 inches long with flow passages formed into the sheets. One sheet carries 0.250 inch x 0.050 inch deep grooves on 2-inch centers for basic heat transfer flow. The other sheet is formed to provide a 2.0 inch x 2.0 inch manifold at one end with a 0.50 inch x 0.50 inch manifold at the other end. The two sheets are assembled by spot and seam welding to form a parallel flow panel with large manifolds and small heat transfer channels. Formed steel tubulations on each corner permit connection by automotive type hose of a series of panels to form an array of parallel collectors with small vertical heat transfer passages. The design minimizes the effects of nonuniform flow through the panel assembly. The final design of a complete collector used two absorber panels assembled within a unit for fabrication conveniences. The two panels are connected by small hoses between the smaller manifold tubulations and are attached to mounting brackets by insulating nylon washers. The completed assembly acts very much like a single panel. Appendixes A, B and C outline detailed analyses of heat transfer characteristics, flow distribution and selective coatings for flatplate collectors. Appendix C also includes discussions of circulation and covers.

Additional design features include a frame which is placed over the outside edges of the glass cover which is 3/16 inch x 34-57/64 inches x 95 inches tempered glass. A sealant tape is used on both sides of the cover near the edges. Figure 3-5 shows the frame and glass assembly. Insulation is 3.5 inches of fiberglass across the back of the collector and along the sides to reduce edge losses. Figure 3-5 also shows the Tedlar sheets and frame installed in a typical collector. Figure 3-6 shows a typical assembly step in the manufacture of the collectors. Collector sizing of 5000 square feet of absorber area dictated the fabrication of 246 individual assemblies which were then transported to and erected on the site.

COLLECTOR SUPPORT

The collector support structure was fabricated from square structural steel tubing 3 inches x 3 inches x 3/16 inch. Three rows of support structure 21 feet apart were needed to support the 246 collectors with each row 256 feet long and divided into two sections for a

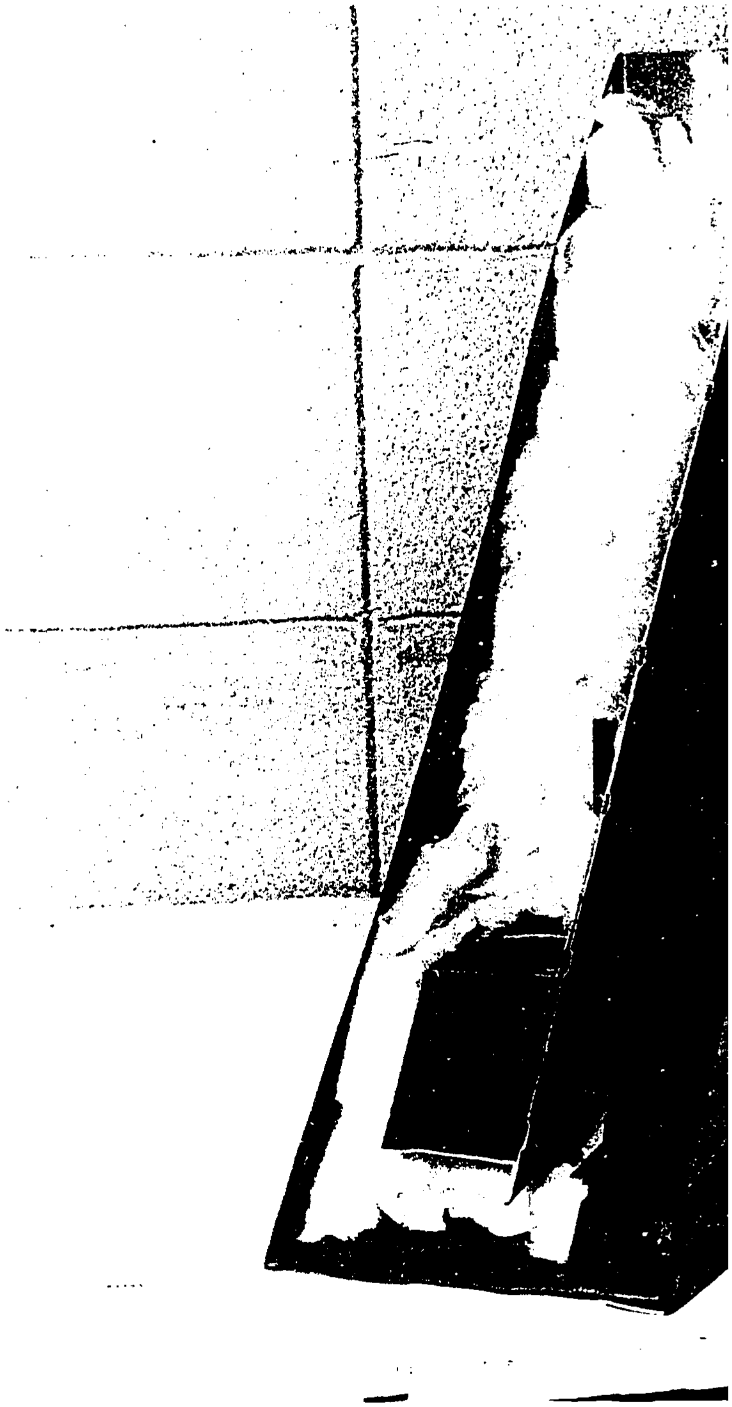


Figure 3-1. Collector Corn

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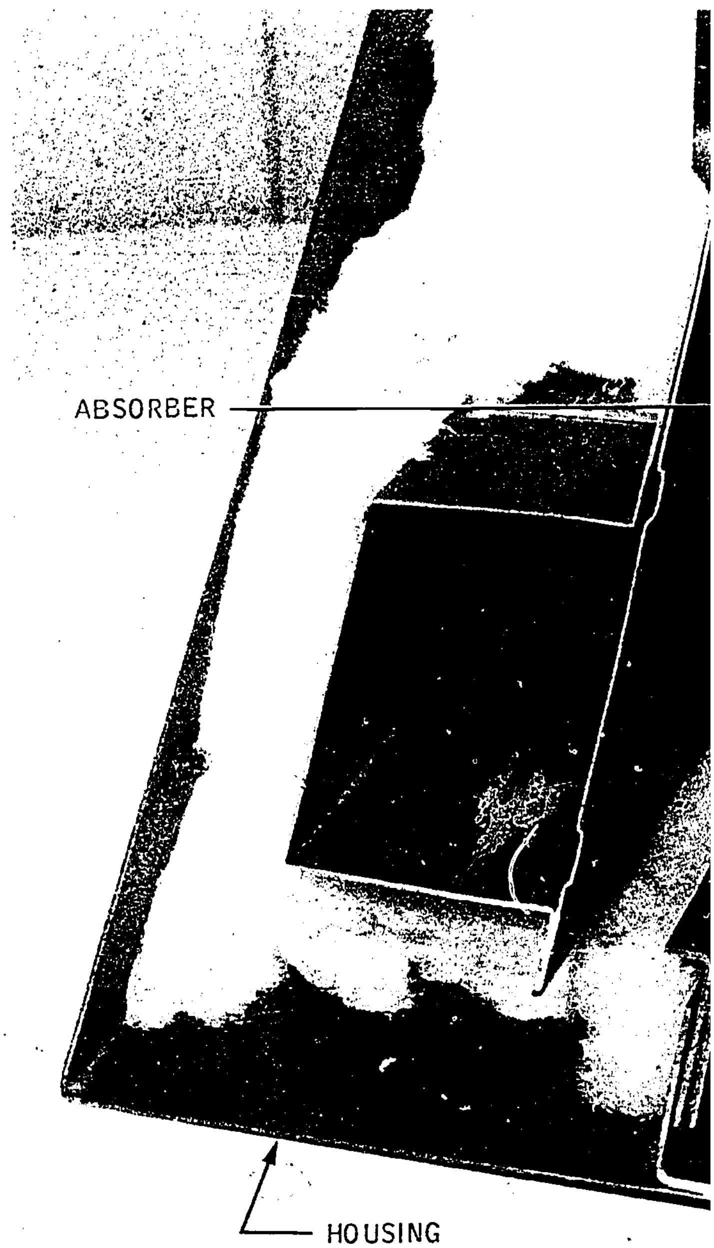

Figure 3-5. Solar Collect

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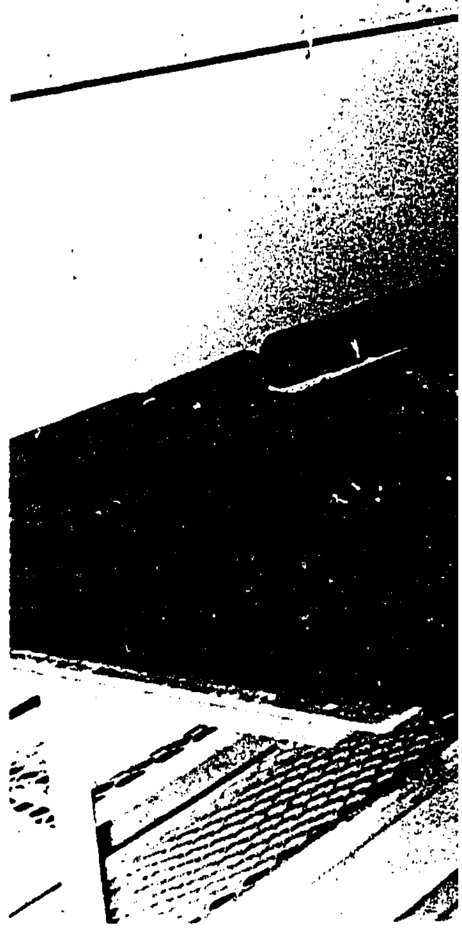
GLASS

DLAR



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total of six sections. Erection of the structure consisted of boring 180 holes on 9-foot centers and then cementing in place the vertical support members. The vertical members were then cut to a 55-degree angle. Longitudinal tubes were then welded to the cut posts forming a pair of rails upon which the collectors could be placed. Additional cross bracings of 1-1/2 x 1-1/2 x 3/16 angles were added to each pair of posts. Figure 3-7 shows a typical cross section of a pair of post supports while Figure 3-8 shows the structure with the panels mounted.

The completed collectors were placed on the rails and moved into position on small rollers. Hoses were connected at the top and bottom manifolds to complete the assembly. The structure itself is designed to withstand 90 mph wind loads with the collectors in place. Figure 3-9 shows the complete solar collector array.

STORAGE TANK

Storage tank sizing was based on storing approximately one day's collection of heat. This was due to the limited amount of space available for storage capacity. Final design size was 3000 gallons and the tank was erected within the building from separate 1/4 inch thick steel sections. Design working pressure is 35 pounds per square inch with burst pressure at 90 pounds per square inch. The tank is supported on a separate concrete slab which was poured to accommodate the additional concentrated load. The tank is insulated with 2 inches of rigid fiberglass. Figure 3-10 shows the storage tank installed in the system.

HEAT EXCHANGERS

As previously mentioned, the system uses heat exchangers located in the pool air circulation loop and in the domestic water feedline. These heat exchangers were sized on the basis of a 10°F rise across the collectors at a flow rate of 80 gpm. The air circulation system has two heat exchangers--one for heating outside air and the other for heating a combination of tempered outside air and pool return air. These heat exchangers are four rows, six fins per inch glycol-water to air with a face area of 42 x 111 inches. A subsequent computer calculation assuming inlet glycol-water at 95°F and outside air at 40°F showed a mixed air temperature of 68°F and a glycol-water return temperature of 85°F with an air flow of 12,000 cfm and glycol-water flow rate at 80 gpm. This corresponds to a heat extraction rate of 360,000 Btu per hour. The domestic water heater is a standard tube and shell heat exchanger and was sized on the basis of a water use which varies from 100 to 700 gph with a maximum outlet temperature of 140°F. Assuming an inlet water temperature of 40°F this exchanger is capable of transferring up to 1,000,000 Btu per hour. Figure 3-11 shows the two air heat exchangers installed in the system while Figure 3-12 is the domestic water heat exchanger.

PIPING AND PUMP SIZING

To minimize pressure losses in the system a pipe size of 3 inches was chosen. Total pipe footage from the collectors through the air heat exchangers back to the collectors is about 1000 feet. Assuming a flow rate of 80 gpm the pressure drop per 100 feet of 3-inch schedule 40 steel pipe is 0.687 psi, or 6.87 psi for the 1000 feet. Each 90-degree elbow in the system is equivalent to the pressure drop in 3 feet of pipe. For approximately 60 elbows this is equivalent to 180 feet of pipe or about 1.25 psi. Five control valves at 0.7 psi/valve contributes a 3.5 psi loss. The two air heat exchangers in series amount to 10 psi loss at 80 gpm. The flow meter contributes 1 psi while the collectors nominally contribute about 0.5 psi. The total calculated loss is about 23 psi. Assuming an 80 gpm flow of 50-50 glycol-water mixture with a specific gravity of about 1.1 would require an ideal pump size of about 1.2 horsepower. A 5 hp. pump was selected which would give a flow rate of 80 gpm at a 32 psi head. This is a centrifugal pump with the capability of

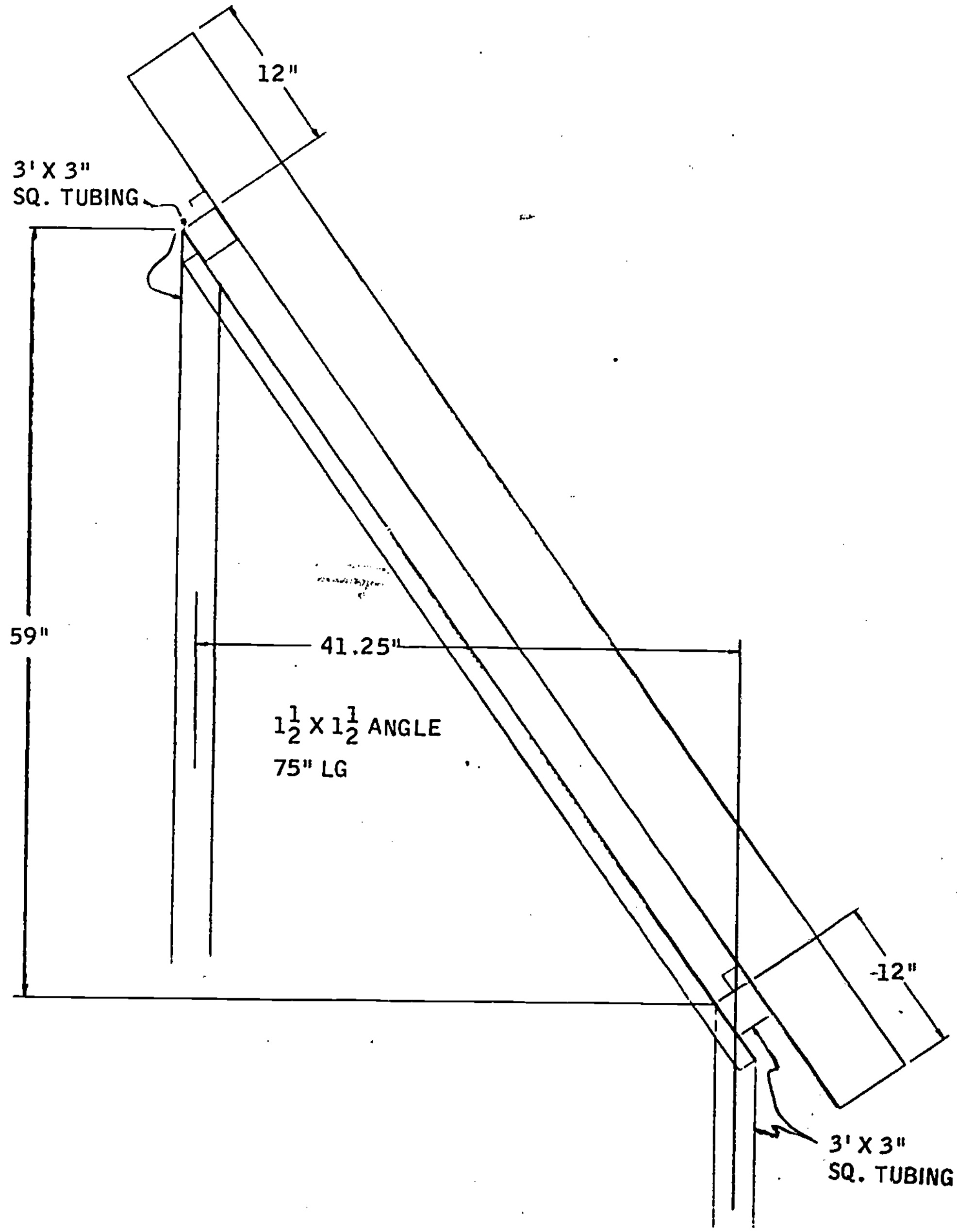

Figure 3-7. Endview Collector Support

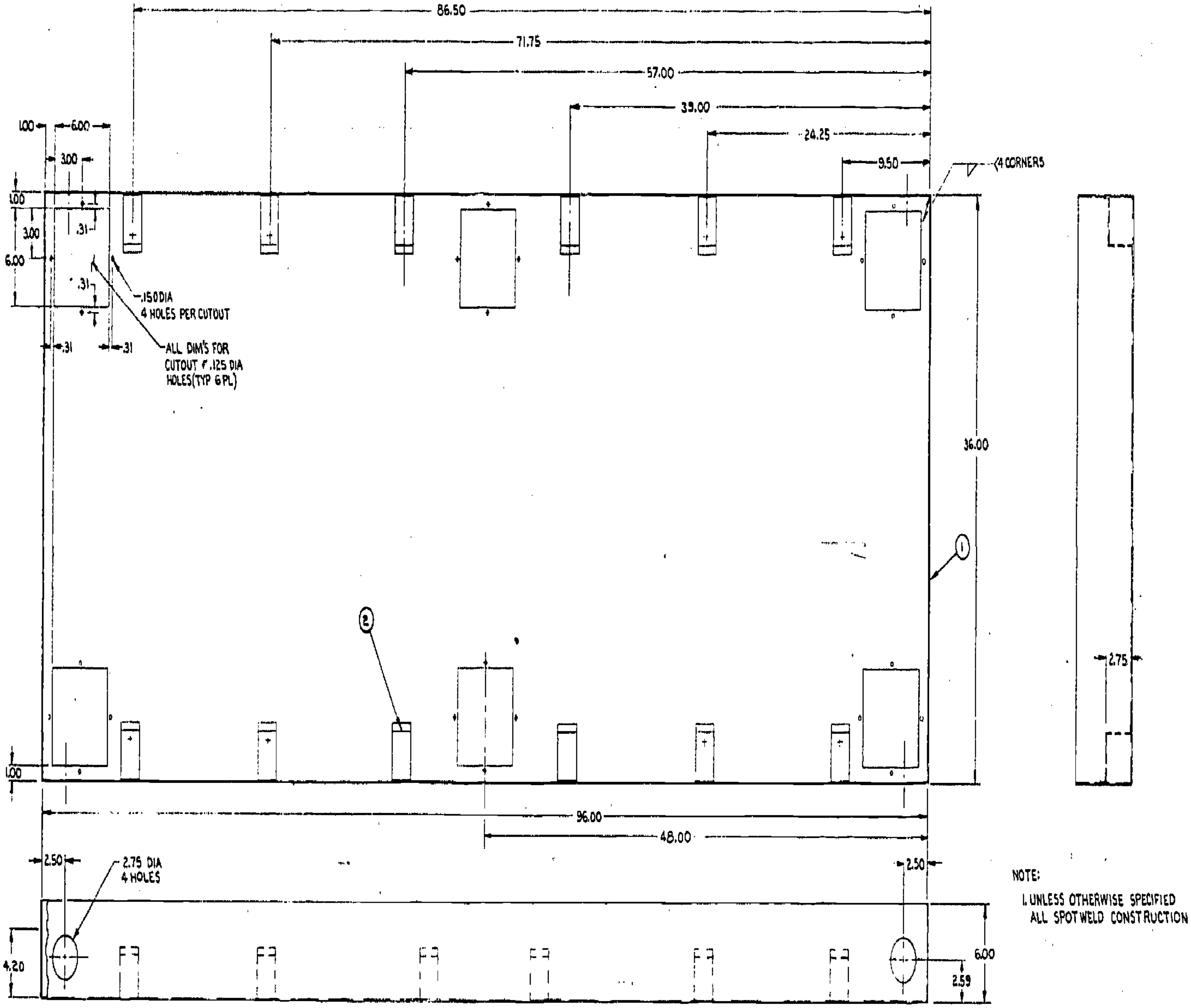

Figure 3-2. Collector Housing Front View

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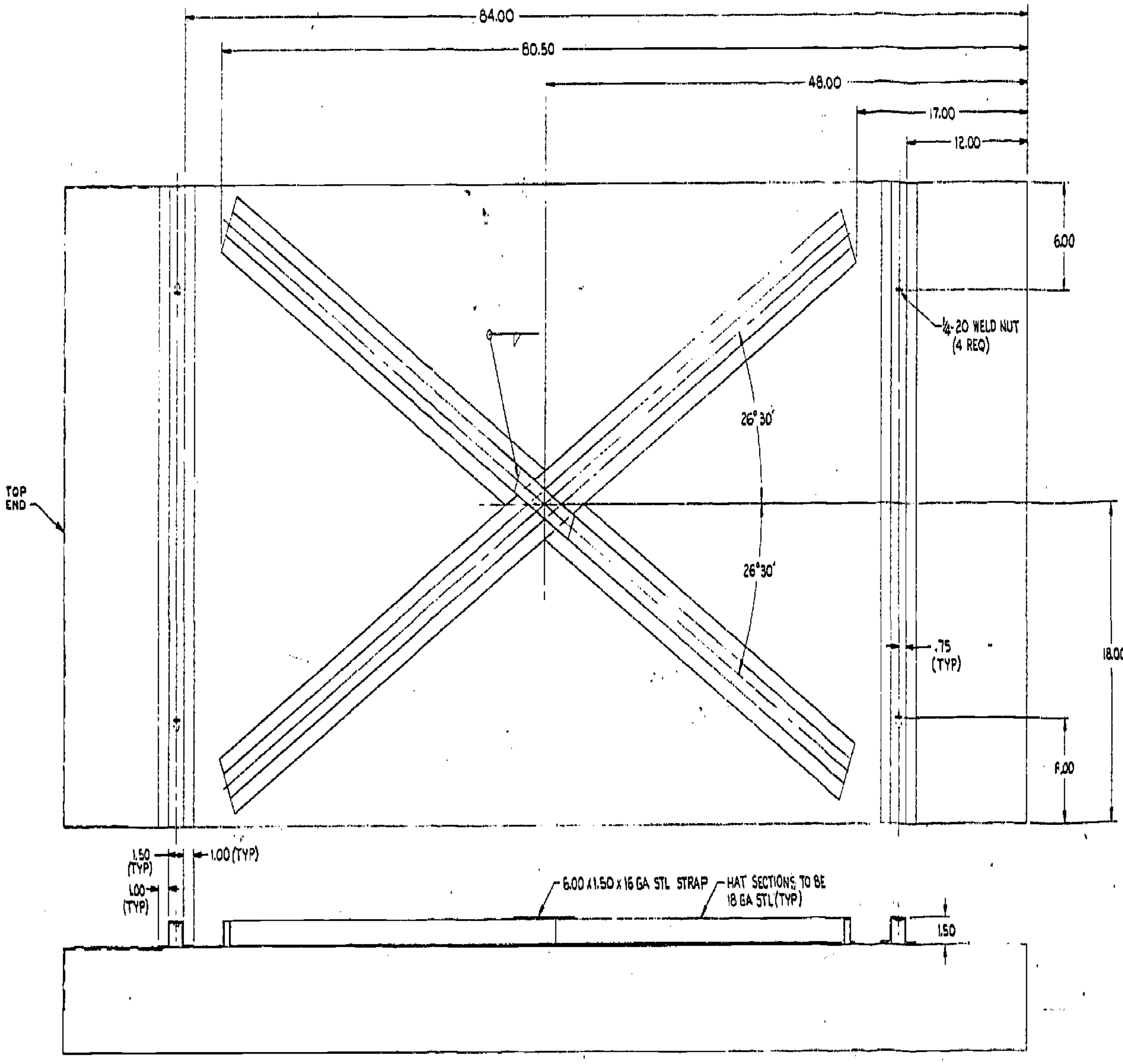

NOTE:
1. UNLESS OTHERWISE SPECIFIED
ALL SPOTWELD CONSTRUCTION

Figure 3-3. Collector Housing Back and Side Views

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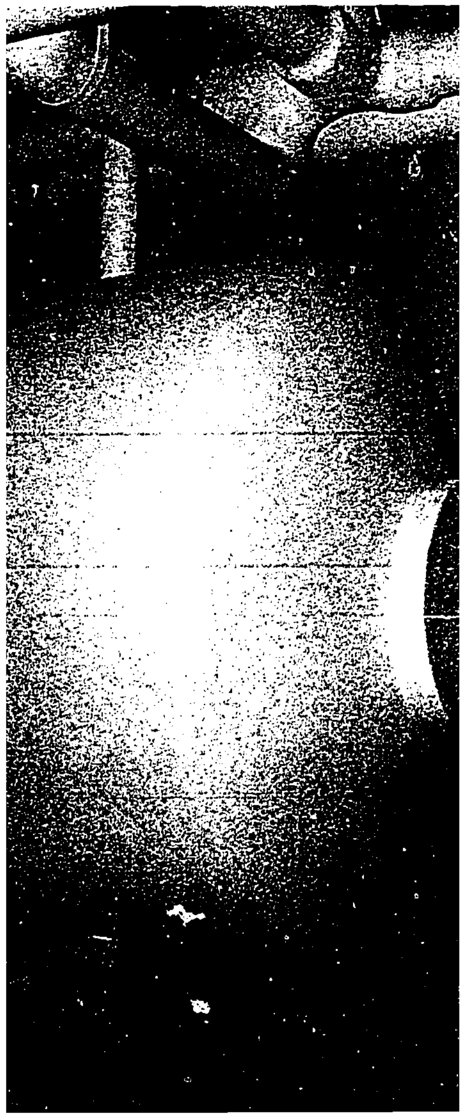
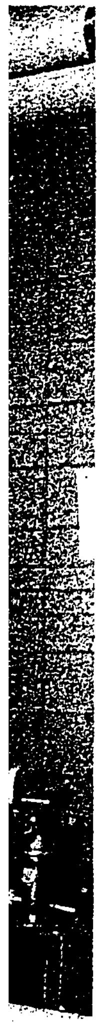
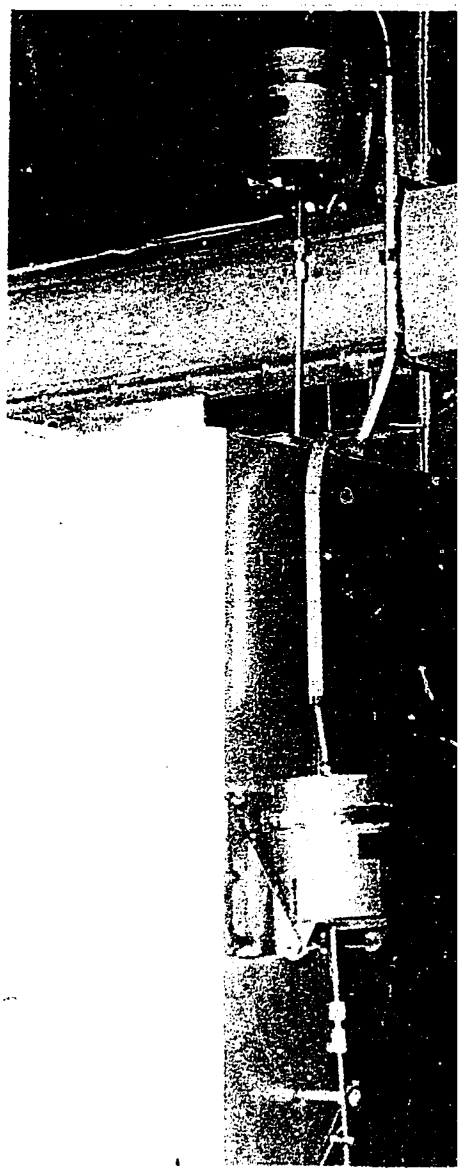
!upport Structure

Figure 3-9. Complete Solar Collector Field



Tank





Figure

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machining the impeller to match the system flow and pressure drops. Subsequent operation of the system shows a nominal pressure drop of 21 psi in the loop. Figure 3-13 illustrates the pump and its location.

SPECIAL FEATURES

To allow for system fluid expansion (estimated at 232 gallons for a 50-50 glycol/water mixture from 40°F to 200°F) four 120-gallon expansion tanks were connected in parallel and installed at the high point in the system. These expansion tanks are connected on the suction side of the pump at the collector common outlet. The system fluid capacities are broken down into 3000 gallons in storage, 312 gallons in the collectors, 400 gallons in the piping and 160 gallons in the expansion tanks. Figure 3-14 shows the four expansion tanks installed near the air heat exchangers.

An air separator was installed on the suction side of the pump. This is a vertical air separator and filter combination with the air leaving at the top center of the unit to an air holding tank located about 10 feet above the separator. Periodic bleeding of air from this holding tank is required. Figures 3-15 and 3-16 show the air separator and holding tank. Additional filters are installed at each collector row inlet and are of the commercial "Y" strainer type.

Balancing valves are installed at each collector row inlet as well as shutoff valves at each outlet. Pressure relief valves are installed immediately after each collector outlet and are set at a nominal 15 to 20 psi range. Figures 3-17 and 3-19 show the installation of these components on the system. Figure 3-17 is the collector inlet balancing valve while Figure 3-18 shows a typical pressure relief valve and shutoff valve.

System insulation other than in the collectors is 2-inch, semi-rigid fiberglass on all interior and above ground exterior pipe. The above ground exterior pipe is also covered by an aluminum protective cover. All underground piping insulation is 2 inches of urethane covered with a waterproof coating. Figure 3-19 shows the above ground pipe with insulation and associated expansion loop.

SYSTEM LAYOUT

Figure 3-20 describes the system layout. As previously described, the solar collectors are divided into six sections with parallel inputs and outputs. The common outputs go (Figure 3-21) into the building to the air separator-filter combination then through the pump, flow control valve, and flow meter towards the storage tank. The flow can be directed into the top of the storage and out the bottom, into the bottom and out the top or bypassed around the tank by appropriately switching Valves 1, 2, and 3. After passing around or through the storage tank, the flow is directed to domestic water heating or to air heating depending upon the position of Valve 5. When directed towards air heating, Valve 4 determines if the return air heating coil will be bypassed. After passing Valve 5, the flow is directed to or around the collectors depending upon the position of Valve 6. The domestic water is directed around or through the heat exchanger depending upon the position of Valve 7. In addition there is a manual bypass installed in the domestic water line in case of a malfunction in the heat exchanger. An additional line from the heat exchanger output to the pool or drain is provided for emergency extraction of heat from the collectors.

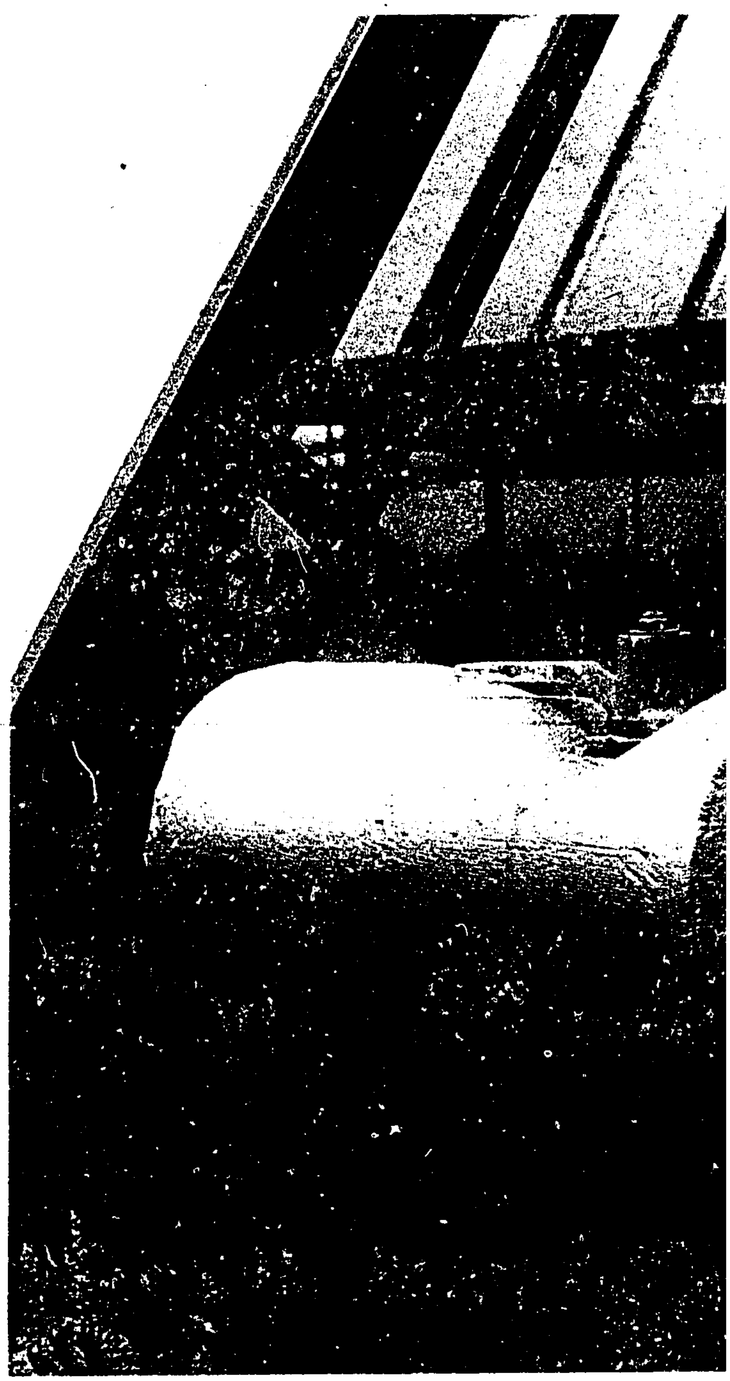

Figure 3-17. Collector Inlet E

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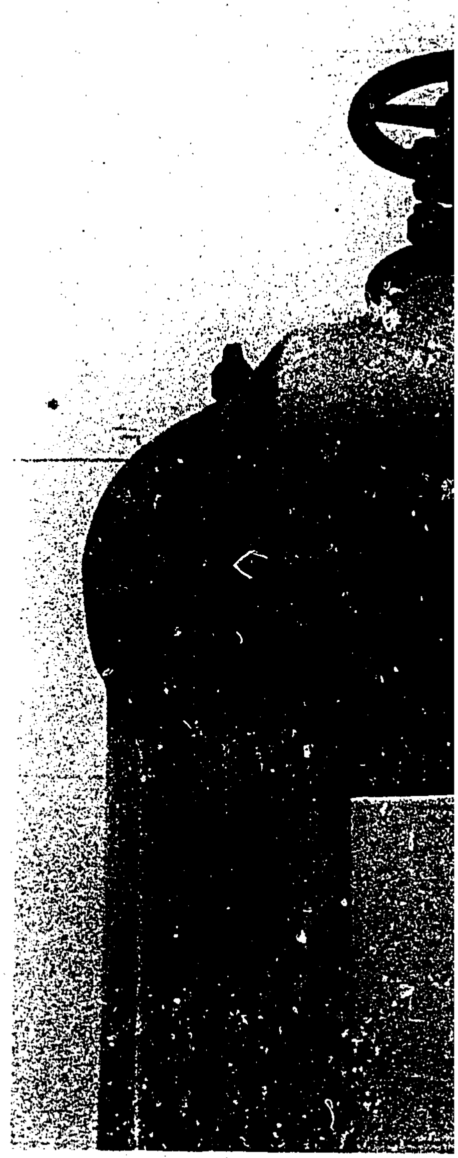

Figure 3-18. Collector

Outlet Shutoff Valve and Pressure Relief Valve

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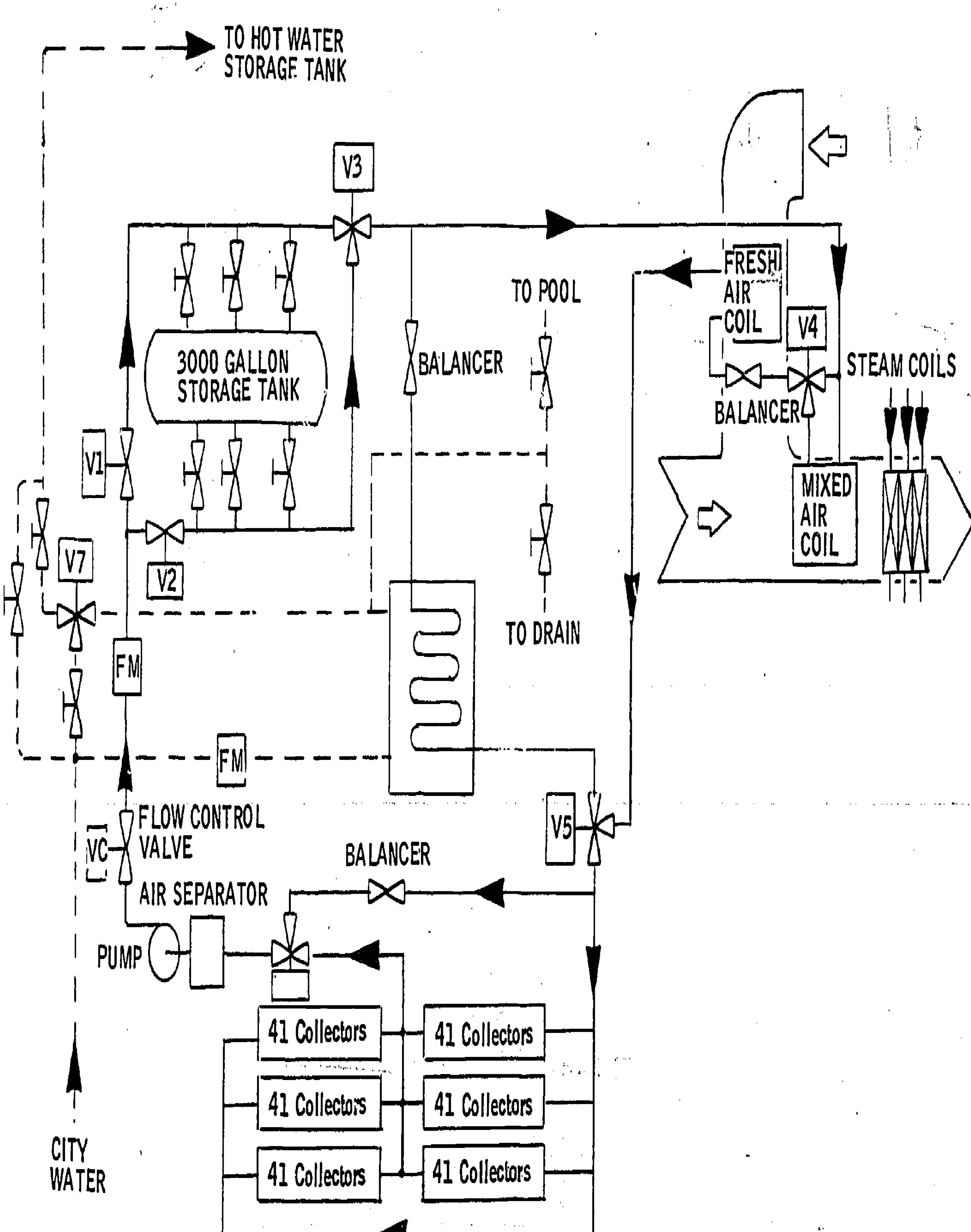


Figure 3-20. System Schematic

Figure 3-21. C

CONTROL SYSTEM

The system incorporates Honeywell pneumatic controllers and valves. Figure 3-22 is a typical three-way modulating valve. The valves are actuated either manually from control switches on the control panel or automatically if the valve switches are set in the automatic mode. Pneumatic temperature readouts are provided on the panel for approximate visual reference to temperatures. The control panel itself is a schematic layout of the plumbing system with valve switches locations corresponding to actual valve locations. Figures 3-23, 3-24, and 3-25 illustrate the control panel layout.

An automatic controller is incorporated into the system and provides for high-temperature protection of the collectors when the system is left unattended or when it is desirable to collect energy in a limited way during weekends or periods of low energy consumption. Figure 3-26 shows the automatic temperature controller installed on the panel.

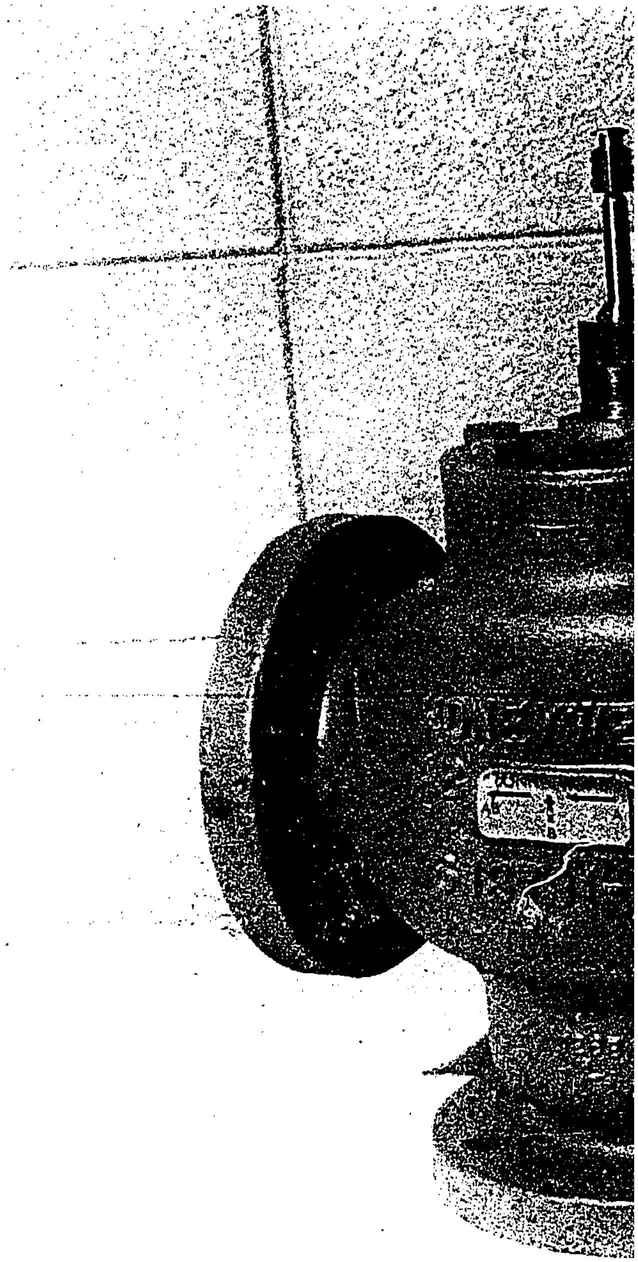
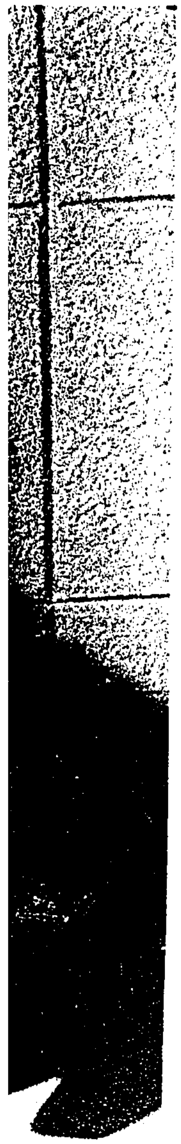
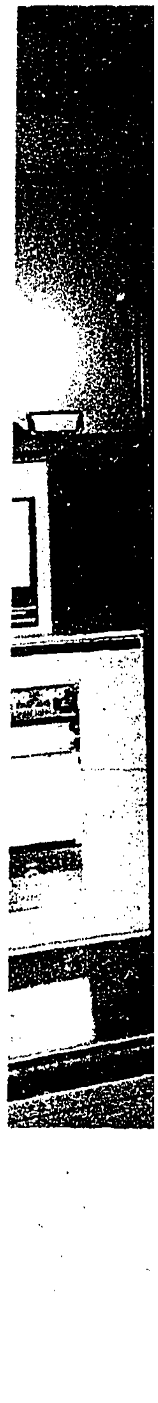


Figure 3-22. Three-W





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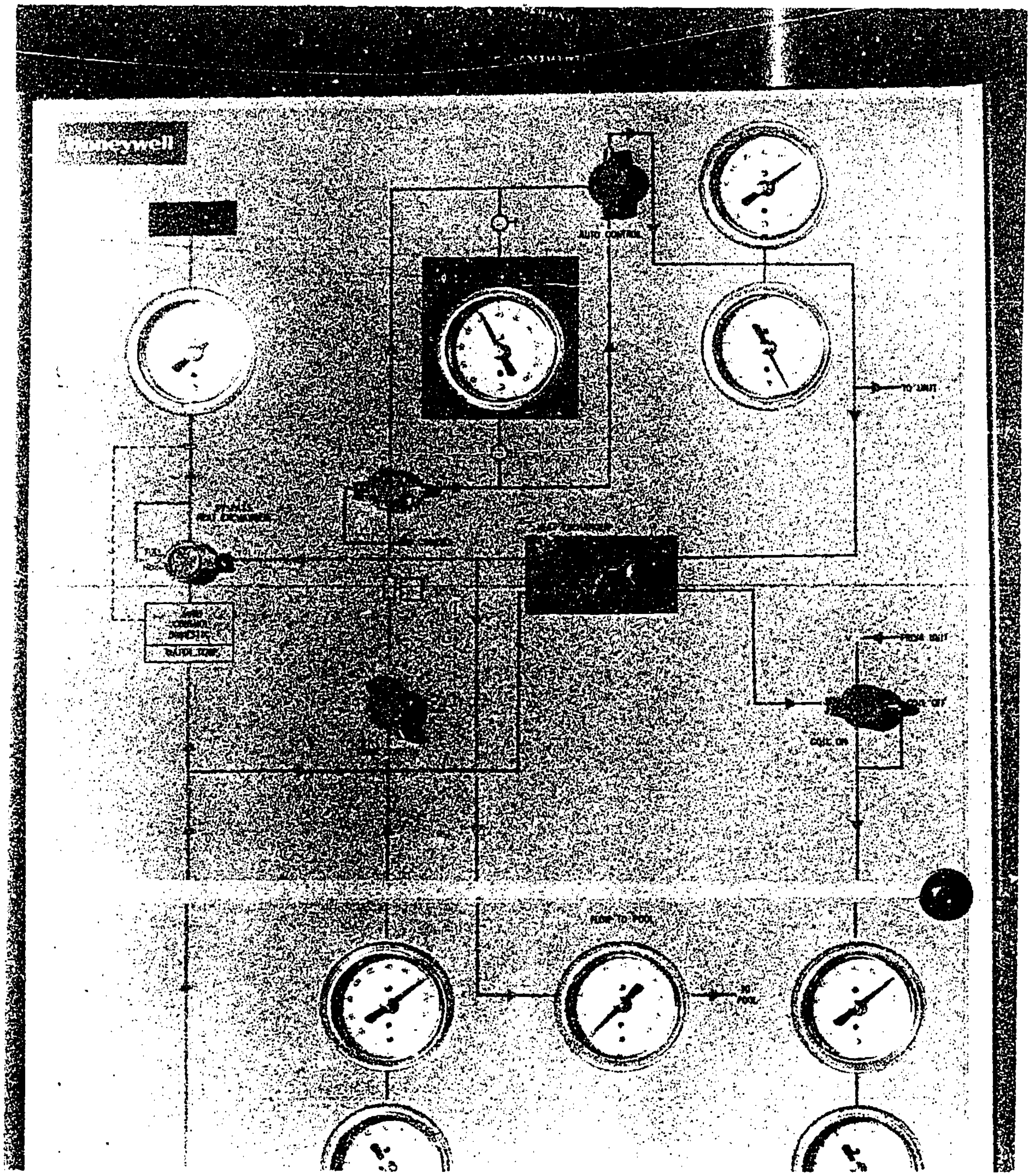
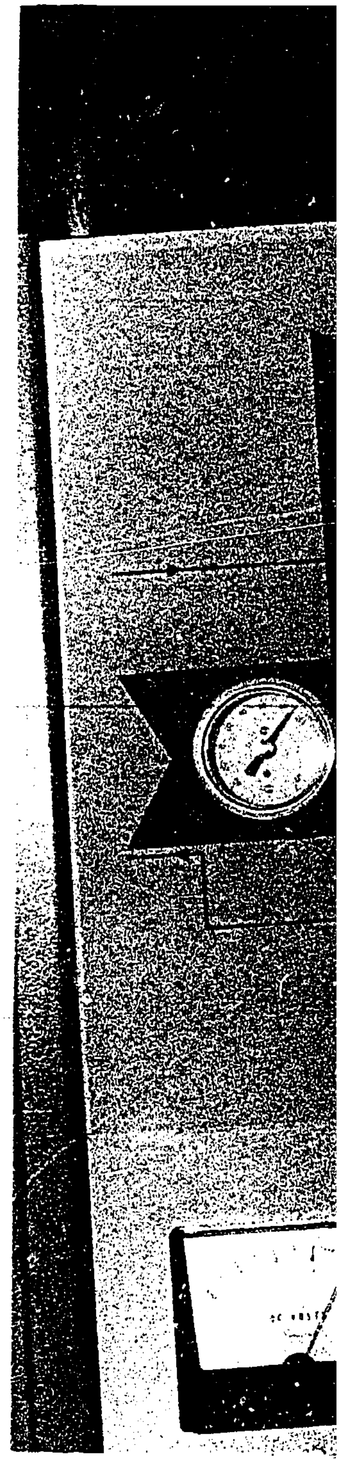


Figure [illegible]-21. Control Panel Closeup



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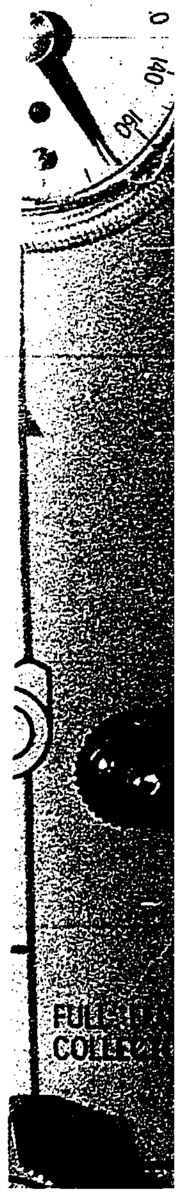

SECTION IV

SYSTEM OPERATION

START UP AND COLLECTOR BYPASS

To place the system in operation the pump switch is placed in the "on" position. This activates the pump through a pneumatic-to-electric interface. Flow is either bypassed around the collectors or through the collectors depending upon the position of the collector bypass switch (Valve 6). The automatic position on this switch allows for future application of a control input which would sense the temperature rise across the collectors and bypasses the collectors if this rise is less than a value to be determined by system testing.

From the discharge of the pump, flow is routed through a flow control modulating valve which is controlled from the flow control switch. The flow can be effectively varied from about 30 gpm to a maximum of 100 gpm. The flow rate is measured by a meter which provides a digital output which is converted to an analog output and displayed on the control panel.

TANK FLOW

Flow around or through the storage tank is controlled by the two tank switches (Valves 1, 2 and 3). Proper positioning of these switches allows flow to be bypassed around the tank in one of two directions or to flow through the tank entering at either the top or bottom of the tank. Selecting the routing is presently a manual function and is determined by operator judgment. Normal operation to charge the storage tank would route flow from the collector to the top of the tank while discharging the tank would require flow to be taken from the top of the tank returning to the bottom. Overnight tank stratification is about 20°F between the top and bottom of the tank and remains stratified from 10 to 12 minutes with a flow through the tank of 80 gpm. Lower flow rates would of course result in longer stratification periods.

The present modes of operation provide for collected heat to be used in domestic water heating, pool air heating or storage tank charging, as well as discharging the tank for air or water heating. The choice of tank charging in series with water or air heating is left with the operator. The flow is routed from the tank area to either the air heat exchangers or water heat exchanger through the position of the heat exchanger mode switch (Valve 5). From this valve, the flow is directed back to the collectors or bypassing the collectors depending upon the position of the collector bypass switch.

AIR HEATING

During operation of pool air heating, the flow is directed through the reheat and fresh air coils or through the fresh air coil alone depending upon the position of the coil control switch (Valve 4). When the valve control is in the automatic position flow is routed through both coils if the pool thermostat is calling for heat. However, if the glycol/water temperature is less than the mixed air temperature, an override controller will automatically bypass the reheat coil and route flow only through the fresh air coil. If the pool air becomes overheated, an override control is provided which controls the heat exchanger mode switch and would direct flow to domestic water heating.

If the solar system does not provide enough heat to satisfy the pool space heating requirement a sensor in the air leaving the reheat coil will act through two controllers to control two steam coil valves. If this air temperature is more than 35°F, both steam valves will be closed. When the air temperature falls to 25°F one steam valve will open, below 20°F both will open. A modulating space thermostat will position a valve on a third steam coil to control the pool air temperature. There is also a low limit discharge controller which prevents discharge air from falling below 65°F.

The air circulation system is provided with dampers which operate in parallel across the return air and fresh air ducts. When one opens, the other closes a like amount assuring a constant air flow in the system. The relative position of the dampers is set by setting a minimum fresh air opening (25 percent by state law during occupied periods). A pool humidistat can override the minimum setting to allow more fresh air intake to reduce the relative humidity in the pool area.

When the building is unoccupied (night setting) the fresh air damper is closed and the space thermostat will cycle the supply fan to maintain the desired temperature. During this period if the relative humidity increases above the humidistat set point the fan starts and the fresh air damper opens to control the humidity level.

DOMESTIC WATER HEATING

During the operation of domestic water heating, incoming water is preheated by the system heat exchanger before entering the domestic hot water storage tank. The water heating control valve has an automatic position which will modulate the water temperature from the exchanger around a predetermined 145°F set point. An additional automatic function prevents glycol from flowing through the heat exchanger when the glycol temperature is less than the incoming domestic water temperature and also if it is below the freezing point of water. When there is insufficient solar heat the conventional steam supplied heat exchanger will supply the additional heat required.

HIGH-TEMPERATURE PROTECTION

High-temperature protection of the collectors is provided by the automatic controller shown in Figure 3-26. This controller has an adjustable high-temperature set point which is presently set at 175°F. A thermocouple located on the back of an absorber plate is the input signal to the controller. When this plate temperature reaches the set point temperature, the pump is automatically started and the collector bypass valve is opened to collector flow regardless of the valve switch position. As flow circulates, the plate temperature will immediately start to drop and would normally shut off the pump. However, a time delay currently set at 15 minutes continues the pump operation to assure that the hot fluid in the collectors is circulated into the building for dissipation in either domestic water heating or storage. During day-long operation in this mode the thermal capacity of the system will prevent the plate temperature from dropping below the set point and the time delay automatically resets and circulation continues. If all system heat requirements are satisfied and the temperature continues to rise, then the emergency dump valve will open and dump water to either the swimming pool filter tank or down the drain depending upon the pool water level. This dumping procedure will not occur until the collector water temperature reaches an adjustable set point of about 185°F.

Figure 4-1 is a complete schematic of the pneumatic control system. This schematic shows all the pneumatic and electrical control devices used in the control system.

SECTION V

INSTRUMENTATION

TEMPERATURE MEASUREMENTS

Temperature measurements are accomplished with the use of stainless steel sheathed copper-constantan thermocouples located at important points of the fluid and air as well as selected points on the collectors. There are additional access points throughout the system for additional temperature measurements should they be needed. For example, each bank of collectors has an access port at the inlet and outlet. All thermocouple access ports incorporate a special Viton sealed plug which permits the insertion and removal of thermocouples by simply pulling the thermocouple out of the port. The plugs are self-sealing upon removal of the probe. Thermocouple locations are shown in Figure 5-1.

Temperature recordings are accomplished with the use of a 24-channel recorder with selectable channels. During operation of a particular mode only those thermocouples needed to evaluate that mode are selected.

ENVIRONMENTAL MEASUREMENTS

Wind speed and direction are measured using a remote sensor and recorder while solar radiation is monitored with a Kipp and Zonen pyranometer installed at the collector plane. The radiation is recorded instantaneously and automatically integrated. Figure 5-2 shows the wind speed and direction sensor as well as the pyranometer while Figure 5-3 shows the recording instrumentation. All measuring instrumentation and recorders were calibrated at Honeywell's instrumentation laboratory.

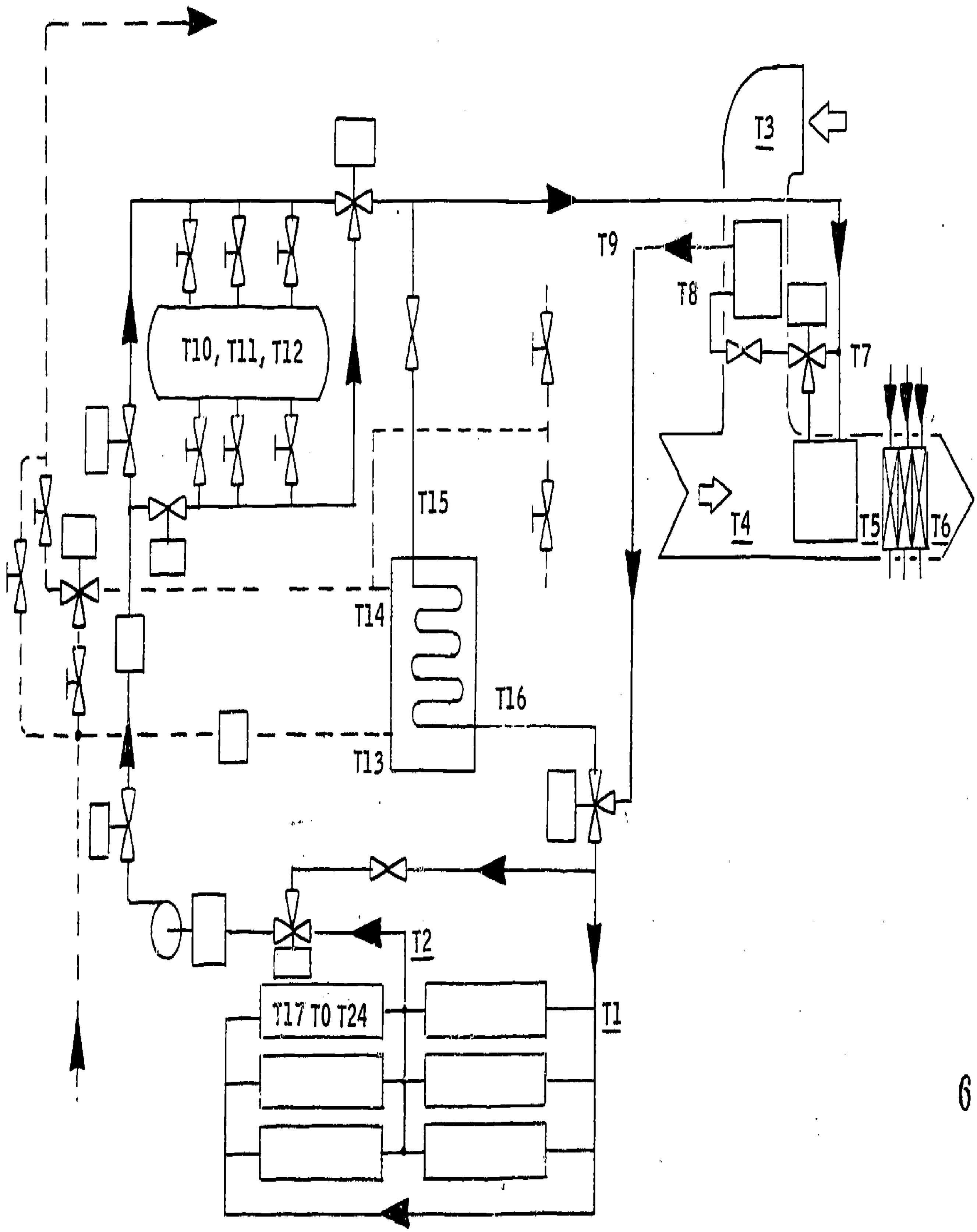

Figure 5-1. Thermocouple Locations

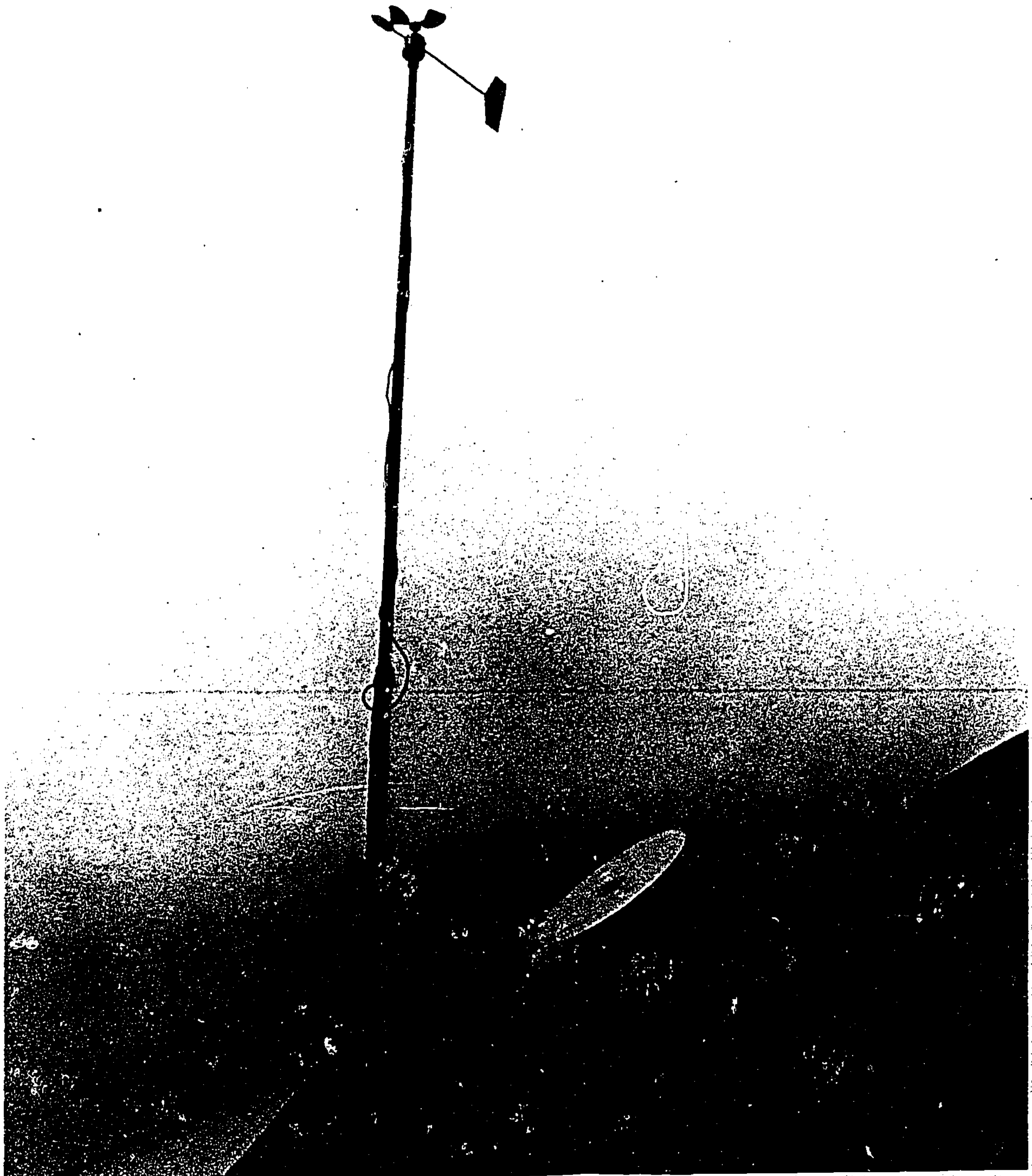

Figure 5-2. Wind Speed, Wind Direction Sensors and Pyranometer Locations

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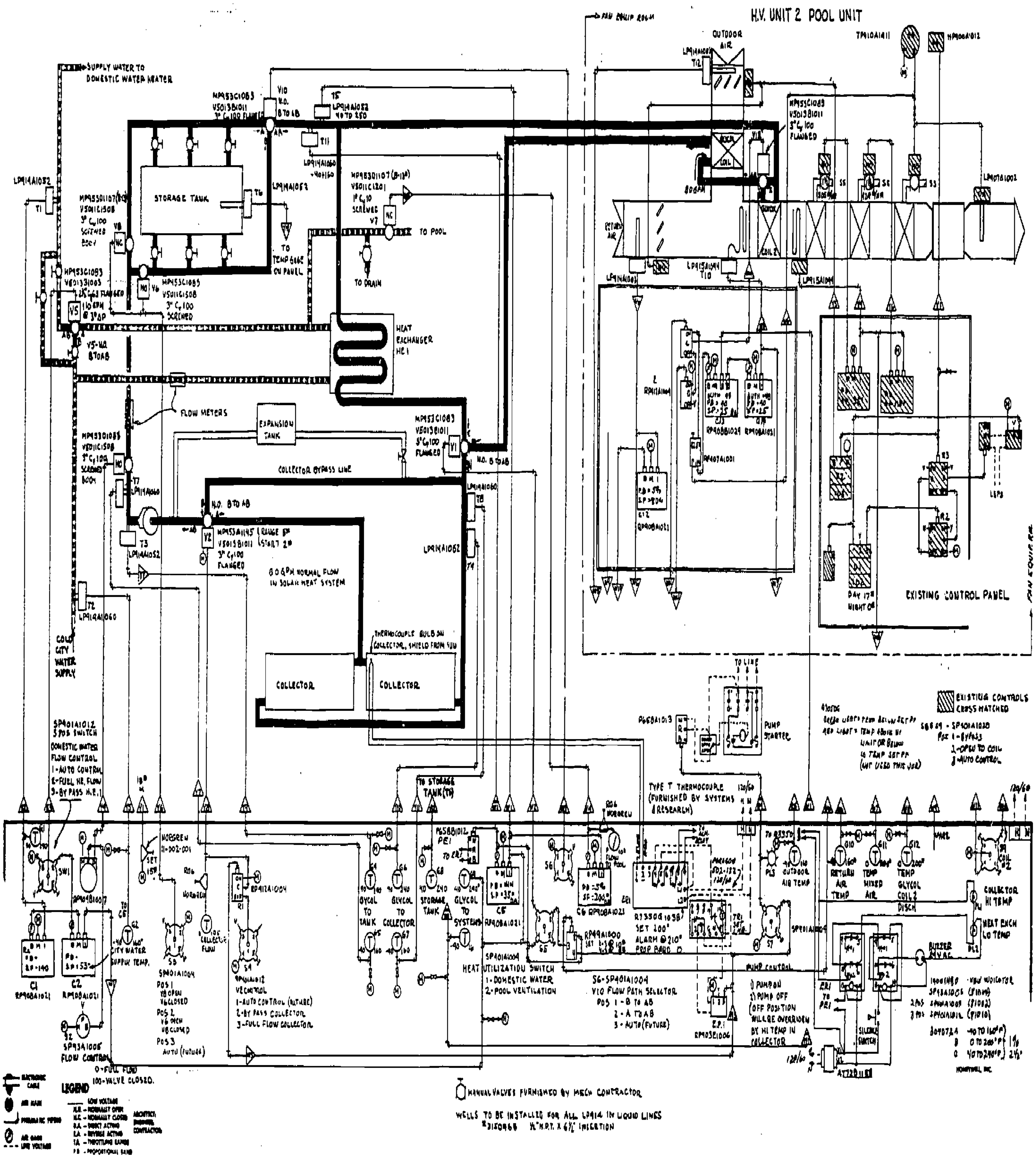

Figure 4-1. Control System Schematic

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SECTION VI
SYSTEM PERFORMANCE

OPERATIONAL MODES

The present operation is regulated by manually selecting various modes, although the safety features such as the prevention of collector overheating will override other functions. The manually selectable modes are:

- Solar collector to fresh air heating coil
- Storage tank to fresh air heating coil
- Solar collector to fresh air and mixed return air heating coils
- Storage tank to fresh air and mixed return air heating coils
- Solar collector to domestic water heating
- Solar collector to storage tank
- Storage tank to domestic water heating
- Solar collector to storage and domestic water heating

In addition it has been possible to heat the pool water by supplying hot make-up water to the pool filter tank. This tank is part of the pool recirculating loop and the incoming hot water is mixed with the pool water during the filtering process. The pool recirculating flow rate is 600 gpm so adequate mixing occurs before the water enters the swimming pool.

The basic objectives of the tests so far have been to evaluate the total system rather than individual components. However, the total system tests do yield insights into the operation and performance of individual components.

Initial collector operations were used primarily to obtain information on the operation of the system. That is, with given ambient conditions, solar intensity and school heating demands the modes of operation that would best match the load to collected energy were determined. The results of these preliminary tests showed the following:

1) Operator selection of heating modes is adequate for specialized testing but inadequate for maximizing the amount of energy distributed to the school. For example, if collector to air heating is sufficient for early morning hours, the operator can not adequately determine when such an operation should cease and another begin due to steadily increasing solar radiation along with a decreasing heat load coupled with long transport delays (15 minutes).

2) The domestic water heat exchanger capabilities were not being maximized because heating of water would occur only during domestic water usage. Hence, going to a domestic water heating mode would not provide an adequate load for the system during low usage hours.

3) Varying heat demands from pool air heating and water heating necessitates using an automatic control system which would provide heat to areas on a priority basis. Hence, air heating demand is first; if this is satisfied, then domestic water heating is used. The present control system is not designed to be operated on a two or three mode simultaneous operation but all control values necessary for this operation are installed.

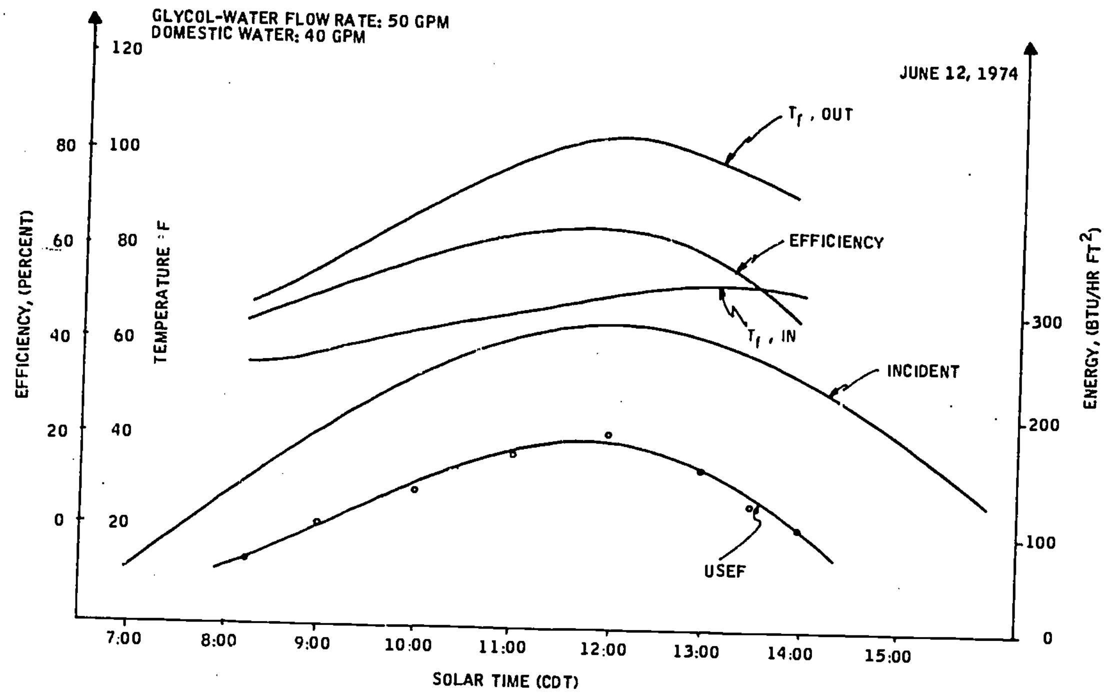

Figure 6-1. Domestic Water Heating from Collector

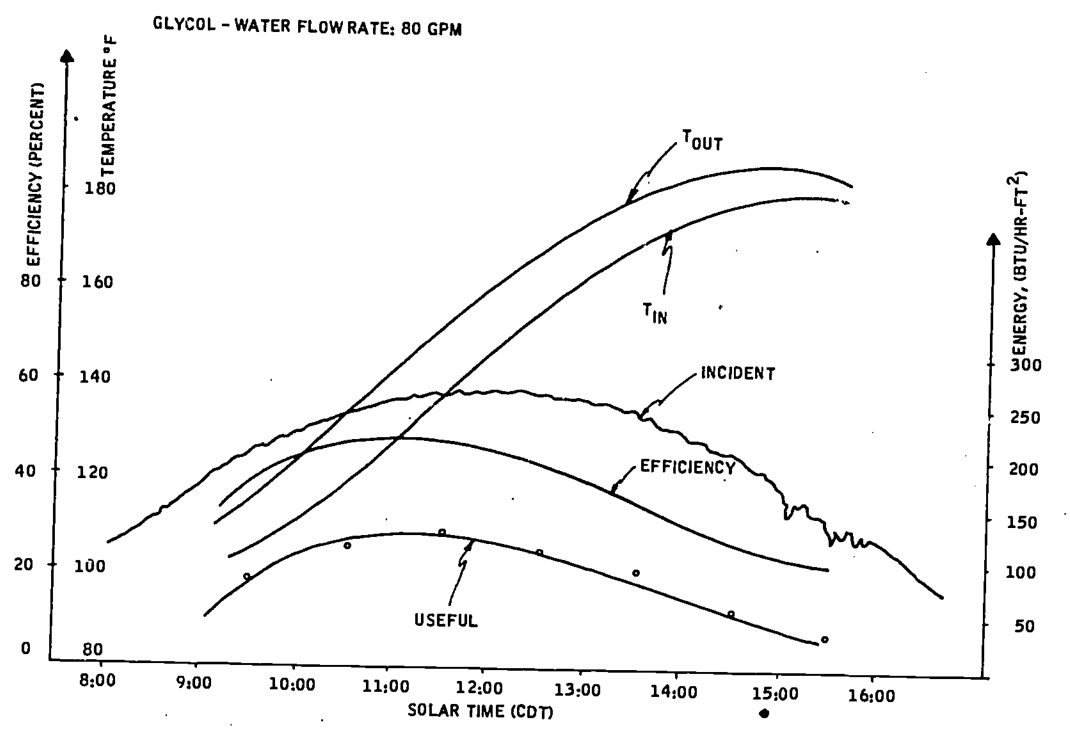

Figure 6-2. Storage Charging from Collector, June 26, 1974

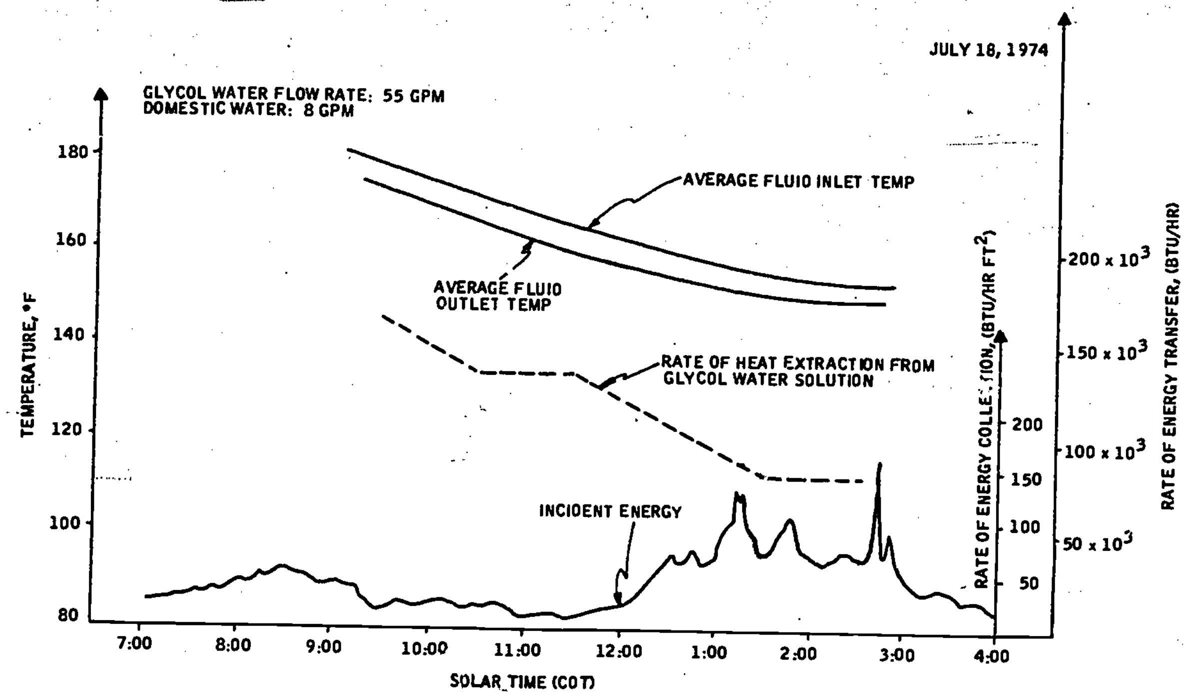

Figure 6-3. Domestic Water Heating from Storage

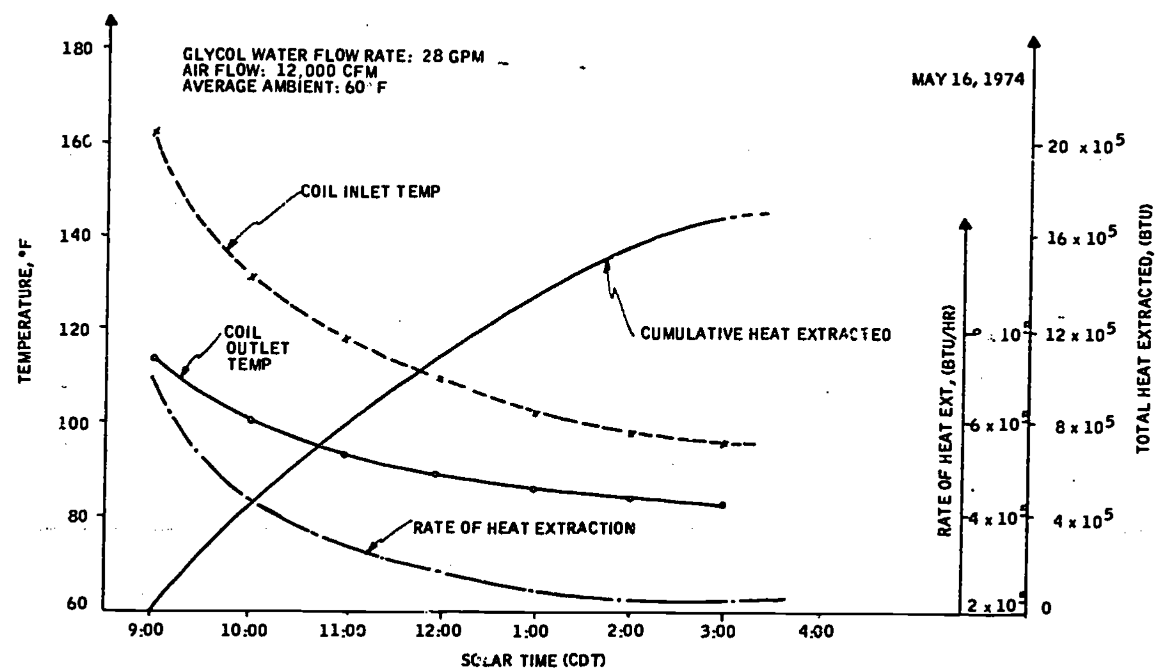

Figure 6-4. Fresh Air Heating from Storage

4) Emergency control functions to prevent collector overheating work quite satisfactorily. Considerable time was spent in checking out their function. Initial tests showed that collector running time when the high-temperature set point was reached was not adequate to extract heat from the collector. An additional time delay was installed to increase this run time to 20 minutes. Sensing of collector temperatures by sensing absorber plate temperatures appears to be an adequate input to the automatic controller.

5) When ambient conditions are above 70°F with high incident radiation, the system load is not adequate for efficient operation except for random periods of domestic water use.

The installation of the collector and test instrumentation checkout was completed during early May 1974. From then until this report was written the following specific tests were conducted:

- Direct heating of domestic water from the collector
- Charging the storage tank from the collector
- Discharging the storage tank for domestic water heating
- Discharging the storage tank for pool air heating

In addition to these tests, which were conducted on a day-long basis without changing the test mode, several multi-mode operations were conducted. These would include heating pool air until approximately 11:00 a.m., switching to domestic water heating over the high-demand noon period, and then completing the day by pool air heating and tank charging.

Figures 6-1 through 6-4 are typical curves of data compiled on day-long tests of the modes previously metnioned. This data is representative of several operations under each mode.

Figure 6-1 shows domestic water heating from the collector on June 12, 1974. The average values are:

- Average incident energy - 1,179,680 Btu/hr
- Average collected energy - 670,000 Btu/hr
- Average efficiency - 57 percent

This data represents what efficiency can be obtained by adequately matching the load to the incident radiation. Domestic water was used in this case to supply heated make-up water to the swimming pool area after a pool shutdown period.

Figure 6-2 shows storage tank charging from the collectors. Total incident energy was 7,114,229 Btu's while total collected energy was 2,767,601 Btu's for an average efficiency of 39 percent. Note that the outlet temperature from the collectors is at 184°F at 3:30 PM. Considerable energy is still available in the thermal capacity of the system. If an adequate load could have been used from 3:30 PM until the outlet temperature had returned to its initial starting temperature, the average efficiency would have increased. Further testing to obtain this information is being formulated.

Figure 6-3 is domestic water heating from the storage tank. This operation consisted of a test performed on an additional circulating loop placed between the domestic water storage tank (3000 gallons) and the solar system heat exchanger providing a continuous flow of 8 gpm. Total heat lost by the glycol-water storage tank was computed at 643,250 Btu's while that gained by the domestic water was 618,250 Btu's. The 24,500 Btu's difference is due to heat losses in the piping and averaging of storage tank temperature. The error corresponds to less than 1°F based on storage tank capacity.

Figure 6-4 shows pool air heating from the storage tank. Fresh air heating without the return air heating coil was used. The operation was terminated at 3:00 p.m. due to pool air overheating caused by a malfunction in the control system.

Appendix D illustrates the methods used to calculate the amount of energy, efficiency, and system performance parameters.

Figures 6-5 through 6-13 illustrate typical data curves for several days during the month of July 1974. Summer modes consisted of domestic water heating from the collector and system storage. Storage discharge was accomplished at night using the recirculating loop recently installed. This loop provides for continuous recirculation between the domestic water storage tank and the solar energy domestic water heat exchanger. Solar collector heating of school space has not been accomplished on a regular basis due to lack of heat demand.

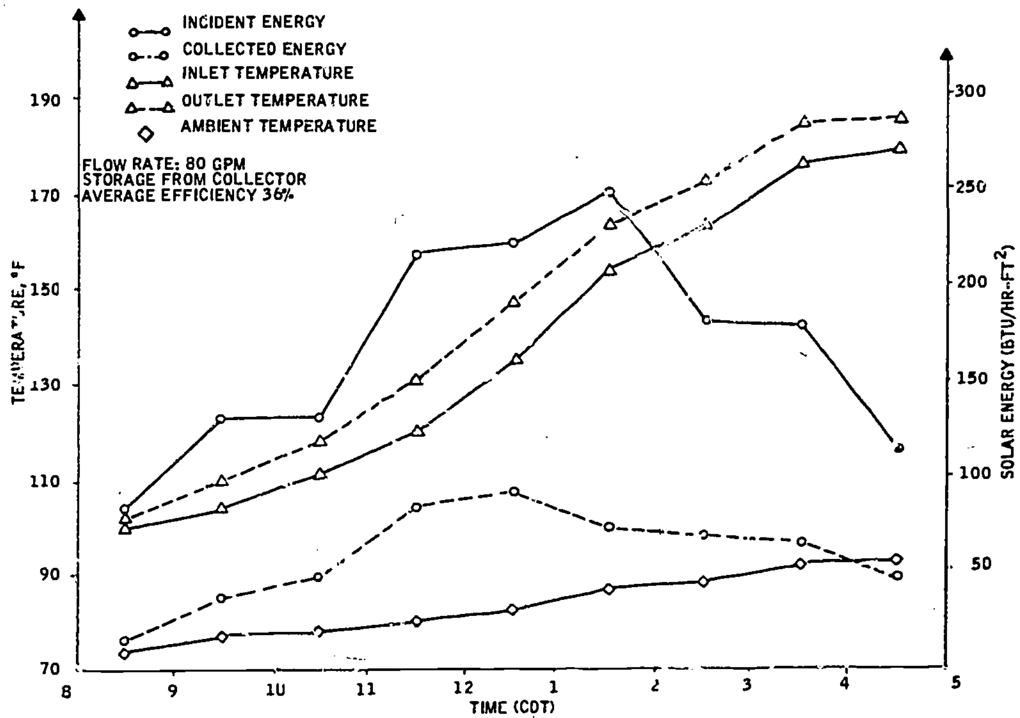

Figure 6-5. Solar Collector Data, Northview Junior High School, Minneapolis, Minn., July 21, 1974

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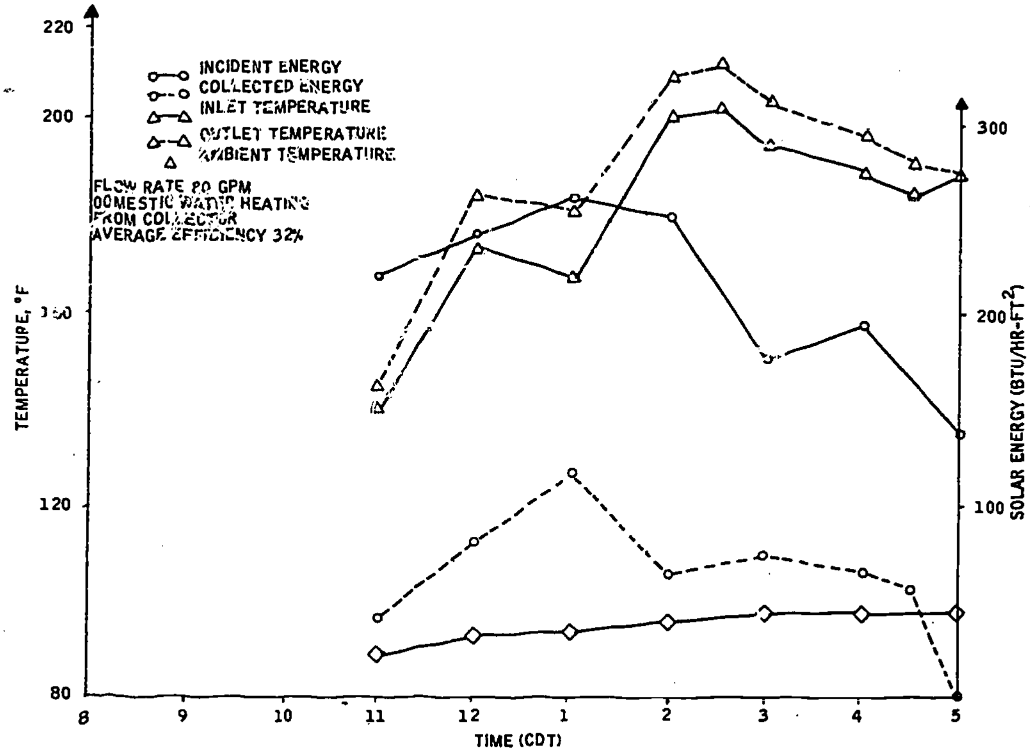

Figure 6-6. Solar Collector Data, Northview Junior High School, Minneapolis, Minn., July 8, 1974

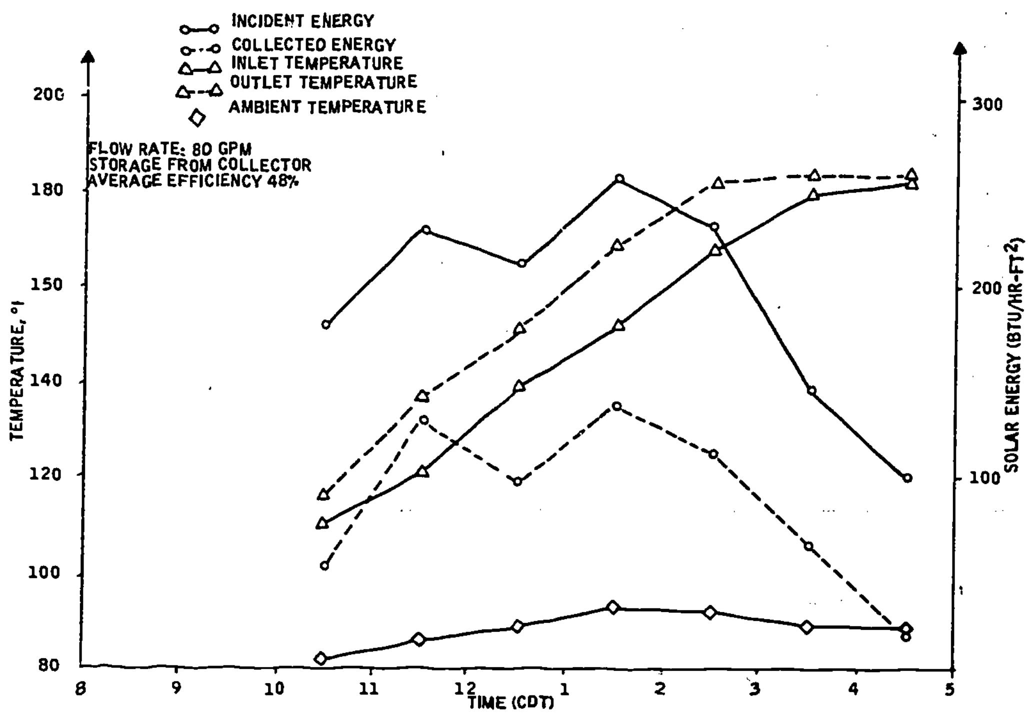

Figure 6-7. Solar Collector Data, Northview Junior High School, Minneapolis, Minn., July 9, 1974

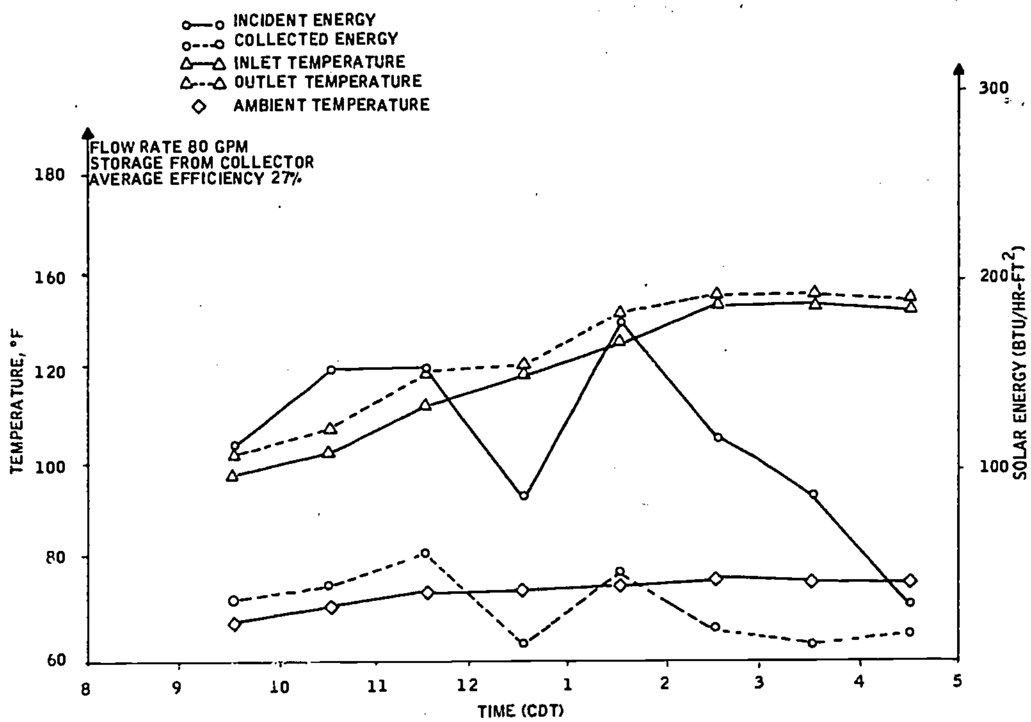

Figure 6-8. Solar Collector Data, Northview Junior High School, Minneapolis, Minn., July 11, 1974

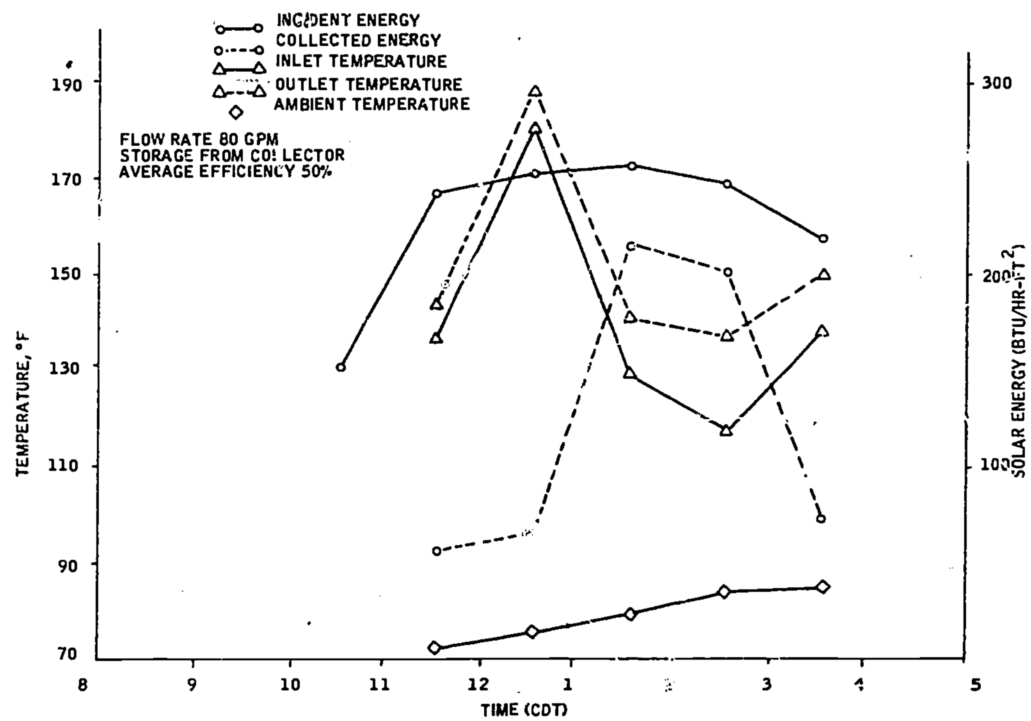

Figure 6-9. Solar Collector Data, Northview Junior High School, Minneapolis, Minn., July 22, 1974

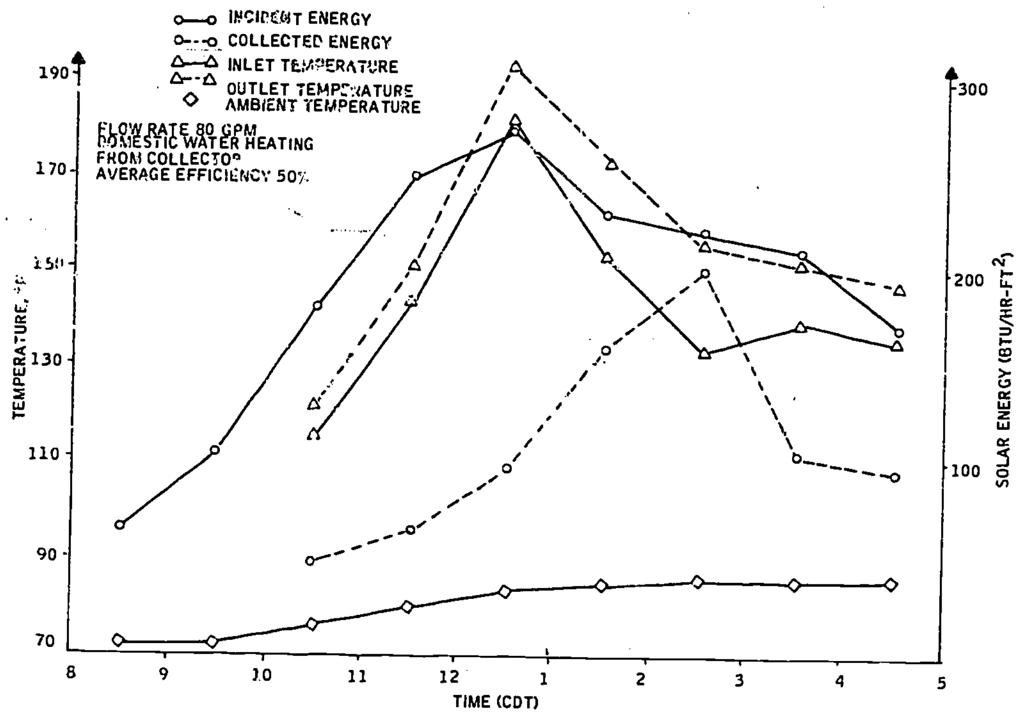

Figure 6-10. Solar Collector Data, Northview Junior High School, Minneapolis, Minn., July 25, 1974

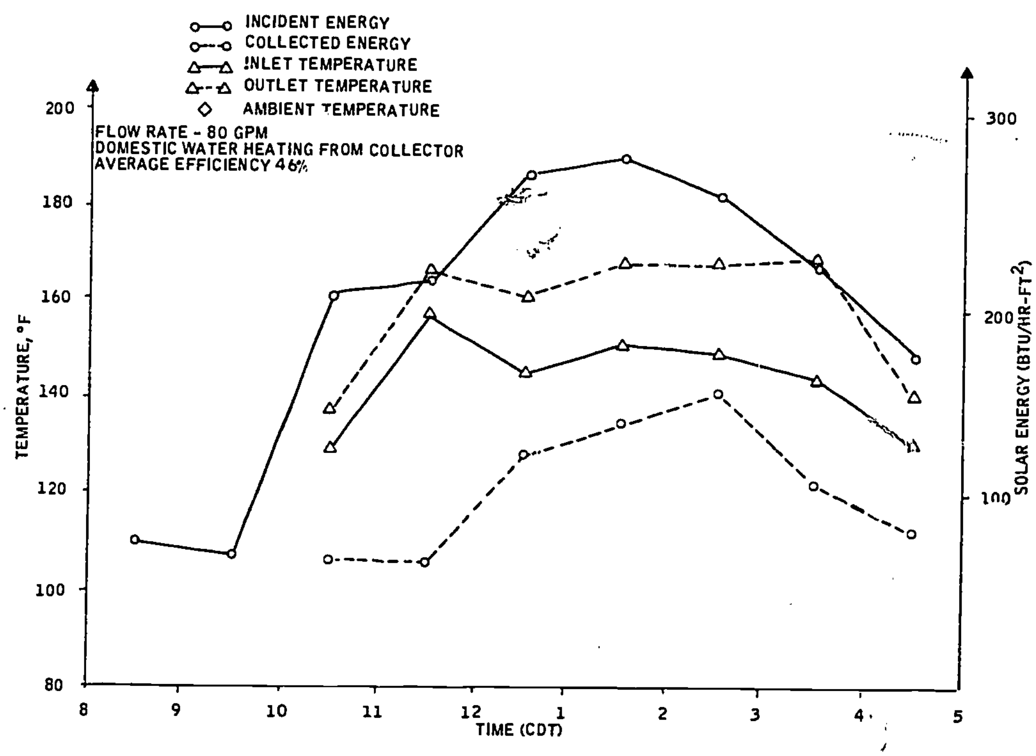

Figure 6-11. Solar Collector Data, Northview Junior High School, Minneapolis, Minn., July 26, 1974

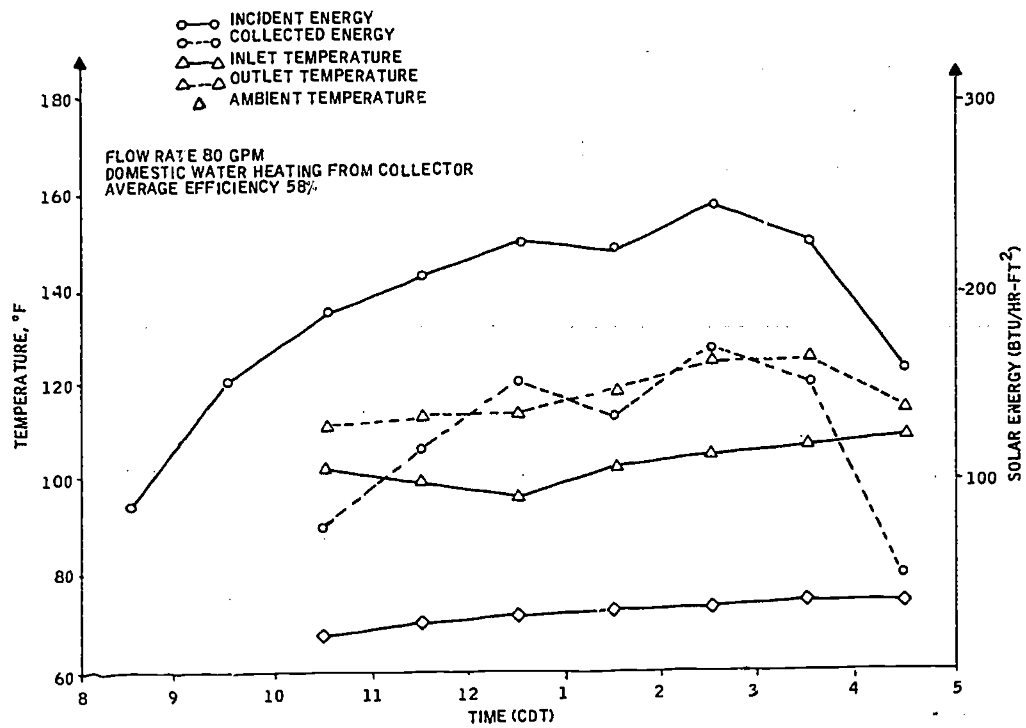

Figure 6-12. Solar Collector Data, Northview Junior High School, Minneapolis, Minn., July 30, 1974

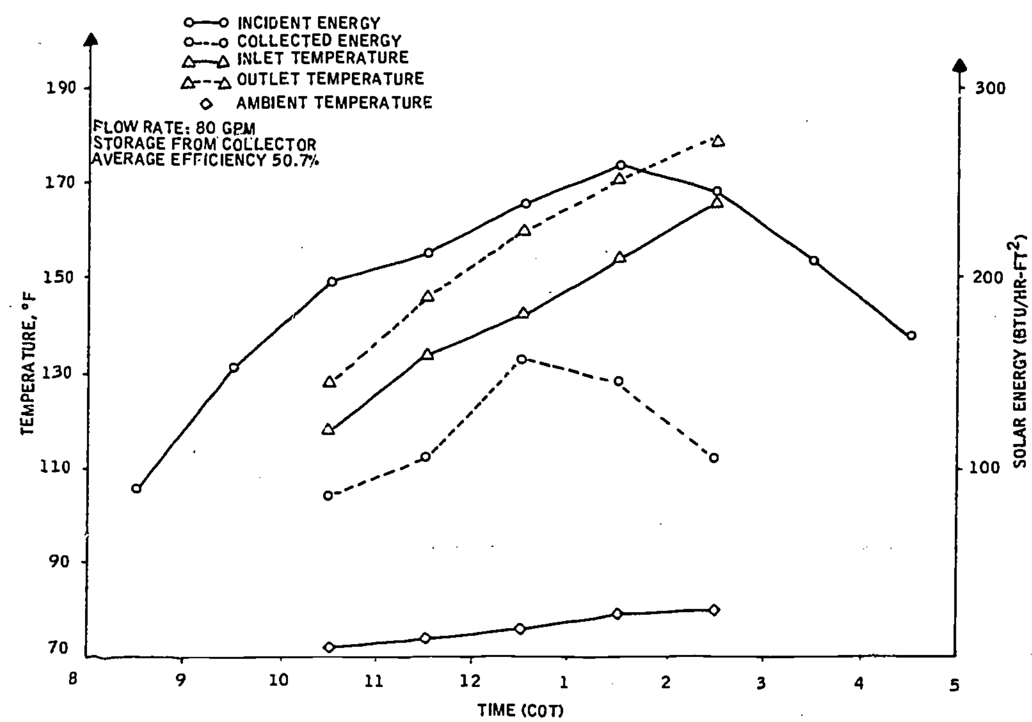

Figure 6-13. Solar Collector Data, Northview Junior High School, Minneapolis, Minn., July 31, 1974

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SECTION VII

COST ANALYSIS

The costs of the solar system assembly subtasks are tabulated below. These costs do not include engineering labor. All work outside of the collector assembly work was completed under fixed-price contracts with outside vendors. Some of these vendors were other Honeywell divisions. The costs reflected below include materials and labor which are difficult to separate on fixed price contracts to outside vendors.

Under the time constraints imposed by the contract and the usually higher costs of a prototype system the costs listed below would not reflect the costs of a similar system installed using the knowledge gained from the installation of the North View project and production techniques aimed at lowering collector costs.

Item	Cost
Collectors (5000 square feet)	$98,600.00
Plumbing (includes manual valves, insulation, pipe, fittings and labor)	63,782.00
Structure (materials and labor)	8,000.00
Components (heat exchangers, pump, expansion tanks, air control)	4,500.00
Storage tank and pad	6,330.00
Control system (panel, valves and controls)	12,000.00
Instrumentation	4,878.00
Electrical	1,300.00
Antifreeze	3,520.00
Fencing, painting and air filters	4,100.00
Collector installation	4,200.00
Total	$211,210.00

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SECTION VIII

CONCLUSIONS

Supplemental heating of large commercial buildings using solar energy is feasible. This concept is being demonstrated by the implementation and testing of a large scale (5000 square feet) flat plate collector array for supplemental air heating and domestic water heating. The system is capable of supplying up to 4 million Btus per day depending upon ambient conditions and building demand. Specific technical conclusions resulting from system operation up to this time are:

(1) Optimum system performance can only be realized if the building heat demand matches the optimum operating conditions of the collector array. If demand is low, system efficiency decreases due to higher collector inlet temperatures.

(2) Instantaneous data values used for system evaluation are nearly meaningless due to the inherent thermal capacity of the system and the large time delays (10-15 minutes) associated with fluid transport times. Integrated values of incident radiation and collected energy over a specified time period should be used to evaluate system performance.

(3) Fully automatic operation of large-scale collector systems is essential for successful acceptance of these systems. Control functions should be designed such that their operation approaches the simplicity of present-day modern heating and cooling facilities. Data for designing such a control system are presently being accumulated and specific recommendations will be made from these data.

(4) System installation did not present any special problems which could not be handled with present-day construction practices. Simpler designs for collectors, storage and plumbing will become apparent as experience is gained from the use and maintenance of the system.

(5) Collector designs should incorporate as few interconnections as possible to minimize installation costs and leakage points. For the high-temperature operation necessary for absorption type air conditioners, the collectors should be designed to operate under moderate pressures in a closed system. Design features should also include a provision for high-temperature control in the event of low usage of incident energy or collector circulation failure.

(6) System operating costs consist of the pump costs of about $.09/hour of operation based on $.03/kilowatt hour plus operator attendance. Presently this time is minimal (manual mode selection) but can be eliminated with fully automatic control.

(7) Maintenance of the collector array has consisted of the replacement of three sheets of glass at a cost of $80.00/sheet including labor plus the elimination of leaks from rubber hose interconnections. These connections have proved to be a consistent source of leaks but design changes and hose replacement procedures will eliminate the leakage problem. Maintenance of leaks has amounted to approximately four man-hours/week.

(8) Three absorber panels have been ruptured due to overpressurization of the system. These ruptures occurred at about 25 psi panel pressure and were directly attributed to operator error. Design consideration should be given to higher safety margins for pressure operations. Replacement of ruptured panels has amounted to six man-hours per panel.

(9) The addition of a heat exchanger to heat pool water should provide an adequate load for summer operation. The pool presently loses about 1,250,000 Btus/day with the boiler plant inoperative.

This study has provided a number of answers to system design problems and continued accumulation of data regarding system performance and maintenance will provide additional information for future system design.

APPENDIX A

FLAT PLATE COLLECTOR THERMAL ANALYSIS

Performance prediction of a flat plate collector design requires an accounting of all heat flow paths by which energy may enter and leave the collector. This accounting includes the calculation of the radiation absorption in the solar spectrum, the radiation losses in the infrared, the natural convection between inclined parallel plates, external natural or forced convection from inclined plates, and heat conduction and heat transfer by forced or free convection to the collector fluid. Analytical correlations which apply to these processes in varying degrees are available in the heat transfer literature. Caution is necessary in the use of these relations since conditions such as transient solar flux, non-uniform surface temperatures, end-effects, temperature-dependent thermophysical properties, large-scale free-stream turbulence, and combined forced and free convection may invalidate the predictions.

The published literature on the analysis of solar energy collection dates from the 1880's: one of the well known treatments of solar energy collection is the work of Hottel and Woertz[1] in 1942. Hottel and Whillier[2] in 1958 and Bliss[3] in 1959 presented certain plate efficiency factors, the use of which simplifies the calculation of the collector performance. While reducing the amount of labor involved, the procedures in the literature are not entirely satisfactory for a detailed design optimization. The principal failure of these methods is their inability to treat adequately the transient condition. A second drawback lies in the method of handling the effects of a non-uniform temperature distribution in the flow direction in the absorber and other components. While the derived efficiency factors partially account for this effect, the variation of the radiation and convection heat transfer coefficients along the surfaces is not included in the analysis.

A factor of major importance in the analysis and design of a solar collector is the time-varying nature of the collection process, the incident solar flux variation from zero to a daily maximum every 24 hours. In addition, short-term fluctuations due to variable cloud cover occur during the day. A comprehensive analysis of a collector must consider this time-dependent solar input to adequately predict the thermal performance.

The computer program for the analysis of the flat plate collector does consider the transient condition and the non-uniform temperature distribution. One of the inputs to the program, in addition to all pertinent geometrical and environmental parameters, is the solar flux and incident angle. The flux and angle may or may not be a function of time, depending on the desired result. The instantaneous and average performance may be calculated by inputting a daily record of solar flux and incident angle. The time constant of the system may be obtained from an analysis of the response to a sudden inception of solar flux.

The flat plate program was developed with the following goals:

- The transient nature of the problem must be adequately treated.

---

[1] H. C. Hottel and B. B. Woertz, "The Performance of Flat Plate Solar Heat Collectors," Trans. ASME 64, 91 (1942).

[2] H. C. Hottel and A. Whillier, Trans. Conf. on the Use of Solar Energy, Vol. 2, Part I, p. 74, Univ. Arizona Press (1958)

[3] R. W. Bliss, "The Derivations of Several "Plate Efficiency Factors" Useful in the Design of Flat Plate Solar Heat Collectors," Solar Energy 3, 55 (1959)

- The effects of non-uniform temperature distributions must be considered.
- The system energy balance and efficiency must be calculated on an instantaneous as well as on a daily basis.
- The input routine to the program must be able to handle conveniently a wide variety of geometrical and environmental parameters.
- The output routine must provide temperatures, temperature rates, heat flows, heat transfer coefficients, etc., to provide good physical insight into the thermal performance.
- The output routine must produce an overall summary of collector performance in the form of a system energy balance.

The following paragraphs describe the computer program; sample calculations are presented for typical summer and winter days.

PROGRAM DESCRIPTION

This appendix presents a short description of the analytical procedures and the main features of the program.

To treat the transient condition and the non-uniform temperature distribution, the collector is arbitrarily subdivided into a number of physical elements called "nodes." A typical subdivision is indicated in Figure A1.

The effect of the fluid temperature from inlet to outlet is best examined by the indicated subdivision. The temperature of each node is computed as a function of time using the straightforward, explicit method in which the temperature rate change of node i is related to its present temperature and the temperatures of the neighboring nodes j by

$$C_i \frac{dT_i}{dt} = \sum_j K_{ij} (T_j - T_i) + S_i$$

where C_1 is the heat capacity of node i, T_i is its temperature, dT_i/dt is its rate of temperature change, T_j is the temperature of node j, and S_i is the solar heat absorption. The coupling coefficients, K_{ij}, called conductances, depend on the heat transfer mode and, in general, on the temperature.

There are M such equations, one for each mass element, and the solution of these simultaneous, nonlinear, first-order, ordinary differential equations yields the temperature history at M discrete points throughout the collector. Since the temperature of each node is assumed to be given at some initial time, t_o, the rates $(dT_i/dt)_{t=t_o}$ are given and the temperatures at $t_o + \Delta t$ are obtained by

$$T_i (t_o + \Delta t) = T_i(t_o) + \left(\frac{dT_i}{dt} \right)_{t_o} \Delta t$$

Then the rates at $t_o + \Delta t$ can be calculated and the procedure is repeated until the desired time range is covered.

Solar Absorption

The solar input (S_i) to each element is obtained by an analysis of the reflection, transmission, and absorption of energy in the solar spectrum by a system of N covers over an absorbing surface. This analysis has been carried out taking into consideration the two components of polarization, the reflection of each cover-air interface, the absorption of each cover, and the absorption and reflection by the absorber. The program is then able to treat the four important combinations of specular and/or diffuse conditions:

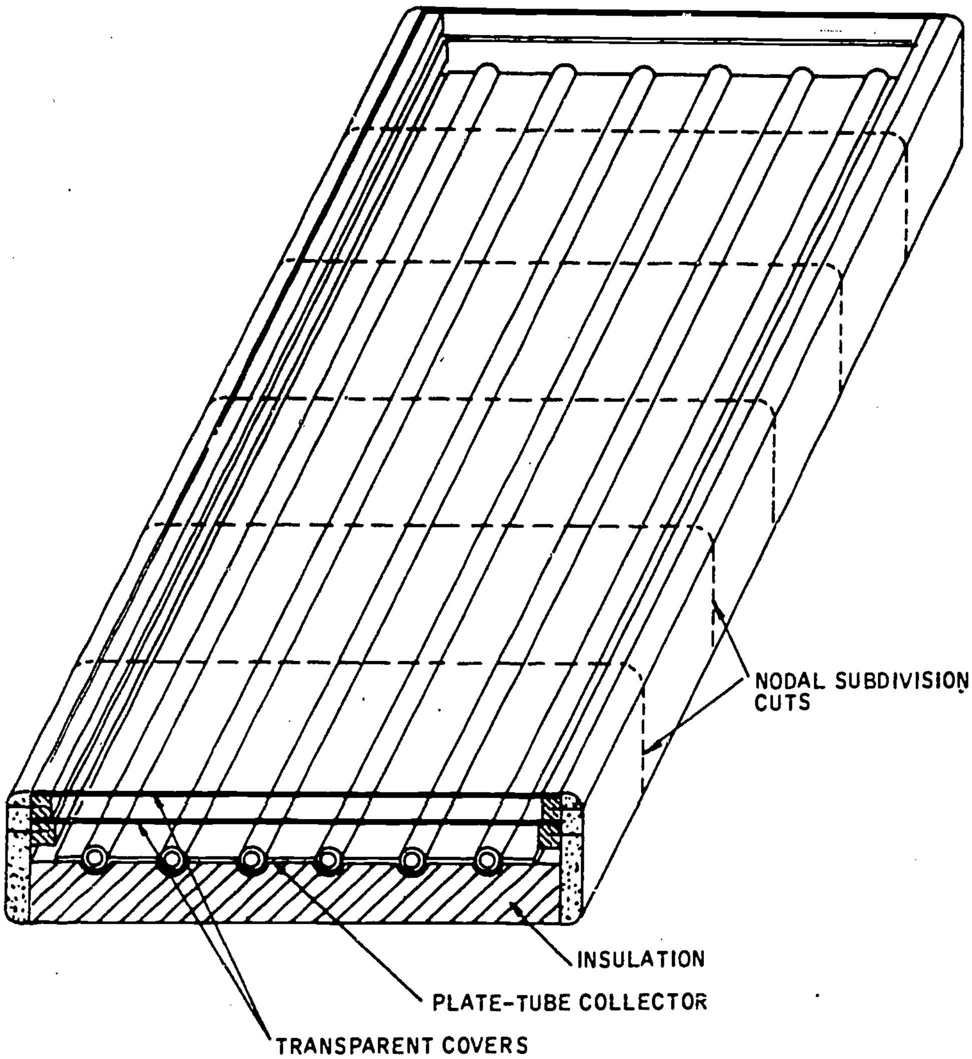

Figure A1. Flat Plate Solar Collector

- Direct solar flux with specular absorber
- Direct solar flux with diffuse absorber
- Diffuse solar flux with specular absorber
- Diffuse solar flux with diffuse absorber

The program analyzes the reflection, absorption, and transmission of the direct and and diffuse components of the incident solar flux separately and then combines the results to obtain the total heat absorption. The effects of specular versus diffuse absorbers may be examined.

Heat Losses

The absorber loses heat to the environment by:

- Emission of energy in the infrared spectrum and subsequent absorption, transmission, and re-emission of this energy by the cover system
- Convection of energy to the adjacent cover and subsequent radiation and convection of the energy by the cover system
- Conduction through the layer of insulation on the rear surface of the absorber
- Conduction, convection, and radiation from the absorber to the side walls of the collector

The final heat rejection to the environment is by convection and radiation from the external surfaces of the collector.

Heat transfer correlations exist in the literature which apply to the above cases. As pointed out earlier, caution must be exercised in the use of the published results due to the fact that the actual physical conditions may not correspond to the ideal conditions under which the correlations were derived.

Heat Transfer to Collection Fluid

The purpose of the solar collector is to add heat to a working fluid which may be liquid or gas. The present version of the computer program assumes an integral tube-plate absorber with parallel flow tubes. The fluid is assumed to enter the tubes from a common header and empty into a collection header.

Intimately connected with the absorber-fluid heat transfer is the question of the temperature drop in the absorber due to the fact that the heat must flow laterally in the absorber to the flow tubes. This temperature drop results in a higher absorber temperature than that which would occur if the absorber were a perfect conductor or were very thick. Since the collector cost depends on the type and amount of material used in the absorber, an optimization problem exists.

Past investigators have treated the absorber-to-fluid tube conduction problem as that of a fin exchanging heat with its surroundings through a heat transfer coefficient constant over the fin surface. But since the largest heat flow quantity is the solar flux and since this flux is uniform over the fin surface, it appears that it would be more proper to treat the fin surface heat flow as uniform. The fin analysis lies between these two approaches, considering the conduction, radiation, and natural convection loss terms to be proportional to the temperature difference between the absorber and its surroundings.

Input/Output

To be truly a useful analytical tool for the optimization of the collector, the input/output routines must be written so as to reduce the labor and complications connected with the input of the physical parameters and the interpretation of the results. Special attention was given to this subject in the Honeywell program.

In its present form the input to the program is via punched cards. The card formats are arranged so that logically related parameters, such as the thermal properties of the absorber, are all on one card. The input information is re-arranged by the computer, and it is output in a convenient format for future reference. Figure A2 is a sample output of the geometrical, physical, and environmental parameters of a typical case.

```
PHYSICAL DATA

   COLLECTOR
      LENGTH, M                                     .12192E 01
      WIDTH, M                                      .12192E 01
      NO. COVERS, 1
      ABSORBER TO COVER [illegible] GAP, CM         .254[illegible]E 01
   STORAGE TANK
      LENGTH, M                                     [illegible]
      DIAMETER, M                                   [illegible]
      VOLUME, CU M                                  [illegible]

   ABSORBER
      MATERIAL, ALUMINUM
      SURFACE CONDITION, SPEC
      SOLAR ABSORPTANCE,                            [illegible]
      INFRARED EMITTANCE,                           [illegible]
      THICKNESS, CM                                 [illegible]
      DENSITY, KGM/CU M                             [illegible]
      SPECIFIC HEAT, W-SEC/KGM-C                    [illegible]
      THERMAL CONDUCTIVITY, W/M-C                   [illegible]
      NO. TUBES, 12
      TUBE ID, CM                                   [illegible]
      TUBE OD, CM                                   .12573E [illegible]
      TUBE LENGTH, M                                [illegible]
      TUBE SPACING, CM                              [illegible]

   COVER NO.1(INNER)
      MATERIAL,      GLASS
      INDEX OF REFRACTION                           [illegible]
      EXTINCTION COEFFICIENT, 1/CM                  [illegible]
      INFRARED EMITTANCE                            [illegible]
      THICKNESS, CM                                 [illegible]
      DENSITY, KGM/CU M                             [illegible]
      SPECIFIC HEAT, W-SEC/KGM-C                    [illegible]
      THERMAL CONDUCTIVITY, W/M-C                   [illegible]

   INSULATION
      MATERIAL,GL FIBER
      SOLAR ABSORPTANCE,                            [illegible]
      INFRARED EMITTANCE,                           [illegible]
      THICKNESS, CM                                 [illegible]
      DENSITY, KGM/CU M                             [illegible]
      SPECIFIC HEAT, W-SEC/KGM-C                    [illegible]
      THERMAL CONDUCTIVITY, W/M-C                   [illegible]


ENVIRONMENTAL DATA

   AMBIENT CONDITIONS
      TEMPERATURE, C                                [illegible]
      PRESSURE, BAR                                 [illegible]
      WIND VELOCITY, M/SEC                          [illegible]
```

Figure A2. Physical Data

The first output, after the listing of the input, is an analysis summary of the reflection, transmission, and absorption of the solar energy by the cover system (Figure A3 is a sample output for a two-cover system). This analysis, which is normalized for a unity input flux, needs to be made only once for any given cover/absorber combination. The results for incidence angles from 0 to 90 degrees at 9-degree intervals are stored in the computer for future use. (A parabolic interpolation routine over the 9-degree increments provides more than sufficient accuracy for all incidence angles.)

Results are presented for both components of polarization and for the average. The output includes the four important combinations of direct or diffuse incident flux with the specular or diffuse absorber.

The next output (Figure A4) lists the solar absorption of each physical element. The absorption of the direct and diffuse components are listed separately. Any percentage of incident diffuse flux may be input.

The internodal conductances are output (Figure A5) at each time interval in order to provide details on the thermal coupling of the various physical elements. The following output gives the corresponding internodal heat flows (Figure A6). This information indicates the magnitude of the various heat flow mechanisms and their relative importance.

ANALYSIS OF 1 COVER SYSTEM PLUS ABSORBER WITH SOLAR ABSORPTANCE = .90

DIRECT INCIDENT, SPECULAR ABSORBER

INC ANGLE DEGREES	RADIATION COMPONENT	SYSTEM REFLECTION	SYSTEM ABSORPTION	ABSORBER ABSORPTION	COVER 1 ABSORPTION	COVER 2 ABSORPTION	COVER 3 ABSORPTION	COVER 4 ABSORPTION
.0	TOTAL	.16204	.83796	.81221	.02575	.00000	.00000	.00000
	PARALLEL	.16204	.83796	.81221	.02575	.00000	.00000	.00000
	PERPENDICULAR	.16204	.83796	.81221	.02575	.00000	.00000	.00000
9.0	TOTAL	.16202	.83798	.81210	.02588	.00000	.00000	.00000
	PARALLEL	.16413	.83587	.81000	.02587	.00000	.00000	.00000
	PERPENDICULAR	.15992	.84008	.81420	.02589	.00000	.00000	.00000
18.0	TOTAL	.16215	.83785	.81157	.02628	.00000	.00000	.00000
	PARALLEL	.17081	.82919	.80294	.02626	.00000	.00000	.00000
	PERPENDICULAR	.15348	.84652	.82021	.02631	.00000	.00000	.00000
27.0	TOTAL	.16303	.83697	.81002	.02694	.00000	.00000	.00000
	PARALLEL	.18343	.81657	.78969	.02688	.00000	.00000	.00000
	PERPENDICULAR	.14263	.85737	.83036	.02701	.00000	.00000	.00000
36.0	TOTAL	.16614	.83386	.80651	.02735	.00000	.00000	.00000
	PARALLEL		.79530	.76758	.02772	.00000	.00000	.00000
	PERPENDICULAR		.87243	.84445	.02798	.00000	.00000	.00000
45.0	TOTAL	[illegible]	.82532	.79537	.02995	.00000	.00000	.00000
	PARALLEL	[illegible]	.76756	.73184	.02872	.00000	.00000	.00000
	PERPENDICULAR		.83008	.86121	.02916	.00000	.00000	.00000
54.0	TOTAL		.82424	.77410	.03015	.00000	.00000	.00000
	PARALLEL	.27[illegible]	.72425	.67448	.02977	.00000	.00000	.00000
	PERPENDICULAR	.09577	.90423	.87371	.03052	.00000	.00000	.00000
63.0	TOTAL	.24512	.75488	.72363	.03125	.00000	.00000	.00000
	PARALLEL	.38609	.61391	.58323	.03068	.00000	.00000	.00000
	PERPENDICULAR	.10414	.89586	.86403	.03182	.00000	.00000	.00000
72.0	TOTAL	.35655	.64345	.61155	.03190	.00000	.00000	.00000
	PARALLEL	.52637	.47363	.44252	.03111	.00000	.00000	.00000
	PERPENDICULAR	.18674	.81326	.78059	.03268	.00000	.00000	.00000
81.0	TOTAL	.58986	.41014	.37878	.03136	.00000	.00000	.00000
	PARALLEL	.72874	.27126	.24084	.03042	.00000	.00000	.00000
	PERPENDICULAR	.45098	.54902	.51672	.03230	.00000	.00000	.00000
90.0	TOTAL	1.00000	.00000	.00000	.00000	.00000	.00000	.00000
	PARALLEL	1.00000	.00000	.00000	.00000	.00000	.00000	.00000
	PERPENDICULAR	1.00000	.00000	.00000	.00000	.00000	.00000	.00000

DIFFUSE INCIDENT, SPECULAR ABSORBER

INC ANGLE DEGREES	RADIATION COMPONENT	SYSTEM REFLECTION	SYSTEM ABSORPTION	ABSORBER ABSORPTION	COVER 1 ABSORPTION	COVER 2 ABSORPTION	COVER 3 ABSORPTION	COVER 4 ABSORPTION
DIFFUSE	TOTAL	.22237	.77763	.74873	.02498	.00000	.00000	.00000
	PARALLEL	.29719	.70286	.67423	.02867	.00000	.00000	.00000
	PERPENDICULAR	.14756	.85250	.82370	.02933	.00000	.00000	.00000

DIRECT INCIDENT, DIFFUSE ABSORBER

INC ANGLE DEGREES	RADIATION COMPONENT	SYSTEM REFLECTION	SYSTEM ABSORPTION	ABSORBER ABSORPTION	COVER 1 ABSORPTION	COVER 2 ABSORPTION	COVER 3 ABSORPTION	COVER 4 ABSORPTION
.0	TOTAL	.15596	.84404	.81[illegible]	.02695	.00000	.00000	.00000
	PARALLEL	.15596	.84404	.81[illegible]	.02695	.00000	.00000	.00000
	PERPENDICULAR	.15596	.84404	.81[illegible]	.02695	.00000	.00000	.00000
9.0	TOTAL	.15595	.84405	.81788	.02617	.00000	.00000	.00000
	PARALLEL	.15829	.84172	.81555	.02616	.00000	.00000	.00000
	PERPENDICULAR	.15362	.84638	.82020	.02618	.00000	.00000	.00000
18.0	TOTAL	.15613	.84388	.81734	.02654	.00000	.00000	.00000
	PARALLEL	.16572	.83429	.80778	.02651	.00000	.00000	.00000
	PERPENDICULAR	.14653	.85347	.82690	.02657	.00000	.00000	.00000
27.0	TOTAL	.15713	.84287	.81573	.02715	.00000	.00000	.00000
	PARALLEL	.17971	.82030	.79323	.02707	.00000	.00000	.00000
	PERPENDICULAR	.13455	.86545	.83823	.02722	.00000	.00000	.00000
36.0	TOTAL	.16052	.83948	.81151	.02797	.00000	.00000	.00000
	PARALLEL	.20316	.79684	.76901	.02783	.00000	.00000	.00000
	PERPENDICULAR	.11788	.88213	.85401	.02812	.00000	.00000	.00000
45.0	TOTAL	.16971	.83029	.80132	.02897	.00000	.00000	.00000
	PARALLEL	.24115	.75885	.73012	.02873	.00000	.00000	.00000
	PERPENDICULAR	.09827	.90174	.87251	.02923	.00000	.00000	.00000
54.0	TOTAL	.19226	.80775	.77767	.03007	.00000	.00000	.00000
	PARALLEL	.30198	.69803	.66834	.02968	.00000	.00000	.00000
	PERPENDICULAR	.08254	.91746	.88700	.03046	.00000	.00000	.00000
63.0	TOTAL	.24493	.75507	.72399	.03108	.00000	.00000	.00000
	PARALLEL	.39776	.60225	.57173	.03051	.00000	.00000	.00000
	PERPENDICULAR	.09211	.90789	.87625	.03164	.00000	.00000	.00000
72.0	TOTAL	.36315	.63685	.60519	.03166	.00000	.00000	.00000
	PARALLEL	.54238	.45763	.42670	.03093	.00000	.00000	.00000
	PERPENDICULAR	.18393	.81606	.78346	.03240	.00000	.00000	.00000
81.0	TOTAL	.60406	.39594	.36477	.03117	.00000	.00000	.00000
	PARALLEL	.74282	.25719	.22689	.03030	.00000	.00000	.00000
	PERPENDICULAR	.46531	.53470	.50266	.03204	.00000	.00000	.00000
90.0	TOTAL	1.00000	.00000	.00000	.00000	.00000	.00000	.00000
	PARALLEL	1.00000	.00000	.00000	.00000	.00000	.00000	.00000
	PERPENDICULAR	1.00000	.00000	.00000	.00000	.00000	.00000	.00000

DIFFUSE INCIDENT, DIFFUSE ABSORBER

INC ANGLE DEGREES	RADIATION COMPONENT	SYSTEM REFLECTION	SYSTEM ABSORPTION	ABSORBER ABSORPTION	COVER 1 ABSORPTION	COVER 2 ABSORPTION	COVER 3 ABSORPTION	COVER 4 ABSORPTION
DIFFUSE	TOTAL	.21906	.78010	.75109	.02901	.00000	.00000	.00000
	PARALLEL	.30057	.69949	.67080	.02869	.00000	.00000	.00000
	PERPENDICULAR	.13935	.86071	.83139	.02933	.00000	.00000	.00000

Figure A3. Analysis of 1 Cover System Plus Absorber with Solar Absorptance = 0.90

SOLAR HEAT ABSORPTION, W

NODE NO.	DIRECT	DIFFUSE	TOTAL
1	.19589E 03	.23872E 02	.21977E 03
2	.19589E 03	.23872E 02	.21977E 03
3	.19589E 03	.23872E 02	.21977E 03
4	.19589E 03	.23872E 02	.21977E 03
5	.19589E 03	.23872E 02	.21977E 03
6	.63863E 01	.92416E 00	.73105E 01
7	.63863E 01	.92416E 00	.73105E 01
8	.63863E 01	.92416E 00	.73105E 01
9	.63863E 01	.92416E 00	.73105E 01
10	.63863E 01	.92416E 00	.73105E 01
11	.00000E 00	.00000E 00	.00000E 00
12	.00000E 00	.00000E 00	.00000E 00
13	.00000E 00	.00000E 00	.00000E 00
14	.00000E 00	.00000E 00	.00000E 00
15	.00000E 00	.00000E 00	.00000E 00
16	.00000E 00	.00000E 00	.00000E 00
17	.00000E 00	.00000E 00	.00000E 00
18	.00000E 00	.00000E 00	.00000E 00
19	.00000E 00	.00000E 00	.00000E 00
20	.00000E 00	.00000E 00	.00000E 00

Figure A4. Solar Heat Absorption

TIME .12:00 HR

INTER-NODAL CONDUCTANCES, W/C

NODE NO							
1	.87258E 00	.27437E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.15869E 01	.23721E 02	.33763E 00	.00000E 00	.00000E 00	.00000E 00	
2	.87274E 00	.28253E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.15569E 01
	.15869E 01	.17045E 02	.33763E 00	.00000E 00	.00000E 00	.00000E 00	
3	.87613E 00	.28737E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.15449E 01
	.15869E 01	.16092E 02	.33763E 00	.00000E 00	.00000E 00	.00000E 00	
4	.87966E 00	.29154E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.15649E 01
	.15869E 01	.15688E 02	.33763E 00	.00000E 00	.00000E 00	.00000E 00	
5	.88265E 00	.29503E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.15649E 01
	.00000E 00	.15466E 02	.33763E 00	.00000E 00	.00000E 00	.00000E 00	
6	.00000E 00	.00000E 00	.87258E 00	.27437E 00	.61601E 01	.18094E 01	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
7	.00000E 00	.00000E 00	.87274E 00	.28253E 00	.45541E 01	.19333E 01	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
8	.00000E 00	.00000E 00	.87413E 00	.28737E 00	.43949E 01	.18445E 01	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
9	.00000E 00	.00000E 00	.87366E 00	.29154E 00	.38219E 01	.18533E 01	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
10	.00000E 00	.00000E 00	.88265E 00	.29503E 00	.36304E 01	.18605E 01	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
11	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.00000E 00	.33763E 00	.00000E 00	.00000E 00	.32387E 00	
12	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.00000E 00	.33763E 00	.00000E 00	.00000E 00	.32056E 00	
13	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.00000E 00	.33763E 00	.00000E 00	.00000E 00	.31931E 00	
14	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.00000E 00	.33763E 00	.00000E 00	.00000E 00	.31847E 00	
15	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.00000E 00	.33763E 00	.00000E 00	.00000E 00	.31784E 00	
16	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.23721E 02	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
17	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.17045E 02	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
18	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.16092E 02	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
19	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.15688E 02	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
20	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.15466E 02	.00000E 00	.00000E 00	.00000E 00	.00000E 00	

Figure A5. Internodal Conductance, W/C

The temperatures of each element at each time interval as well as the rate of change of temperatures (Figure A7) are also given in the output.

The final output is an energy balance summary (Figure A8). This is given on an instantaneous as well as on an accumulated basis so that current, accumulated, or average results may be obtained. The energy terms are given by major physical component; e. g. the net rate of energy absorption for the absorber is obtained by summing this parameter for the individual nodal elements of the absorber. The capacitive heat storage is immediately obtained from this summary for all major physical components. The parameter of primary interest, efficiency, is also listed. Efficiency is defined as the rate of energy absorption by the fluid divided by the rate of solar energy incident on the collector.

To provide for the addition of new features and the modification or replacement of such routines as those used to compute heat transfer coefficients, the program was written in a modular fashion. The program consists of a main, executive routine plus 17 subroutines, and is written exclusively in Fortran so that implementation on any computer may be economically effected.

SAMPLE CALCULATIONS

Performance of a flat plate collector exposed to winter and summer solar fluxes is presented in Figures A9 and A10. The solar data -- sun position, direct normal flux, and diffuse flux - were obtained from the ASHRAE Handbook of Fundamentals[4] for 40 degrees north lattitude, a representative latitude in the United States. From this data the incident angle and total flux were computed and are plotted in Figures A11 and A12. February 21 was selected as representative of the available winter solar energy and July 21 was selected for the summer conditions. The collector tilt angle from the horizontal was assumed to be equal to the latitude, 40 degrees, a compromise setting for [illegible] and summer collection. The pertinent physical and environmental parameters [illegible]ed are listed in Table A1.

In each run the collector was assumed to be at ambient temperature at sunrise. The flow of fluid through the collector was assumed to be zero until the temperature of the absorber at the fluid inlet end was 5°C greater than the incoming fluid from storage. The time at which this occurs depends on the collector design. In this example it occurs shortly after 9:00 AM for the winter run and about an hour earlier in the summer run. The fluid flow was assumed to cease in the afternoon when the incoming fluid temperature dropped below the collector temperature. This occurs a little after 3:00 PM in the winter and about 4:30 PM in the summer.

Figure A9, winter performance, indicates that the peak efficiency is 38 percent at noon. This is a satisfactory value considering that the outside temperature assumed was 0°F. The accumulated efficiency (i. e., the efficiency up to any given time) is 30 percent for the day. Note that the accumulated efficiency does not drop appreciably toward the end of the day. This is due to the fact that the amount of solar energy falling on the collector from 2:00 PM until sunset is small compared to the accumulation up to 2:00 PM.

The rate of collection per square foot of collector reaches a peak of 12[illegible] BTU/hr-ft^2 at noon. The integration of this curve gives the daily collected energy, which is [illegible]05 BTU/ft^2.

The summer collector performance, Figure A10, is somewhat better than winter due to the higher ambient temperature, assumed to be 90°F on days on which air conditioning is necessary. The instantantous efficiency is 50 percent, and the daily efficiency is 41 percent. The peak collection rate at noon is 144 BTU/hr-ft^2, and the daily total is 847 BTU/ft^2.

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[4] ASHRAE Handbook of Fundamentals, American Society of Heating, Refrigerating and Air-Conditioning Engineers, 1972.

INTERNODAL HEAT FLOWS, W

NODE NO.

1	-.21444E 02	-.16733E 02	[illegible]	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.64305E 01	-.14704E 03	-.11342E 03	.00000E 00	.00000E 00	.00000E 00	
2	-.52444E 02	-.17114E 03	[illegible]	.00000E 00	.00000E 00	.00000E 00	-.64305E 01
	-.11770E 01	-.13557E 03	-.11042E 03	.00000E 00	.00000E 00	.00000E 00	
3	-.54277E 02	-.17563E 03	[illegible]	.00000E 00	.00000E 00	.00000E 00	-.41770E 01
	.36904E 01	-.13482E 03	-.12364E 03	.00000E 00	.00000E 00	.00000E 00	
4	-.55483E 02	-.18454E 03	[illegible]	.00000E 00	.00000E 00	.00000E 00	-.36904E 01
	.33904E 01	-.13227E 03	-.12793E 03	.00000E 00	.00000E 00	.00000E 00	
5	-.56439E 02	-.19014E 03	[illegible]	.00000E 00	.00000E 00	.00000E 00	-.33904E 01
	.00000E 00	-.12770E 03	-.13029E 03	.00000E 00	.00000E 00	.00000E 00	
6	.00000E 00	.00000E 00	.51546E 02	.16234E 02	-.57999E 02	-.17038E 02	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
7	.00000E 00	.00000E 00	.52446E 02	.17114E 02	-.55014E 02	-.22148E 02	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
8	.00000E 00	.00000E 00	.54277E 02	.17503E 02	-.54646E 02	-.24603E 02	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
9	.00000E 00	.00000E 00	.55483E 02	.18454E 02	-.54726E 02	-.26528E 02	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
10	.00000E 00	.00000E 00	.56439E 02	.19014E 02	-.54654E 02	-.28109E 02	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
11	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.00000E 00	.11342E 02	.00000E 00	.00000E 00	-.11339E 02	
12	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.00000E 00	.11[illegible]E 02	.00000E 00	.00000E 00	-.11346E 02	
13	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.00000E 00	.12364E 02	.00000E 00	.00000E 00	-.12353E 02	
14	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.00000E 00	.12723E 02	.00000E 00	.00000E 00	-.12717E 02	
15	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.00000E 00	.13029E 02	.00000E 00	.00000E 00	-.13022E 02	
16	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.14704E 03	.00000E 00	.22057E 05	-.22204E 05	.00000E 00	
17	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.13557E 03	.00000E 00	.22204E 05	-.22339E 05	.00000E 00	
18	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.13482E 03	.00000E 00	.22339E 05	-.22474E 05	.00000E 00	
19	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.13227E 03	.00000E 00	.22474E 05	-.22607E 05	.00000E 00	
20	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.12770E 03	.00000E 00	.22607E 05	-.22734E 05	.00000E 00	

Figure A6. Internodal Heat Flows, W

TIME =12.00 HR

TEMPERATURES, C, AND RATES, C/HR

NODE NO.	TEMP	RATE
1	.10082E 03	.17069E-01
2	.10483E 03	.38781E-01
3	.10751E 03	.58816E-01
4	.10983E 03	.78926E-01
5	.11178E 03	.98847E-01
6	.41635E 02	.34788E-01
7	.44301E 02	.71068E-01
8	.45558E 02	.97875E-01
9	.46534E 02	.12219E 00
10	.47328E 02	.14430E 00
11	.67232E 02	.41452E-01
12	.69484E 02	.70363E-01
13	.70907E 02	.92363E-01
14	.72151E 02	.11317E 00
15	.73191E 02	.13231E 00
16	.94624E 02	.24550E-02
17	.96923E 02	.10189E-01
18	.99129E 02	.21620E-01
19	.10140E 03	.34795E-01
20	.10352E 03	.51112E-01
FLUID IN	.93330E 02	.82706E-05
FLUID OUT	.10459E 03	.59271E-01
STORAGE		
INLET	.10459E 03	.59271E-01
EXIT	.93330E 02	.82706E-05

Figure A7. Temperature, C, and Rates, C/H

ENERGY SUMMARY

TIME =12:00 HR

	INSTANTANEOUS	ACCUMULATED
	W	W-HR
SOLAR ENERGY		
PHI = 20.60		
INCIDENT	.13668E 04	.51977E 04
ABSORBED		
ABSORBER	.10988E 04	.40204E 04
COVER NO.1	.36553E 02	.14744E 03
REFLECTED	.23143E 03	.10299E 04
HEAT STORAGE		
ABSORBER	.11711E 00	.14966E 03
COVER NO.1	.82177E 00	.11227E 03
INSULATION	.24285E-01	.10362E 02
FLUID IN COLLECTOR	.26136E-01	.72751E 02
FLUID IN STORAGE	.67740E 03	.19322E 04
HEAT LOSSES		
EMISSION FROM TOP COVER	.11843E 03	.47988E 03
CONVECTION FROM TOP COVER	.27719E 03	.11402E 04
EMISSION AND CONV. FROM INSULATION	.61377E 02	.27046E 03
ENERGY BALANCE		
INCIDENT SOLAR - REFLECTED SOLAR	.11354E 04	.41678E 04
COLLECTOR STORAGE +		
FLUID IN STORAGE +		
HEAT LOSSES	.11354E 04	.41678E 04
EFFICIENCY		
FLUID IN STORAGE/INCIDENT SOLAR	.49560E 00	.37175E 00

Figure A8. Energy Summary

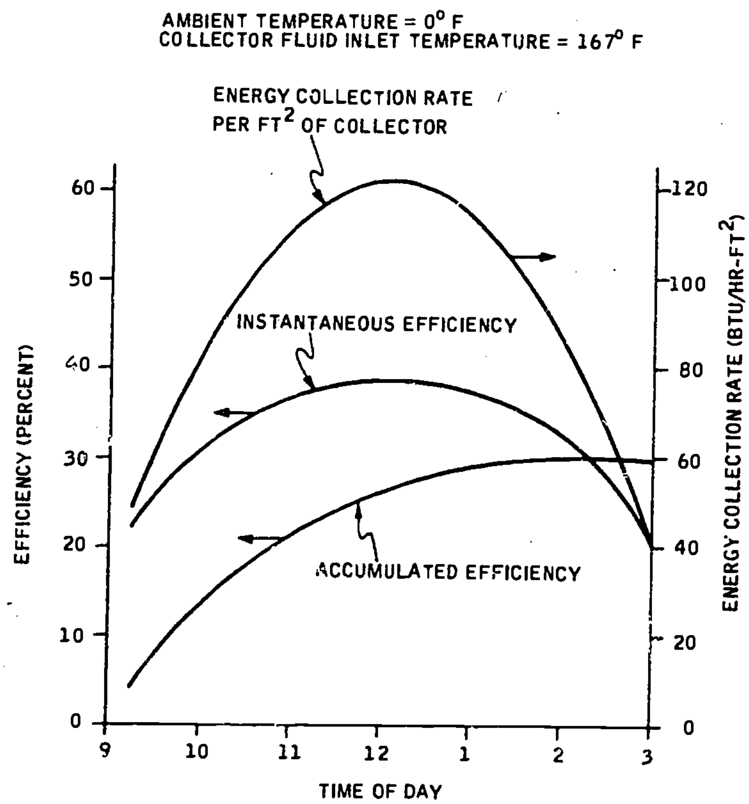

Figure A9. Flat Plate Collector Performance at 40°N. Latitude on 21 February

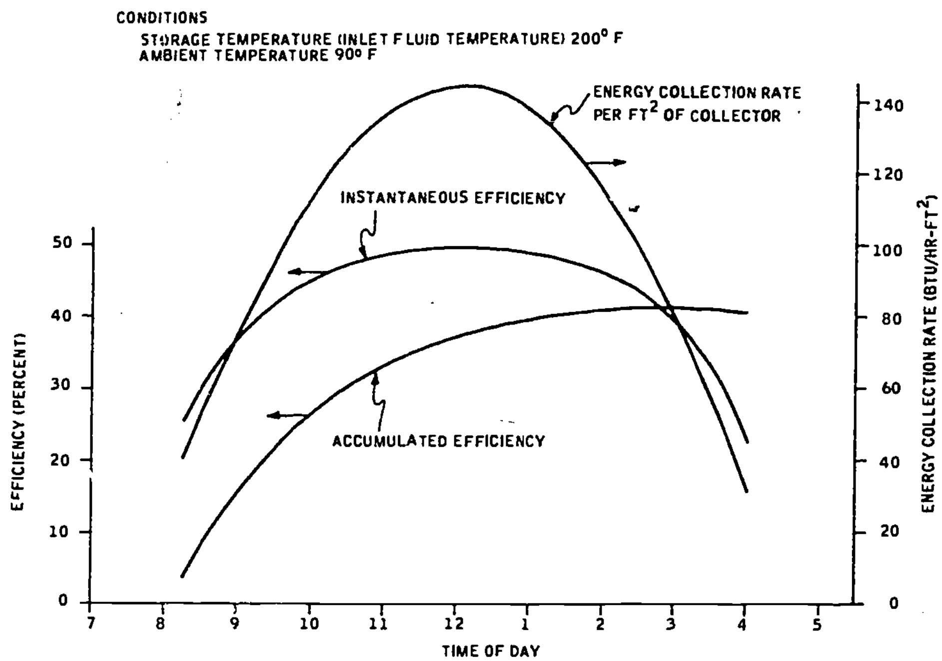

Figure A10. Flat Plate Collector Performance at 40° N. Latitude on 21 July

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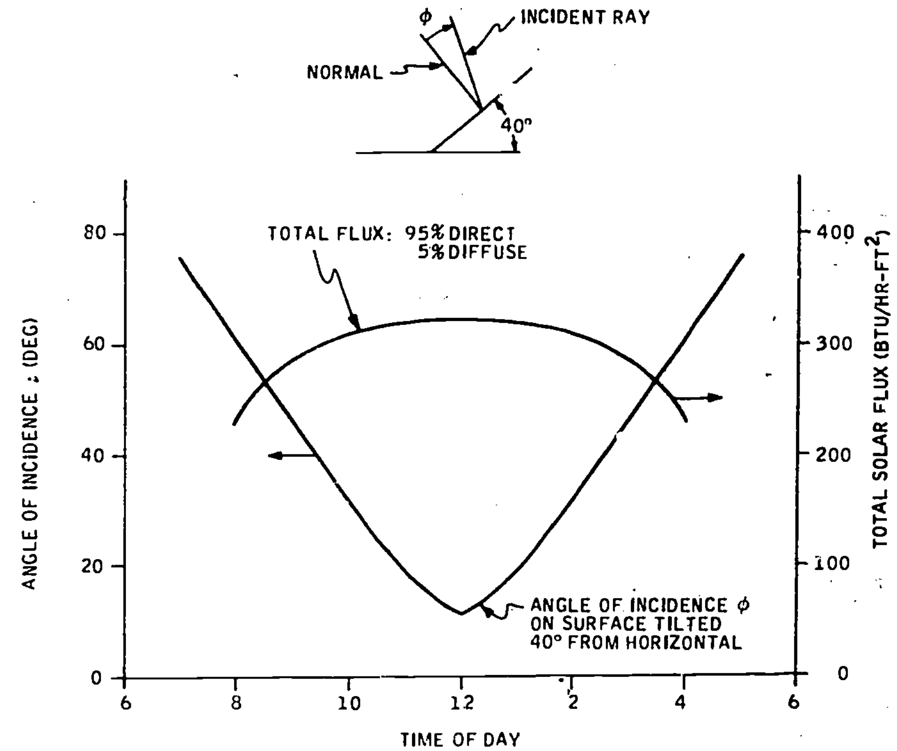

Figure A11. Solar Flux and Angle of Incidence at 40° N. Latitude on 21 February

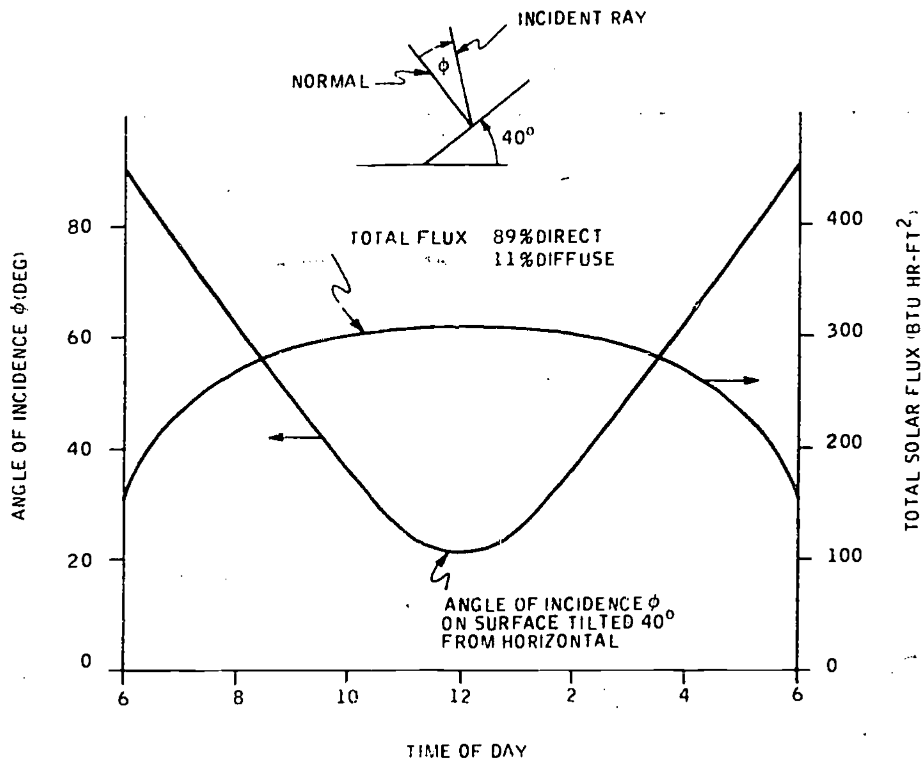

Figure [illegible] Solar Flux and Angle of Incidence at 40° N. Latitude on 21 July

Table A1. Physical Conditions for Flat Plate Collector Calculations

Collector	
Length:	4.0 ft
Width:	4.0 ft
Area:	16.0 ft^2
Cover (One)	
Glass	
Thickness:	0.125 in.
Index of Refraction:	1.526
Extinction Coefficient:	0.2 in^{-1}
Absorber	
Material:	Aluminum tube in sheet
Thickness:	0.060 in.
Tube Diameter:	0.375 in., 12 tubes on 4-in. centers
Insulation	
Material:	Fiberglass
Thickness:	3.0 in.
Ambient Conditions	
Temperature:	-20°F to 60°F for heating; 90°F for cooling
Pressure:	1 ATM
Mean Radiant Temperature of Sky:	Same as ambient temperature
Wind Velocity:	7.5 mph in summer; 10.0 mph in winter
Collection Fluid	
Material:	Specially inhibited 50/50 ethylene glycol and water
Flow Rate:	0.25 gpm

APPENDIX B

FLOW DISTRIBUTION IN SOLAR COLLECTOR ARRAYS

A typical solar collector installation will consist of a large number of flat plate collector modules assembled as an array on or in the structure roof. Collection fluid will be supplied to the modules in some form of series-parallel network. The piping network must be designed to provide the proper flow to each module. Also, within a module, the goal is to provide uniform flow per unit area of the collector. In the usual case of uniformly spaced tubes running from a supply header to a collection header as indicated on Figure B1, it is desirable to provide, as nearly as possible, equal flow to each tube. This latter problem, uniform flow within a module, is addressed in the following paragraphs.

Flow Analysis

The flow in a tube is determined by the difference in pressure between [illegible] pply header and the collection header at the tube ends. The header pressures wi [illegible] g their length due to (1) wall friction losses and (2) momentum flux changes [illegible] rs as fluid is withdrawn or added. The pressure in the supply manifold w [illegible] flow direction due to wall shear stress but the drop will be reduced by pres [illegible] recovery at each cross-tube due to extraction of fluid and subsequent loss of momentum in the supply header flow. In the collection header the frictional drop is reinforced by the acceleration drop due to the increased momentum. The resulting pressure difference at the ends of a tube will, therefore, vary from one tube to the next and the flow rate will vary from tube to tube.

Header Pressure Drop Due to Wall Shear

Pressure drop in the flow in a header between adjacent tubes depends upon the flow rate and the flow condition, e.g., laminar or turbulent, developing or fully developed. Since only a small percentage of the header flow is extracted or added at each cross-tube, the header flow is assumed to be fully developed at all locations. The flow is assumed to be laminar for Reynolds numbers below 2000 and turbulent above 2000.

Cross-Tube Pressure Drop

The above discussion applies equally well to the cross-tube pressure drop. However, the flow in a cross-tube will almost always be in the laminar regime.

Pressure Changes due to Removal or Addition of Flow at Branches

Fluid is removed from the supply header and added to the collection header at each cross-tube branch.

Taking into account the above factors, the flow distribution in the tubes is calculated. This analysis has been programmed for solution on a digital computer with an on-line plotter. Results have been obtained for a row of ten collectors as indicated on Figure B1. This number of collectors was chosen as typical. A complete array would consist of several rows. The present analysis is concerned with the distribution of flow in one row. The pressure drop in the supply and collection headers in the collectors can be reduced by providing a central supply manifold as indicated on Figure B1 with the flow passing both ways to the outer edge of the array. This produces a significant improvement in the flow uniformity between tubes.

The analysis has thus been carried out for a row of five collectors each having twelve cross-tubes for a total of 60 tubes as shown on Figure B1. The supply and collection headers are on 42-inch centers and the cross-tubes are 40 inches long on 3.552 inch

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centers. The headers are rectangular ducts with 1/2 inch by 2-inch inside dimensions. The cross-tubes are also rectangular in section and will have 0.050 inch by 0.500 inch inside dimensions. These spacings and dimensions have been arrived at from the results of the present analysis.

Figures B2 to B4 present the results of the flow distribution calculations for three possible cross-tube dimensions, 0.050 inch by 0.500 inch, 0.100 inch by 0.500 inch and 0.200 inch by 0.500 inch. The sensitivity of the flow distribution to the cross-tube dimensions is immediately evident. Referring to Figure B2 for the 0.050 inch by 0.500 inch tube it is seen that the cross-tube pressure drop at a nominal flow of 10 lbm/hr is about 2.5 inches of water. This is a relatively large pressure difference compared to the header pressure variation and, consequently, the cross-tube flow distribution is quite uniform.

As the cross-tube inner height is increased the associated pressure drop falls rapidly and, as shown in Figure B3 for the 0.100 inch by 0.500 inch tubes, the header pressure variation causes a significant non-uniformity in the flow distribution. The distribution for the 0.200 inch by 0.500 inch tubes, Figure B4, is badly out of balance.

Increasing the cross-sectional dimensions of the headers results in less header pressure variation and more uniform flow. However, the mass of fluid and collector heat capability increase as the header dimensions are increased. For 0.5 inch by 2.0 inch headers and 0.050 inch by 0.500 inch cross-tubes the fluid heat capacity is about 25 percent of the total collector heat capacity and 90 percent of the fluid is in the two headers.

It should be noted that if a non-uniform flow is present, the resulting variation in the fluid temperature will produce a variation in the hydrostatic pressure distribution which acts in a direction to reduce the non-uniformity. Since the hydrostatic pressure of a 40-inch column of a 50 percent mixture of ethylene glycol and water decreases 0.016 inches of water per °F, the temperature variation will produce a significant improvement in the flow distribution for the 0.200 inch by 0.500 inch tubes, but very little improvement for the thinner tubes.

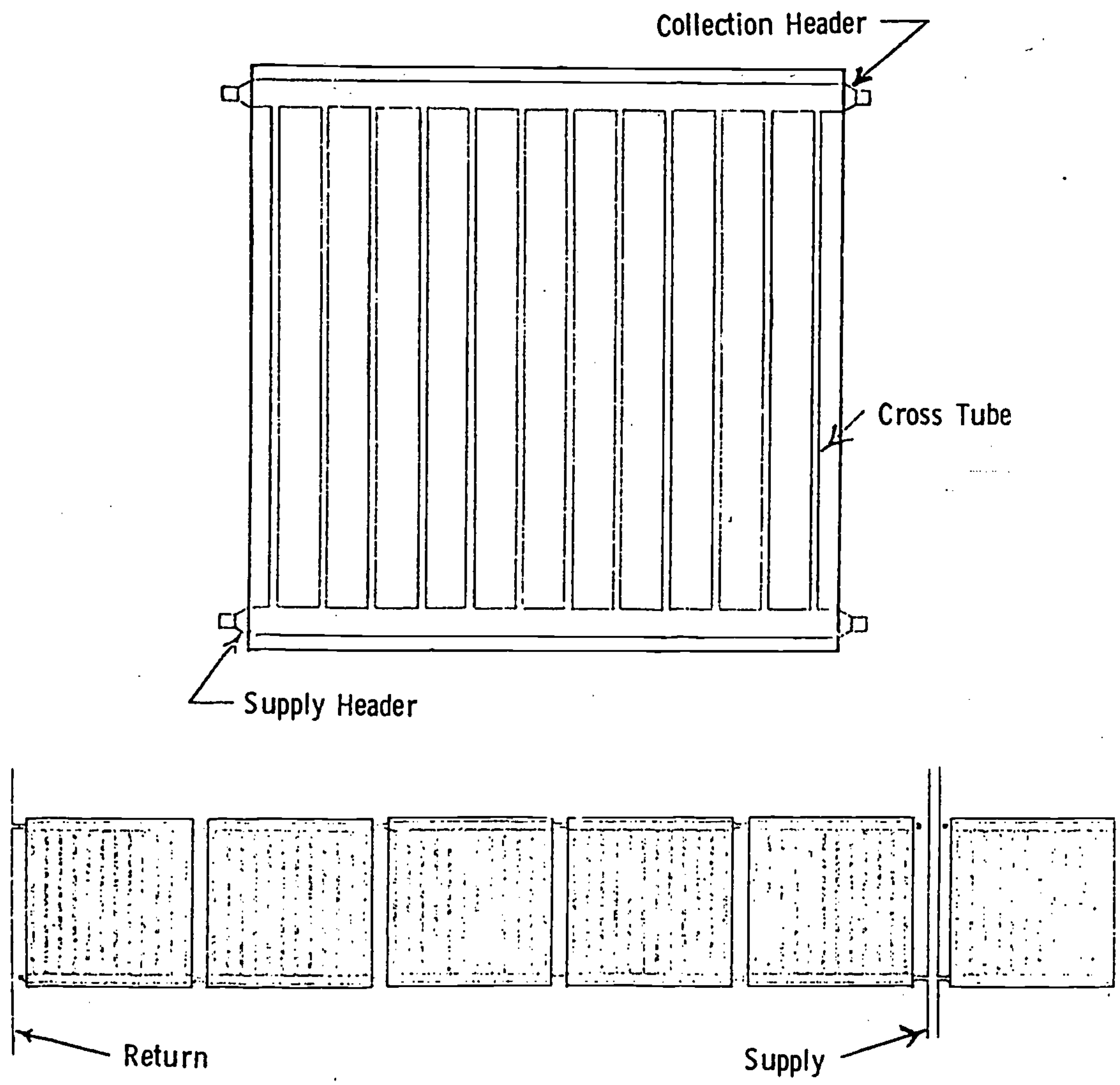

Figure B1. Schematic of Individual Collector and Potential Array Configuration

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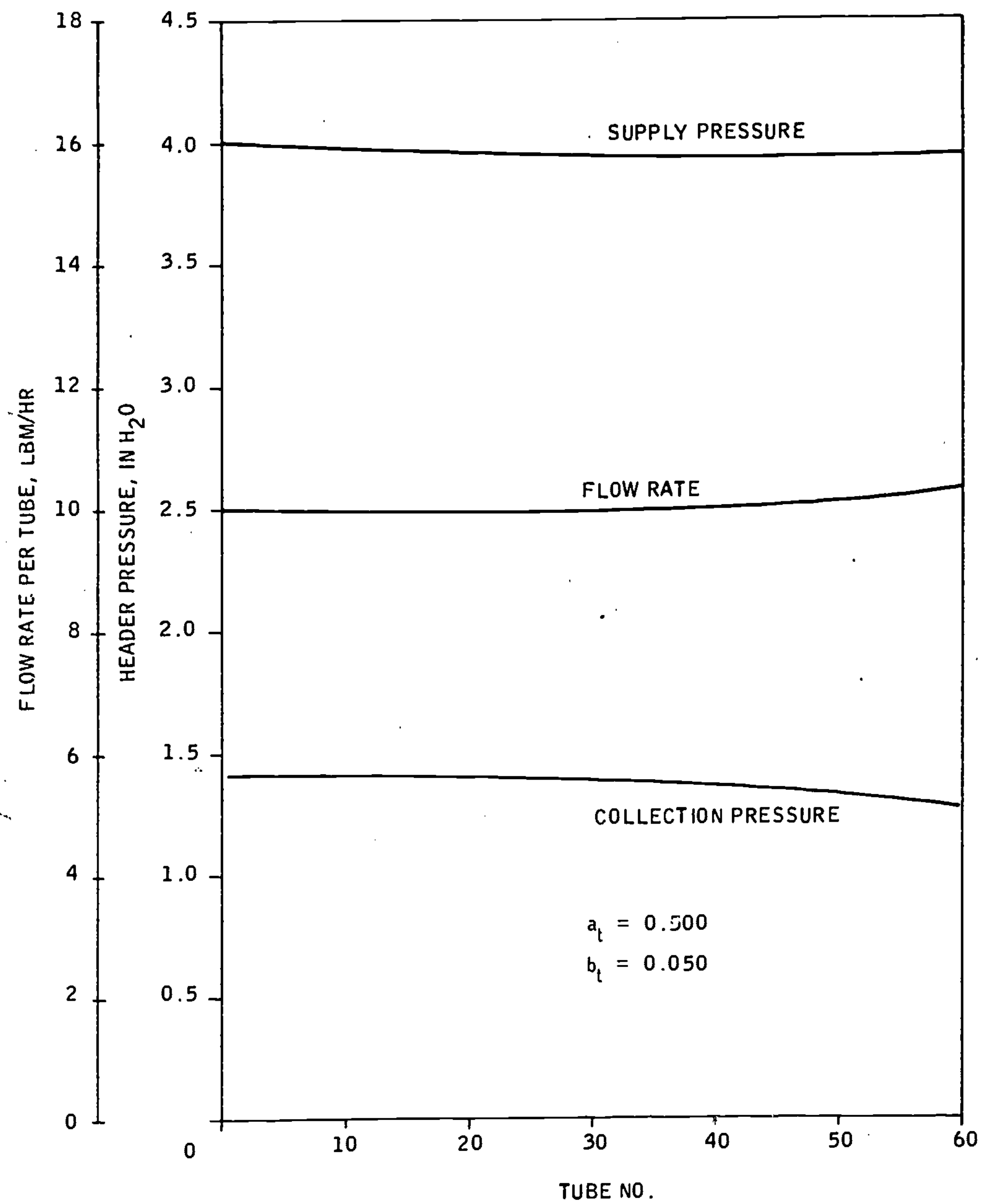

Figure B2. Flow Distribution for 0.05 Inch by 0.50 Inch Cross Tube

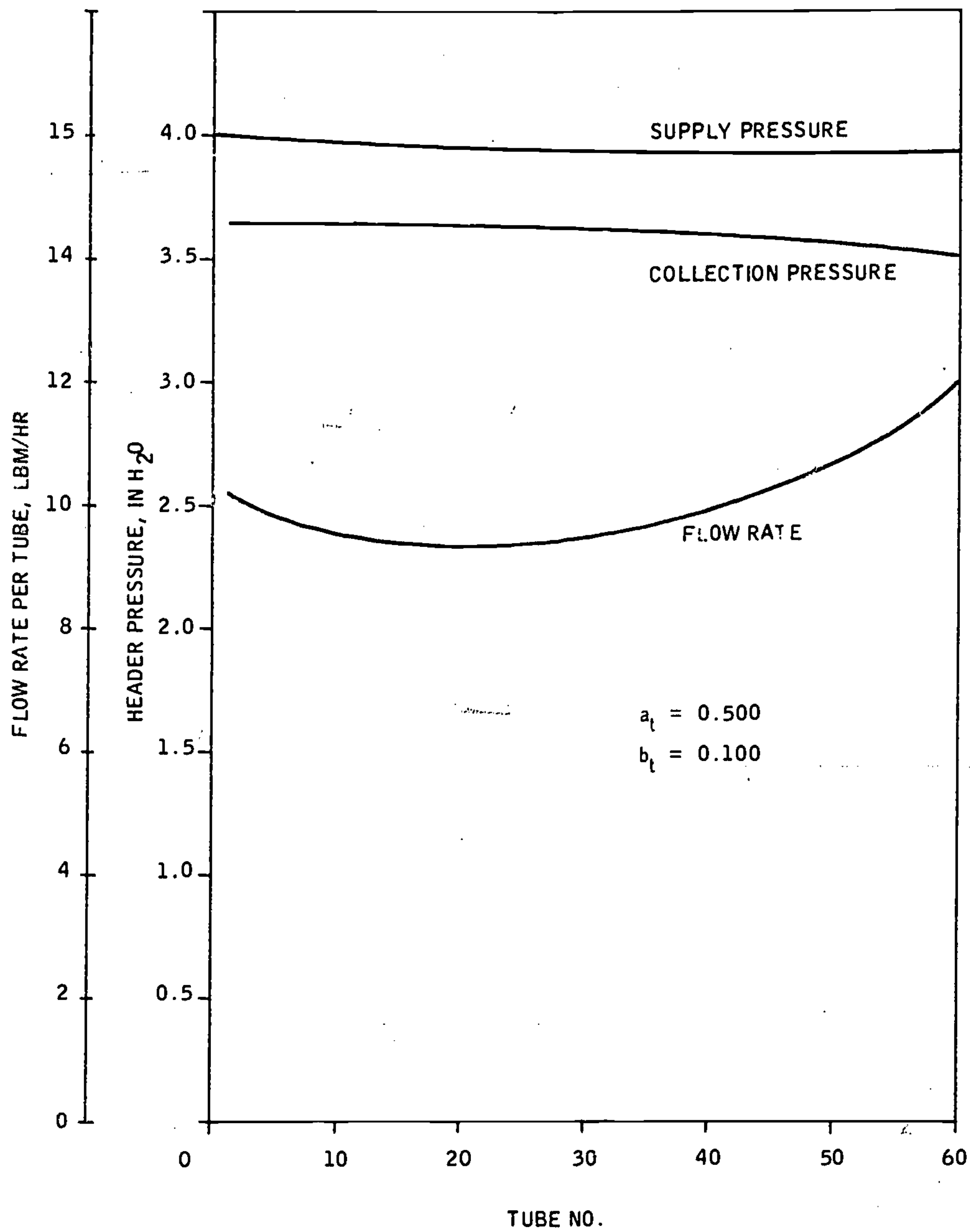

Figure B3. Flow Distribution for 0.1 Inch by 0.5 Inch Cross Tube

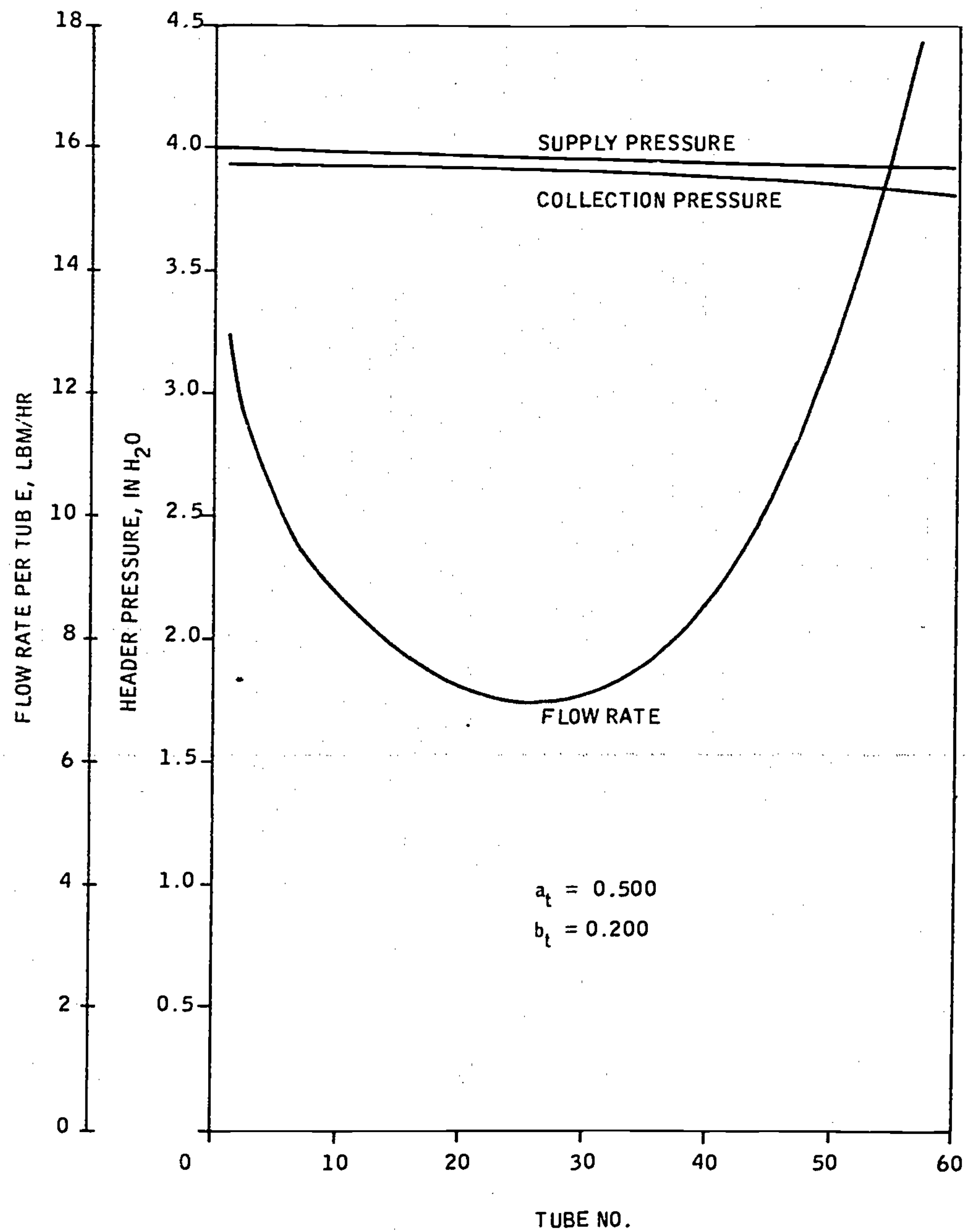

Figure B4. Flow Distribution for 0.2 Inch by 0.5 Inch Cross Tube

APPENDIX C

COATINGS, COVERS AND INSULATION

Since this program was conducted in a very short time, the choice of material was influenced greatly by what was currently available. For instance, sheet steel was selected for the module housing because it was abundant and there were local fabricators in the area. Fiberglass insulating material, glass, and tedlar were also readily available locally or from out-state distributors.

This appendix describes the tradeoff analyses accomplished in selecting the collector module components for the school.

ABSORBER COATINGS

The absorbing coating for a flat plate collector may be either selective or non-selective. Collector performance would be about the same with either type coating at low temperatures, but the nonselective coating may offer cost and possibly durability advantages. If the collector application is for heating and cooling, then a selective coating would be preferable. Increased collector performance will more than offset the higher cost of high-performance selective coatings.

For a heating and and cooling system the primary requirements for the absorber coating are high optical efficiency (high solar absorptance, α, and low infrared emittance, ϵ), low cost, and satisfactory environmental durability. A list of some properties of coating-substrate systems for flat plate collectors is shown in Table C1. Among the coatings shown, Honeywell has experience with the three with highest optical efficiency: black nickel (Ni-Zn-S), black chrome (CrO_x), and CuO. These coatings are discussed below followed by a discussion of possible substrates.

Table C1. Performance Values for Some Solar Absorber Coatings

Coating	Substrate	α	ϵ(at T°C)	Breakdown Temperature (°C)	Reference No.
Ni-Zn-S	Ni	0.96	0.07 (100)	280	Honeywell
CrO_x	Ni	0.96	0.12 (100)	450	Honeywell
FeO_x	Fe	0.85	0.10 (40)	?	1
CuO	Cu	0.90	0.14 (20)	200	2
CuO	Al	0.93	0.11 (80)	200	3
PbS/Silicone Paint	---	0.94	0.4 (?)	>350	4,8

Black Nickel

Black nickel is a nickel-zinc-sulfur complex which can be applied to many substrates by an electroplating process. This coating achieves high solar absorption through the combined effects of interference and absorption and is transparent in the infrared (2-20 μm) so that a low emittance metal substrate will show through in that region.

Honeywell's preliminary durability tests on black nickel indicated it could withstand one week in air at approximately 550°F. ~1/3 sun years of ultra violet, and the equivalent of 40 years of thermal cycles from room temperature to ~220°F. Therefore, a program to improve the optical efficiency of the coating was initiated in which the effects of bath composition, temperature, and pH and plating current densities and times were evaluated. It was found that the composition could be altered so that the maximum effect of a natural absorption of the coating in the solar wavelengths and an optical interference effect could be obtained. These studies enabled us to improve the coating absorption from ~ 86 percent (typical of industrial plating job shops) up to ~96 percent while achieving an emittance of 7 percent at 200°F. The spectral reflectance of such a coating is shown in Figure C1.

Honeywell's investigation of black nickel coatings included process scale-up to 3 x 4 foot panels, evaluation of long-term bath degradation, and optical reproducibility. The scale-up was successful except for a tendency for color fringes to form near the edges of large panels due to optical interference effects. This problem does not significantly affect performance and can be minimized with careful placement of the panel and anodes during the electroplating. The bath itself was found to be remarkably stable over 5 months of use during which ~150 panels were plated. Under constant use it was necessary only to adjust pH every other day and maintain a critical thiocyanate ion concentration every week or two.

All panels that were measured (~15) had ≥ 94 percent solar absorptance with emittance less than 10 percent at 200°F.

There have, however, been variations in panel resistance to the combined effect of thermal and humidity cycling. A test we have regularly used follows the procedure of MIL-STD-810B, Method 507, Procedure I. This test consists of a thermal and humidity cycle, from room temperature to 71°C (160°F) at 95 percent RH and from 71°C to room temperature at >85 percent RH, over a 24-hour period. This is a very severe, accelerated environmental test. The test conditions impose a vapor pressure on the panels which constitutes the major force behind moisture migration and penetration. Some coated panels have survived over one week under this test, while others have completely corroded after one day. Some coating parameters which may be important to humidity resistance include:

- Low thiocyanate concentration
- "Old" (oxidized) bright Ni substrates
- Pitted Ni substrate (galvanic cell problems)

There may, however, be other parameters not yet identified.

A possible solution to the humidity-induced corrosion problem might be the use of humidity-resistant, silicone-based coatings which can be applied over the solar absorber coating. These coatings have high-temperature stability, do not greatly increase the overall emittance values, and in most cases increase the solar absorptivity due to their low refractive index. These coatings provide a degree of corrosion protection, but it is now known if they lead to a significant long-term improvement.

Black Chrome

Black chrome is a commercial electroplated chrome oxide coating with diffuse reflectance properties. Manufacturer data indicates that the coating remains black to temperatures of 900°F in air. Significantly, a Honeywell black chrome coating showed no change in optical properties after one week in the MIL-STD-810B humidity test. No UV, thermal cycle, or other durability tests have been performed at Honeywell.

We have briefly studied the effects of current density, bath temperature, and plating times on the optical performance of black chrome. Our best black chrome coating had an α of 96 percent with ϵ (200°F) of 12 percent (on Ni substrate).

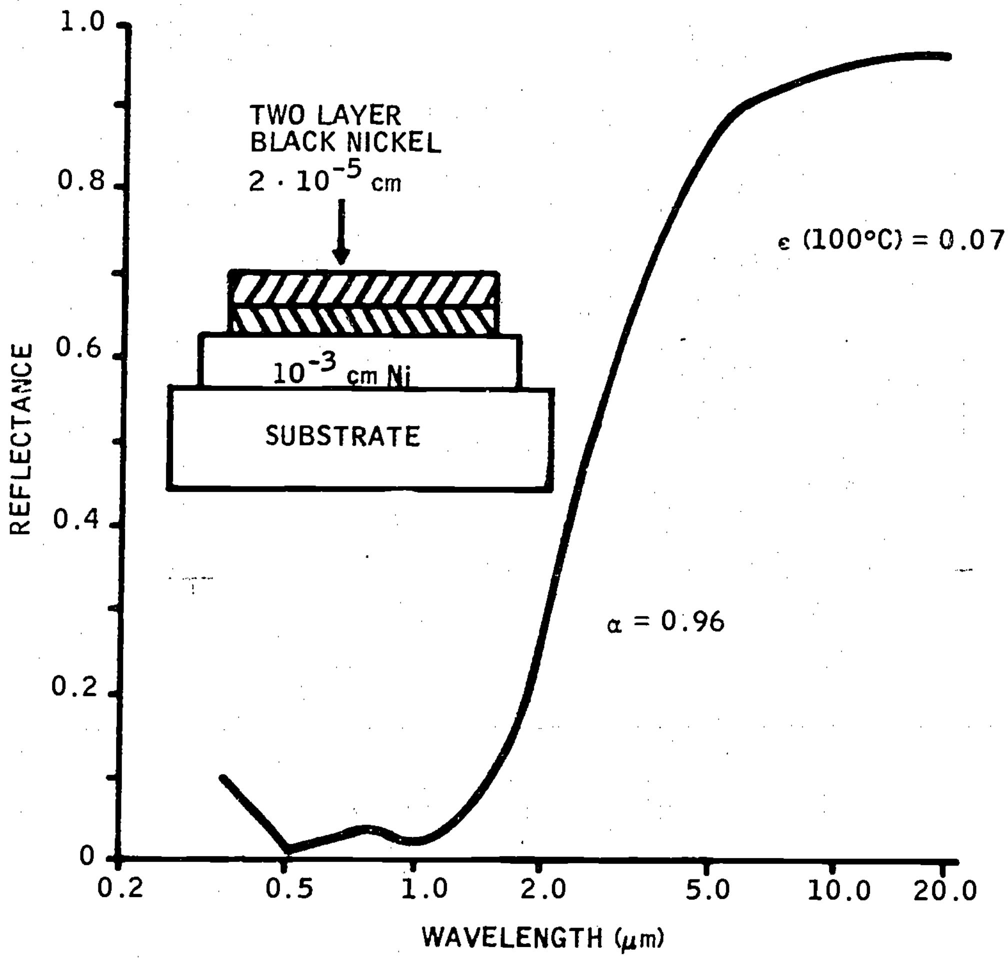

Figure C1. Spectral Reflectance

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Copper Oxide

Our experience with CuO coatings is rather limited since the performance achieved in early studies could not greatly improve upon literature values of $\alpha = 0.90$ and $\epsilon = 0.20$. More work with this chemical-dip type coating is justified, however, due to its relatively low cost.

Substrates

The primary optical requirement of the substrate* is to provide a surface with low infrared emittance. The material also must not easily corrode since the thin black absorber coating provides little protection.

Aluminum, zinc (galvanized steel), and copper provide substrates somewhat stable to corrosion when oxidized but have fairly high emittance under those conditions (greater than 10 percent). Nickel has often been used since it forms a stable coating for several metals and results in a low emittance (~ 0.07) substrate.

Honeywell experience has been primarily with Ni coated steel substrates. Steel was selected because of its low cost, high strength, ease of fabrication, and compatibility with electroplated Ni. The requirements for the Ni layer are quite severe, since any pores or pin-holes through the Ni will quickly lead to corrosion of the steel (due to galvanic coupling) and subsequent failure of the panel circulation system. A straightforward solution to the problem, i. e., using very thick Ni layers (greater than 2 mil), leads to cost penalties ($8¢/\text{mil ft}^2$ for Ni alone).

Since the greatest amount of experience had been obtained in the use of a black nickel coating over a bright nickel substrate it was decided to use this process as a selective coating. In addition it was anticipated that solar cooling experiments requiring higher collector temperatures would require a selective coating on the collectors.

TRANSPARENT COVERS

Solar collector cover design starts with two basic questions: (1) what cover material to use and (2) how many covers to use. Since material choice is somewhat dependent on cover configuration, the actual number of covers should be considered first.

Recent testing on a selective black nickel collector module with one or two glass covers has produced data which relates collector efficiency with one or two glass covers as a function of input conditions, i. e., average fluid temperature and incident flux level.

This data has been graphed and is shown in Figure C2.

As can be seen from the graph, better collector efficiency can be achieved by using two glass covers for applications requiring a high fluid temperature, such as cooling. For heating applications, i. e., those only requiring fluid temperatures around 140°F, the choice is, at best, marginal. However, if the same collector is to be used year around, then the two-cover configuration is preferable.

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*Substrate here refers to the surface on which the absorber coating is deposited. It can be the same as the bulk substrate material or it can be a thin layer of material plated onto the bulk substrate.

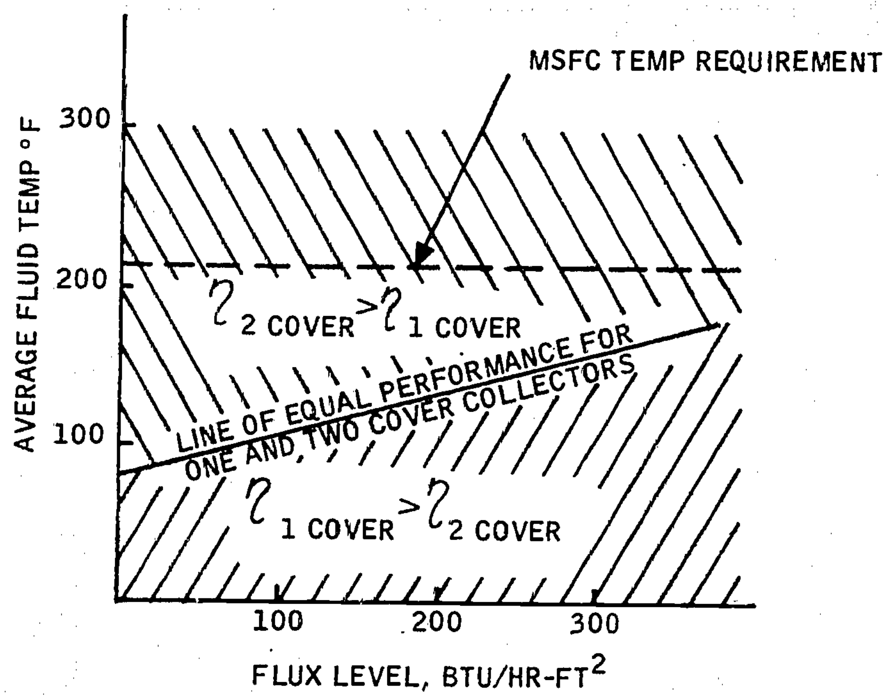

Figure C2. Performance Preference Curve

The choice of materials is constrained by several factors: transmission factor, resistance to UV degradation, mechanical strength, weight, and cost. Judicious use of materials in a two-cover system can, however, adequately overcome many of the constraints. Materials presently being used include: glass, tempered and antireflection coated; polyvinyl fluoride such as Tedlar; polycarbonates such as Lexan; and polyesters such as Mylar. Most other plastics tend to degrade with exposure to UV, and when UV inhibiters are used, the transmission is degraded. Also, most plastics are expensive relative to glass except in very thin sheets. Lexan is highly shatter-resistant and has fairly good transmission, in the range of 84.5 to 88 percent. Tedlar is of more interest in that it weathers well, has a transmission greater than 90 percent, and is projected to have a low price in large quantities. Its major disadvantage is the need for a supplementary support system to form a mechanically sound external cover as it fails in fatigue. This increases cost and reduces the effective transmission value.

A glass cover achieves good transmission, generally in the range 85 to 88 percent, and in the case of thin low iron glass as high as 91 percent, is self-supporting, weathers well, and is moderately priced. It has two primary disadvantages: it is heavy and tends to shatter under the force of a well-aimed projectile. Shatter-resistance can be improved by using tempered glass, but this again increases cost. Another refinement relevant to glass as a material choice is the new process of glass etching to minimize reflections. Glass subjected to the etching process can produce a transmission factor of 97 percent, but surface etching increases the collector cost.

A reasonable compromise of constraints can be achieved by combining a glass outer cover and a Tedlar inner cover. This combination significantly reduces the weight of the cover system while maintaining mechanical integrity.

In anticipation of favorable performance results, an estimate of the cost of this cover combination has been assembled. In production quantities of 100,000 ft^2/year, a cover system consisting of a double-strength, tempered glass outer cover and a Tedlar inner cover is estimated to cost $\$1.01/ft^2$. This includes appropriate brackets and supports to fasten the covers to the collector housing.

HOUSING

The initial design consideration for a choice of solar collector housing is, in fact, whether or not to have a housing for each collector. The two candidate approaches are (1) to create collector modules, each containing one or more absorber panels, each module completely enclosed by a housing and cover, with discrete inlet and outlet plumbing either internal or external to the box; and (2) to create a collector array, composed of a large backing plate covered with insulation, all the absorber panels mounted on the plate and insulation and plumbed together, a frame enclosing the edges of the backing plate and supporting a cover system that covers the entire collector array.

Both candidate approaches have distinct advantages yet both also have drawbacks significant enough to warrant a detailed examination of their impact on each specific collector application and its installation requirements.

Recently, an experiment was performed to isolate heat loss components on the solar collector designed for their use, a 4 x 4 ft aluminum absorber enclosed in a sheet metal housing and two glass covers. By iterating the housing design and measuring the resultant heat loss, the more significant heat loss factors were readily determined. Heat loss through the sides of the housing was shown to be a major factor in poor collector performance. This loss component may be limited by reducing the heat path from the absorber to the housing, i.e., increase the insulation thickness, eliminate mechanical absorber supports that connect to the housing, or move the sides of the housing away from the absorber panel.

Reflecting this experience back to the two candidate housing approaches, the collector array appears to have an inherent advantage in large collector installations since only the outside rows of the array have housing edges in close proximity, so edge loss should be reduced. Of course if the collector module approach is used, edge losses can also be reduced by designing the housing sufficiently larger than the enclosed absorber panel to accommodate several inches of insulation.

Apart from heat loss, the other major concern for evaluation of the candidate approaches is installation and maintainability once installed. The collector array lends itself readily to integration into the roofline of buildings, perhaps using the existing roof as a backing plate and thus reduces to merely constructing a frame to support the cover system. A modular collector installtion, however, would probably use a separate supporting framework. This makes it well suited for ground installations and building installations where it is either not feasible or simply undesirable to utilize an existing roof.

Actual installation and subsequent collector maintenance is where the collector module approach shows a significant advantage. The collector array must be mounted from above, necessitating cranes or scaffolding; maintenance to the installed system must also be accomplished from above, unless parts are made in the backing plate, and that severely challenges the weatherproof integrity of the roof. Furthermore, if the cover sections are reasonably large, a disproportionate amount of effort must be expended to uncover and reach minor repair items, such as a leaking connection.

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*Based on June 1974 material and labor costs.

The module approach, when mounted on a supporting frame, enables easy manual installation on top of the building, but more important, maintenance is readily accomplished by either disconnecting and bypassing a particular module or by working through access ports in the back of the housing. Here, of course, the actual building roof is not affected by collector maintenance requirements. Since it is quite reasonable to expect fairly extensive system maintenance problems on new technology installations, such as solar collection systems, ease of installation and maintainability should be a significant consideration in choosing housing design.

Regardless of which housing system is chosen, the materials problem remains the same. The housing must be physically sound, durable, weather-tight, relatively light, nonflammable, and aesthetically appealing(or at least neutral), and reasonably priced. Investigations of alternate materials have failed to indicate a plastic of reasonable cost that has sufficient durability and is not a fire-hazard; however, further investigations must be performed before eliminating plastics. Wood or wood composites have also been considered. While appearing to have sufficient strength, heat resistance, and marginal durability, wood housings have been at least temporarily discarded as not being cost effective, particularly in large production quantities. At present the most likely candidate housing material is sheet steel. It possesses the necessary strength, durability (particularly when galvanized), work-ability, and is cheaper than other potential metals. Its primary disadvantage is weight. Overall, however, mild steel appears, at present, to have the most potential as the best housing material, although the introduction of formable wood composites, such as Blandex, will require further investigation.

To provide a baseline for cost analysis of potential housing designs, an estimate has been assembled for the cost of producing sheet metal housings for a modular collector. If produced in quantities of 100, 000 ft^2/year, it is estimated that housings could be produced at a cost* of \$0. 69/ft^2.

Alternate designs are certainly feasible but will require further investigations to determine if they can be euqally cost effective.

INSULATION

The back surface of the collector absorber plate will be thermally insulated to minimize the amount of heat lost to the collector housing. In the construction industry, many insulation materials are available that could be utilized in the collector assembly for reducing heat losses. A list of a few thermal insulations is presented in Table C2. The federal specification numbers are included for reference.

In addition to reducing the heat loss during normal operation, the insulation must be selected or designed to withstand the high temperature which will occur under conditions of no heat removal (e. g., no flow of the heat transfer fluid). The absorber plates of well designed collectors can reach temperature in excess of 400°F for a condition of no heat removal. This high temperature imposes severe restrictions on the use of plastics and foam insulations. For example, the maximum temperatures allowed for urethane and polystyrene listed in Table C2 are 225°F and 165°F, respectively.

The angle of the collector assembly dictates that the insulation used should not settle or compact near the bottom, which is the case with loose or poured insulations. The settling of the insulation would decrease the efficiency of the collector by increasing the heat loss from the absorber plates. A loose insulation would also not be desirable during repair or maintenance operations.

*Based on June 1974 labor and material costs.

Table C2. Thermal Insulations

1.	Thermal insulation blanket (cellulose)	HH-I-515B
2.	Thermal insulation blanket (mineral fiber)	HH-I-521E
3.	Thermal insulation board (polystyrene)	HH-I-524B
4.	Thermal insulation board (mineral fiber)	HH-I-526C
5.	Thermal insulation board (urethane)	HH-I-00530
6.	Thermal insulation board (urethane)	HH-I-530A
7.	Thermal insulation board	LLL-I-535
8.	Thermal insulation board	LLL-I-535a
9.	Thermal insulation board (cellular glass)	HH-I-551D
10.	Thermal insulation black (asbestos)	HH-I-561a
11.	Thermal insulation (perlite)	HH-I-574a
12.	Thermal insulation (vermiculite)	HH-I-00585a
13.	Thermal insulation (vermiculite)	HH-I-585B
14.	Thermal insulation (mineral fiber)	HH-I-1030A
15.	Thermal insulation (aluminum foil)	HH-I-1252A

The lowest cost insulation available today is fiberglass, which is produced in a variety of densities (affecting thermal conductance) and a variety of binder conditions depending on the application. Manufacturers of regular building fiberglass insulation with a bakelite binder specify an upper use temperature of 350°F. When first heated above 350°F, the binder burns, giving off odor and fumes. If the fumes encountered in this burning of the binder are not objectionable, the material can be used to 700°F. The insulating value is not degraded at the higher temperatures provided the material does not become compressed.

Fiberglass insulation with little or no binder is made specifically for higher temperature applications. Usually there is a small amount of binder which is allowed to burn off during the first heating of the insulated device.

The binder residue could collect on the collector covers which would degrade the collector performance. To alleviate this possible problem area, a design consisting of two types of insulation could be used. Immediately behind the absorber plate a high temperature, thin sheet of insulation would be used. An additional layer of low cost insulation would complete the design to give the desired thermal resistance to heat flow.

APPENDIX D

CALCULATIONS OF ENERGY, EFFICIENCY AND SYSTEM PERFORMANCE PARAMETERS

MODE 1. COLLECTOR TO FRESH AIR HEATER

Solar Flux (Leeds and Northrop Integrator)

- Instantaneous Flux = I (Btu/Hr - Ft^2)
  Full Scale = 100 divisions = 2 cal/cm^2 - min = 442 Btu/Hr-Ft^2
  I = Number of divisions x 4.42 Btu/Hr-Ft^2
- Integrated Flux = Φ (Btu/Hr-Ft^2)
  Φ = Number of Traverses/Hour x Conversion Factor 7.367 Btu/Hr-Ft^2

Solar Energy Input

Q incident = Φ x Net Collector Area A(Ft^2) Btu/Hr

Useful Solar Energy Collected

Q collected = Fluid Flow Rate G (gpm) x 60 Min/Hr x

- Fluid Density ρ (lbs/gallon) x
- Specific Heat of Fluid Cp (Btu/lb°F) x
- Temperature rise across the Collector
  $(T_2 - T_1)$ Btu/Hr

System Efficiency --

η = Q Collected/Q Incident

Fresh Air Coil -- Heat lost by fluid:

$Q_{L,F}$ = Fluid Flow Rate G (gpm) x 60 Min/Hr x
Fluid Density ρ(lbs/gallon) x
Specific Heat of Fluid Cp (Btu/lb°F)

- Temperature difference $(T_8 - T_9)$ °F Btu/Hr

Heat gained by Air

$Q_{G,A}$ = Air Flow Rate Ft^3/Min x 60 Min/Hr

- Air Density ρ_a (Lbs/Ft^3) x
  Specific Heat of Air $C_{p,a}$ (Btu/lb°F)
- Temperature difference $(T_4 - T_3)$°F Btu/Hr

Note air density $\rho_a = 0.0865 \times \dfrac{\text{Barometric pressure in Hg}}{29.92} \times \dfrac{460}{\dfrac{T_3 + T_4}{2} + 460}$

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MODE 2. STORAGE TO FRESH AIR HEATER COIL

Fresh Air Coil equations - same as Mode 1.

Heat extracted from storage tank:

$Q_{S,E}$ = Fluid Flow rate G (gpm) x 60 Min/Hr

- Fluid density ρ (lbs/gallon)
- Fluid specific heat Cp (Btu/lb°F)
- Temperature difference $(T_{10} - T_{12})$ °F Btu/Hr

MODE 3. COLLECTOR TO REHEAT COIL

Heat lost by fluid

$Q_{L,F}$ = Fluid Flow rate G (gpm) x 60 Min/Hr

- Fluid density ρ (lbs/gallon)
- Fluid specific heat Cp (Btu/lb°F)
- Temperature difference $(T_7 - T_8)$ °F

Heat gained by air

$Q_{G,A}$ = Air Flow rate Ft^3/Min x 60 Min/Hr

- Air density ρ_a (lbs/Ft^3)
- Specific heat of air $C_{p,a}$ Btu/lb°F
- Temperature rise $(T_5 - T_4)$ °F Btu/Hr

MODE 4. STORAGE TO REHEAT COIL

All equations same as Mode 2 except fresh air heating coil is replaced with reheat coil.

MODE 5. WATER HEATING FROM COLLECTOR

Solar collector performance same as Mode 1.

Water Heater Performance

Heat lost by fluid:

$Q_{L,F}$ = Fluid Flow rate G (gpm) x 60 Min/Hr

- Fluid density ρ (lbs/gallon)
- Fluid specific heat Cp (Btu/lb°F)
- Temperature difference $(T_{15} - T_{16})$ °F Btu/Hr

Heat gained by water:

$Q_{G,W}$ = Water flow rate G (gpm) x 60 Min/Hr

- Water density ρ (lbs/gallon)
- Water specific heat Cp (Btu/lb°F)
- Temperature difference $(T_{14} - T_{13})$ °F Btu/Hr

MODE 6. WATER HEATING FROM STORAGE

Same as Mode 2. Water heater performance defined in Mode 5.

MODE 7. STORAGE TANK CHARGING FROM COLLECTOR

Solar Radiation
Solar Collector Performance } Same as Mode 1.

Solar Tank Capacity:

$Q_{storage}$ = Total fluid volume V gallons

- Fluid density ρ (lbs/gallon) x
- Fluid specific heat Cp(Btu/lb°F) x
- Average temperature rise °F

$$\left(\frac{T_{10} + T_{11} + T_{12}}{3}\right)_{final} - \left(\frac{T_{10} + T_{11} + T_{12}}{3}\right)_{initial} \text{ °F Btu/Hr}$$

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