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DESIGN AND TEST REPORT FOR TRANSPORTABLE SOLAR LABORATORY PROGRAM

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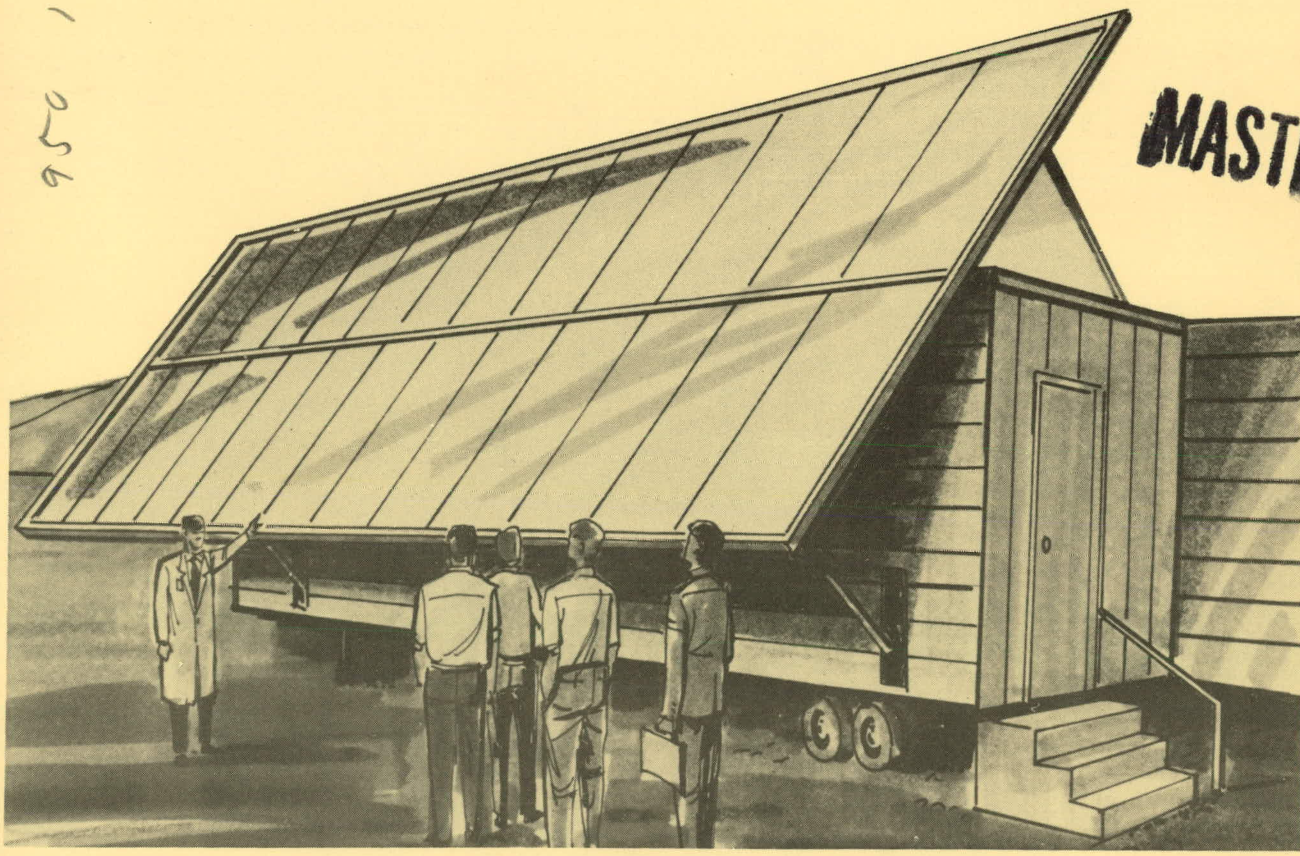

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OCTOBER 1974

DESIGN AND TEST REPORT FOR TRANSPORTABLE SOLAR LABORATORY PROGRAM

NSF Grant PTP74-01555

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Printed in U.S.A. 2852-41433

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SECTION I
INTRODUCTION

This report describes a portable solar heating and cooling laboratory, designed and built under the joint sponsorship of the National Science Foundation (NSF) and Honeywell Inc. The primary purpose of this laboratory is to provide a means to test the baseline design for a solar heating and cooling system under a variety of climates. By providing a calibration reference as well as a baseline, the laboratory also provides a means for evaluating and comparing both current and new solar heating and cooling designs.

A secondary purpose of the portable laboratory (while it collects experimental data at many population centers) is to demonstrate solar heating and cooling concepts to interested public officials, the building trades, and the general public for educational purposes.

Portability is accomplished by housing the complete solar laboratory in a 45-foot electronics van. A 10 x 50 foot mobile home accompanies the laboratory and provides additional heating and cooling space as well as a display area for educational materials.

The solar collector used is a state-of-the-art flat plate collector design with minor compromises in the design to make it portable. The total collector area is 700 square feet (including the support structure); 571 square feet of this is usable collector area. This area is comprised of 64 steel flat plate panels with selective black nickel coating, the panels being arranged in eight rows with eight panels in each row. Six inches of insulation are provided in the back of the panel and a double glazing is used on the front of the panel with an inside layer of four-mil Tedlar and an outside glazing of 3/16-inch, low-iron tempered glass. The entire collector array can be tilted from 30 to 60 degree elevation angles.

Two 475-gallon storage tanks provide heat storage. This collector/storage system represents a typical residential solar heating system. The collector area comprises approximately 60 percent of the heating/cooling space and the storage size is approximately one gallon per square foot of heating/cooling space.

The control system has been designed to evaluate solar heating/cooling concepts. This control system is conventional, but versatile and flexible. It is designed to provide maximum efficiency by taking advantage of a stratified storage concept; it attempts to use all available solar heat. This control concept is designed to operate under normal conditions and will control automatically, in any one of 20 different heating control loops or 24 cooling control loops. A two-stage heating and cooling thermostat is the primary input to the digital logic. At any time a manual override system can interrupt the automatic sequence so that special testing can be performed in any one of the control modes.

A complete data taking system is provided and provisions are made for rapid data processing. All data points can be read automatically using a data multiplexer which will print out the information on magnetic tape. At any time, the data on the magnetic tape can be transmitted back to Minneapolis for analysis using the Honeywell central site computer. Information is gathered on all important temperatures and flow rates. Complete weather data are gathered including insolation, temperature, humidity, barometric pressure, and wind speed. The control loop is monitored automatically to determine the percentage of times during which the system will operate in any given mode. Under normal circumstances these data will be gathered over a period of a full day, collected on the magnetic tape, and reduced at the end of each day.

Two 3-ton solar power air conditioners are being investigated: an ARKLA absorption-type air conditioner and a Barber-Nichols Rankine cycle air conditioner. The ARKLA air conditioner is a conventional gas-fired absorption system converted to operate from a hot water input. The Rankine cycle air conditioner uses a conventional vapor refrigeration cycle driven from a Rankine cycle engine and using the hot water to boil a freon working fluid which in turn drives a turbine wheel to produce shaft horsepower. The Rankine air conditioner provides a great deal of flexibility in that a motor generator is connected to the output shaft and can generate electrical energy as well as air conditioning. The motor can augment the solar system to provide additional cooling in the absence of solar inputs. Both air conditioners are normally augmented by a propane-fired boiler under low solar input situations.

A test tour is being conducted to accumulate data as well as to demonstrate lab operation under a variety of U.S. climates. Included in the tour will be the cold/moist northern states, and hot/dry southwest states, the hot/moist southeast states and the moderately cold/moist northeast states. The practicality of the solar heating and cooling system in these areas will be demonstrated to groups interested in designing or using solar energy for residential or commercial applications.

SECTION II
SUMMARY

The transportable solar laboratory was designed and built primarily as a means to evaluate solar heating and cooling systems under various climatic conditions. The laboratory also demonstrates the use of these systems to the solar energy community and acquaints public officials, business men, and other interested groups, including the general public, with solar energy as an alternate or augmentative method for heating and cooling buildings.

This laboratory is capable of supplying (with only solar inputs) 75-90 percent of the heating requirements of the 1000 ft^2 lab area in virtually any part of the continential U.S. It is equipped with two air conditioners; an absorption cycle and a Rankine cycle powered vapor compression cycle; it can supply 3 tons of solar-powered air conditioning. A 700-square-foot collector supplies the solar energy. Only 571 square feet are usable due to the high shadowing losses of the window mullions and the frame -- this is approximately 60 percent of the heated area. The steel collector is a conventional flat plate design having a selective black nickel plating. A double glazing covers the absorbers and 6 inches of fiberglass insulation covers the back surface.

A 70,000 Btu propane-fired boiler augments the solar collection system.

Under clear sky conditions, a surplus of solar energy would be collected. This energy is stored as heated water in two 475-gallon tanks and represents an energy storage capability of one million Btus. This storage was sized for 1 gallon per square foot of floor space.

Heating and cooling is distributed with a forced air system which transfers heat from solar-heated hot water by means of fin and tube heat exchangers. Air conditioner evaporators are located in the same air plenum. Hot tap water is provided with an in-line heat exchanger at the pump outlet.

The collection and heating/cooling distribution systems are fully automatic operating from a commercially available two-stage thermostat. Specific experiments can be conducted using an alternate manual control system.

Instrumentation and data processing equipment can also be operated either automatically or manually. A 120-channel Kaye Electronics data multiplexer collects the information and is used to store information on magnetic and paper tape. A Honeywell 6080 time share computer processes the plots and the magnetic tape daily.

An 8 x 45 foot semitrailer houses all the solar collection and engineering equipment. A companion 10 x 50 foot mobile home provides additional heating and cooling load as well as office space and a place to display solar energy educational material.

SOLAR COLLECTION SYSTEM

The solar collector assembly is comprised of 64 individual absorber panels. These panels are arranged in four quadrants of 16 panels each. In each quadrant there are two rows of eight panels for a total of eight rows. Figure 2-1 shows this arrangement; all 64 panels are connected in parallel.

The selective black nickel plating is designed for 90 percent absorption minimum and 10 percent emittance maximum. The performance for the collector with this plating and with the double glaze and 6-inch insulation will approach 75 percent efficiency as the plate temperature approaches ambient. At 210°F, efficiencies of 50 percent can be expected.

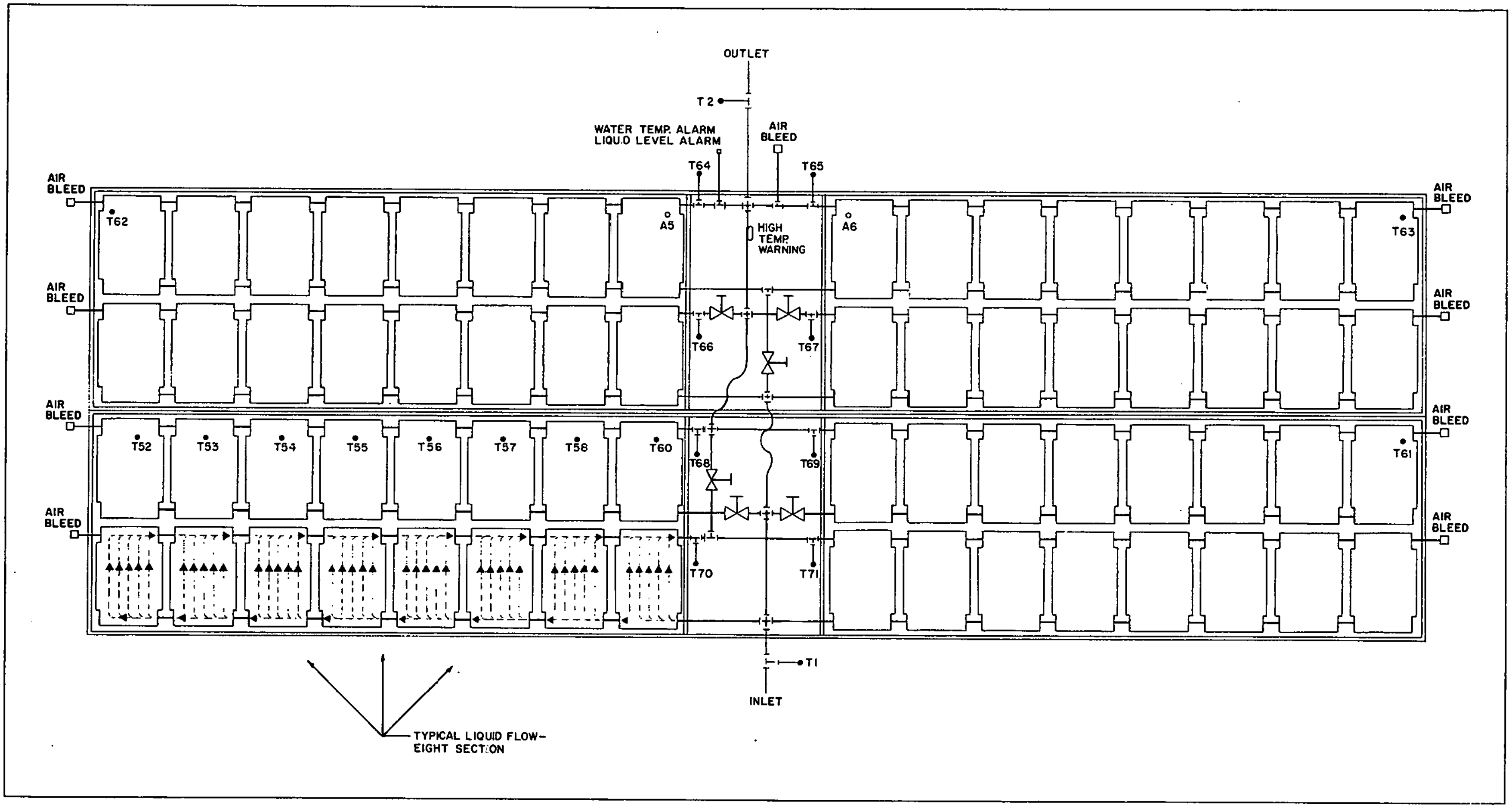


Figure 2-1. Collector Panel Layout

A double glaze was selected as a best compromise recognizing that a single or triple glaze might be an optimum for certain latitude and seasonal conditions. The outer glaze is 3/16-inch-thick tempered glass to withstand the rigors of transportation. An inner layer of Tedlar was selected for its low weight and cost with negligible performance compromises.

The collector array is hinged and supported at the upper left corner of the truck. Hydraulic actuators position the array to the proper angle and have an angular adjustment capability of 30-60 degree tilt which will satisfy most U.S. latitudes and seasons.

The collector fluid is common to the entire system and is a 50 percent by weight mixture of ethylene glycol and water. This fluid provides a good compromise of high specific heat, corrosion inhibitors, and freezeup protection. The maximum collector flow rate is 8 $lb/hr\text{-}ft^2$ which is suitable for air conditioning but marginal for heating.

HEATING AND COOLING SYSTEM

The heating and cooling system is compromised of (1) the plumbing system to transfer fluid to and from the collectors (Figure 2-2), (2) heat exchangers in a forced-air blower plenum to transfer heat to the air, (3) storage tanks to store excess energy, (4) expansion tanks to provide expansion volume at high temperatures without loss of system pressure, (5) pumps, (6) an auxiliary boiler, (7) two separate air conditioning systems, (8) a domestic, propane-fired hot water heater for supplemental energy when insufficient solar energy is available, and (9) the necessary valves to direct the fluid flow as required during the various modes of operation.

The forced-air blower system distributes the air from the heat exchangers to the equipment van and the display mobile home. The two trailers are interconnected with flexible ducts for supply and return air. The blower plenum contains two fin and tube heat exchangers for heating and two evaporator coils for air conditioning. The design air flow rate is 1200 CFM for both heating and air conditioning. A centrifugal pump is used to circulate the fluid through the system. This pump is located between the outlet of the storage tanks and the inlet to the solar collectors. A second pump is located between the solar collectors and the inlet to the air conditioners. This second pump is needed during the cooling season to offset the additional pressure drop in the air conditioners.

The two air conditioning systems will be operated individually to evaluate their operation and efficiency under a range of climatic conditions. The absorption A/C is a modified gas-fired unit designed and built by ARKLA Corporation. It uses solar heated water to replace the gas combustion unit. This absorption unit operates at water temperatures from 196°F to 210°F with condenser water of 85°F.

The second system uses a Rankine cycle engine (heated by solar heated water) to drive a conventional vapor cycle air conditioner. This system also has a motor-generator which can be used to drive the compressor electrically or to generate electric power.

Both air conditioners are rated at 3 tons at 80°F dry bulb and 67°F wet bulb. Inlet water required is 11 gpm with inlet temperature of 210°F for the absorption unit and 215°F for the Rankine unit. The COP is rated at 0.7 for both units. Predicted minimum operating inlet temperature is 196°F for the absorption unit and 177°F for the Rankine unit. Rated air flow of both units is 1200 cfm.

Heat rejection from both air conditioners is supplied by an 8-ton Manley evaporative cooling tower.

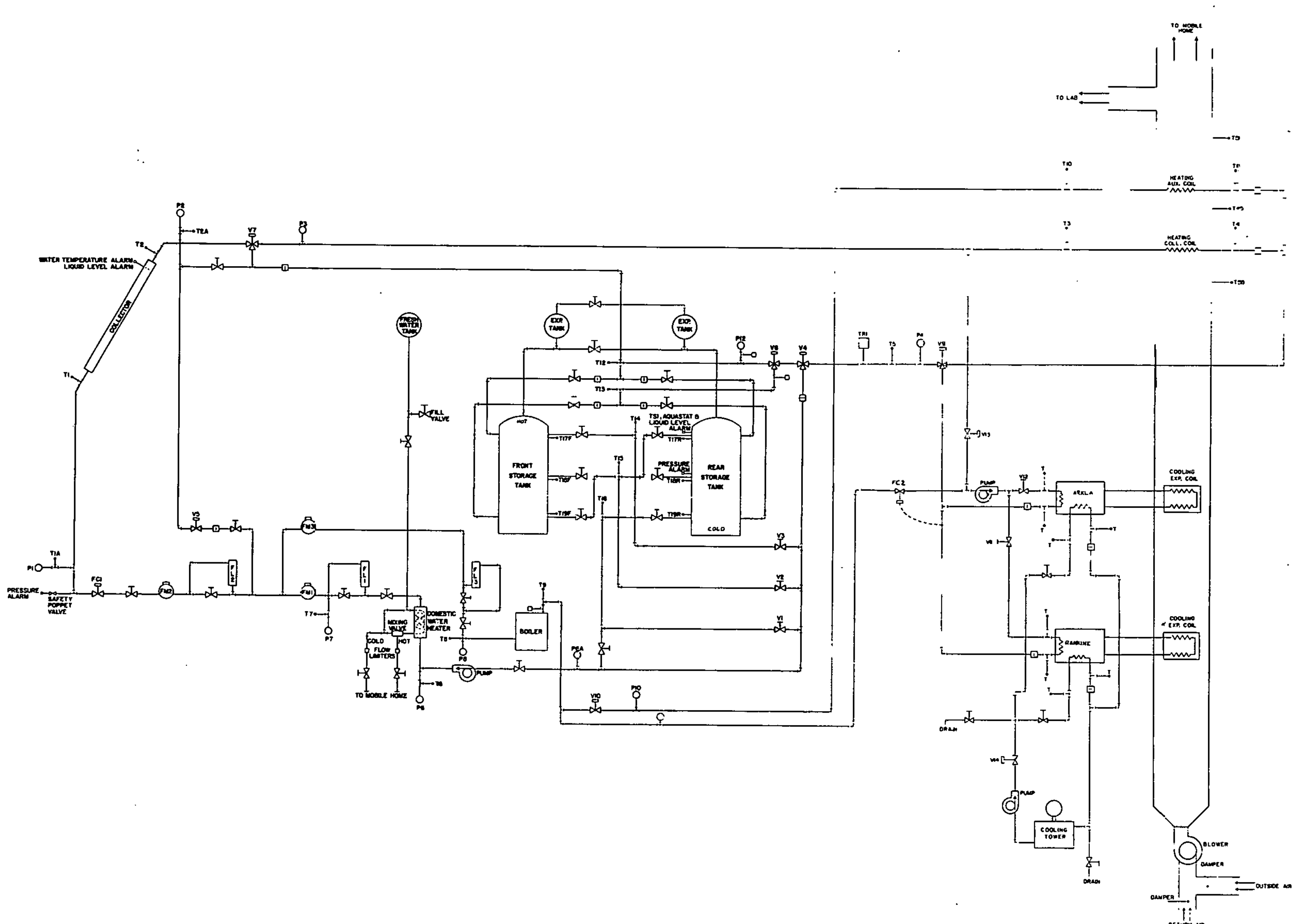


Figure 2-2. Solar Heating/Cooling System Diagram

CONTROLS

A control console located in the equipment van is capable of operating the heating/cooling system either manually or automatically. In the manual sequence the operator is able to independently control the flow pattern and to operate in any heating or cooling mode. At present there are approximately 20 heating modes and 24 cooling modes.

In the automatic sequence the system will maintain a set room temperature and will collect solar energy at a preset outlet temperature. As a surplus of energy becomes available it will be diverted to the storage tanks. Inputs to the logic control are a two-stage thermostat, percent collector flow, storage temperature, and return water temperature.

All the valves involved in the control system are commercially available and are electronically operated. Some of the valves are motor-driven with limits to lock the valves in the open or closed position. No current is drawn when at either position. Other valves are motor operated with spring-return actuators. These require power when in the open or on position.

Control modes can be divided into heating and cooling. The primary heating modes are as follows:

- Direct solar heating supplied directly from collectors
- Surplus solar energy returned directly to storage tanks with no heating requirement
- Solar heating using stored energy -- no collection available, i.e., no sunlight
- Heating using auxiliary boiler system -- no collection and no stored energy available
- A combination of auxiliary energy supplied with either collector energy or stored heat

The primary cooling modes are as follows:

- Direct solar energy powering either air conditioner
- Surplus energy returned directly to storage tanks with no cooling requirement
- Air conditioning using stored energy -- no collection available, i.e., no sunlight
- Air conditioning using the auxiliary boiler to provide necessary hot water to operate air conditioners -- no collection available, i.e., no sunlight
- Air conditioning using an electric motor to drive Barber-Nichols Rankine cycle air conditioner
- A combination of auxiliary power with either direct solar or storage water

Several of the mode combinations during both heating and cooling use three levels in the storage tanks. These modes are primarily in the system to increase efficiency by taking advantage of the storage tank stratification. To implement usage of different levels, water is returned to the top or the bottom of the storage system. Stratification is achieved by combining tank baffles and the use of two tanks connected in series.

The automatic control system uses TTL logic and a combination of triac and relay valve interfaces.

INSTRUMENTATION AND DATA ACQUISITION

The system is fully instrumented to evaluate performance of all heating and cooling components. Instruments for both manual and automatic monitoring are included. Temperature, water flow, air flow, solar radiation, relative humidity, barometric pressure, and wind speed/direction are parameters measured. High system pressure and temperature safety alarms are also part of the instrumentation system.

Copper-constantan thermocouples are used throughout the entire system. Temperature measurements are made on 11 solar panel surfaces, key locations in the system fluid, air in the distribution ducts, specified critical locations on the air conditioners, ambient air, and outside walls of both the equipment and display vans. There are approximately 100 thermocouples at various locations throughout the system.

Water flow (fluid) is measured in three locations using two flowmeters in each location to permit manual and automatic readout. Total system flow is measured after the main system pump. The collector flow and the auxiliary boiler flow are also measured. This combination makes it possible to calculate the flow in any mode. Brooks Rotometers are used for visual flow measurement at each location,and Brooks nutating disk flowmeters in combination with pulse generators and D/A converters, produce a voltage output proportional to the fluid flow. Air flow measurements are made in the blow cabinet after the heating coils and also in the fresh air intake duct. A thermo systems hot wire anemometer sensor is used to calibrate airspeed.

Two Kipp Zonen pyranometers are used to measure the solar radiation intensity. A voltage is produced which is proportional to the solar radiation in Langleys. One sensor is located on the solar collection panel to determine the incident radiation on the panels. The other is located at the top of the collector to determine the solar radiation in a horizontal plane. An analysis of tilted and horizontal radiation will yield the direct and diffuse radiation components.

A complete weather station measures outdoor ambient air temperature, relative humidity, barometric pressure, and wind speed/direction. All of these devices have electrical and visual readouts.

In the interest of safety and to warn the operator of unusual conditions, red warning lamps are lit when abnormal conditions occur. There is a high-temperature warning at the top of the solar collector to warn of boiling temperatures. Water level sensors are located at the top of the solar collector and near the top of the storage tanks to warn of low water levels. Pressure sensors are located in the storage tanks and at the inlet to the solar collectors. The solar collector inlet pressure sensor is to warn of high pressure while the storage tank sensor warns of too low a pressure in the storage tanks.

All sensor signals used to analyze system performance are fed to a data multiplexer where the signals are periodically recorded on paper printout or on magnetic tape. The magnetic tape can later be transmitted back to the 6080 time share computer at Honeywell via a telephone link. A video display, keyboard, and telephone link to the computer allows the operator to process data on site as desired. The computer is used to compile the data and calculate functions for data plotting.

A photo of the complete control and instrumentation system is shown in Figure 2-3.

VEHICLES

The transportable solar laboratory (Figure 2-4) is a two-trailer unit. The equipment van is an 8 by 45 foot semitrailer which contains the solar collector panels, fluid and air distribution system for both solar heating and air conditioning, plus the instrumentation and data acquisition systems. The display van is a 12 by 50 foot mobile home. It serves as additional heating and cooling load and houses the engineering office and a conference room outfitted with educational solar energy material.

Figure 2-3. Control Console, Instrumentation, and Data Acquisition

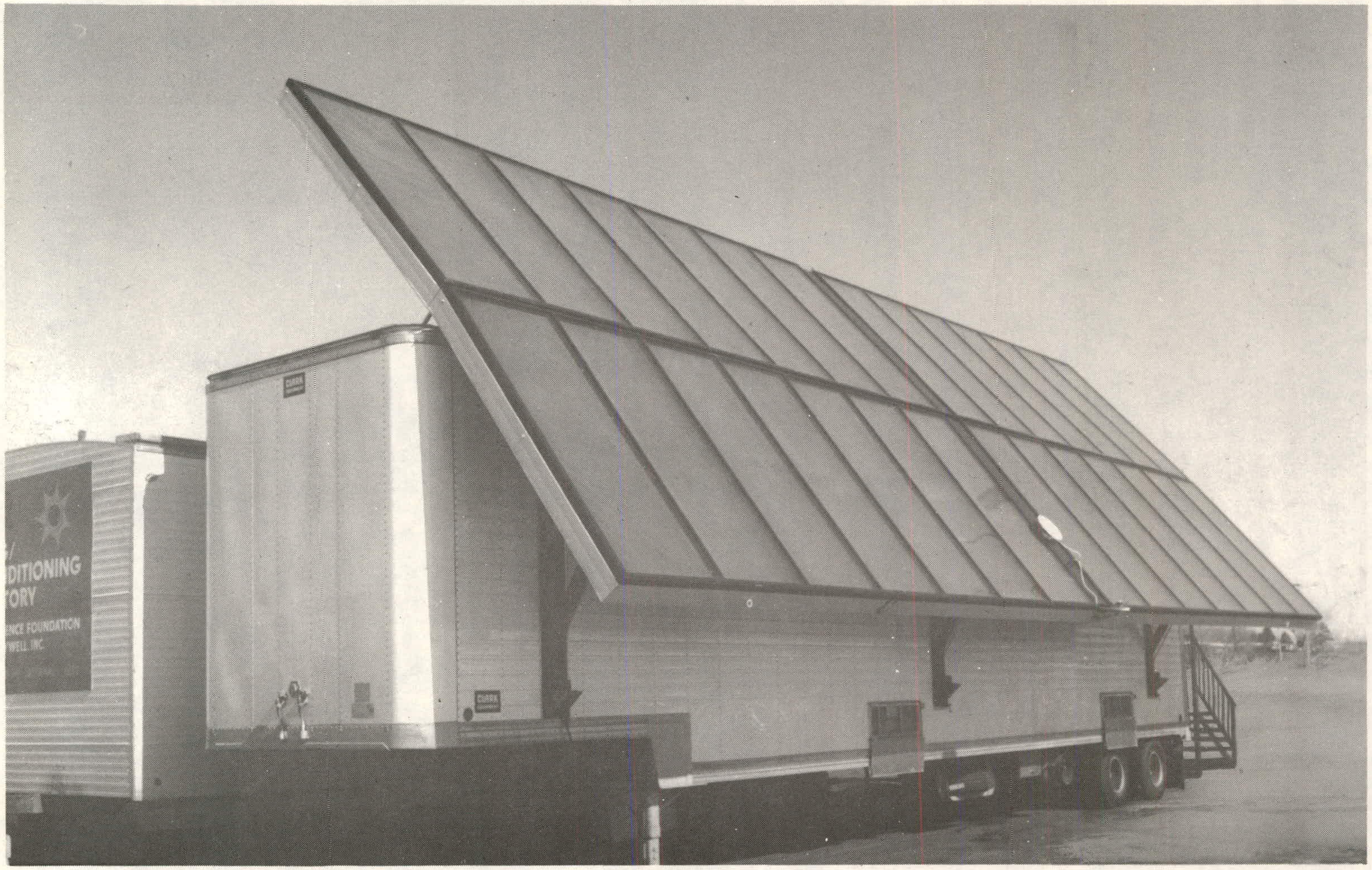


Figure 2-4. Solar Transportable Laboratory

The two trailers are interconnected by a walkway. Air supply and air return ducts are also used to interconnect the two trailers for air distribution. The control thermostat for automatic temperature control is located in the display van.

LABORATORY MATERIAL COSTS

Major cost items for the solar laboratory are listed below. These costs do not include the design and development costs associated with the program but do reflect fairly accurate replacement costs of the equipment.

Mobile home - unfinished	$ 8,250	
Mobile home - finishing and display	42,000	
Semitrailer - unmodified	8,500	
Solar collectors	12,000	
Storage tanks - two 450-gallon	1,500	
Absorption air conditioner	2,000	
Rankine cycle air conditioner	82,000	(includes development)
Consulting services	9,000	
Data acquisition system	22,000	
Flowmeters, valves, pump, plumbing	5,600	
Solar collector hydraulics	800	
Semitrailer modifications	6,200	
Weather instrumentation	900	
Pyranometers	1,800	

SECTION III
SOLAR LABORATORY DESIGN

ABSORBER PANEL DESIGN

The solar collector assembly is comprised of 64 flat plate steel collector panels 47 inches high by 32 inches wide. The panels were fabricated by Honeywell and electroplated with the selective black nickel coating by a vendor.

The collector panel is fabricated from two sheets of 20-gauge cold roll steel. One sheet is embossed with the 14 flow passages which are 0.24 inch wide and 0.050 inch deep formed on two-inch centers running the length of the sheet. The other sheet is embossed with the inlet and outlet manifolds which are two inches wide and 0.5 inch deep formed along the 32-inch dimension and intersecting with the flow passages. The two sheets are spot welded at 0.75-inch intervals on either side of the flow passages. The assembly is seam welded around the entire periphery to form a water-tight seal. Steel connectors that transition from a 1.625-inch diameter round tube to a 2-inch by 0.5-inch rectangular cross section are welded into the open ends of the manifolds to provide a plumbing connection. These panels are pressure tested at 40 psi. These panels weigh approximately 30 pounds.

Solar Collector Mounting

Design of a suitable mounting structure for the collector assembly to satisfy the transportability requirement presented unique problems: (1) It has to support the collector weight under the stress of over-the-road travel; (2) It had to be adjustable to accommodate all latitudes and sensors; (3) It could not detract from the collector performance; and (4) It had to be transportable in terms of size and shape. To satisfy these design conditions, a two-unit collector assembly was used which could be hinged about the upper corner of the truck. Each half was subdivided to form four equal quadrants.

Each solar collector frame is 44 feet long and 8 feet wide, and each of these halves is subdivided into two sections with a 12-inch space between them. Figure 3-1 shows a plan view and cutaway of the solar collector panel. The main support frame is a 6-inch by 2-inch steel channel with stiffening stringers which are 2 inch deep channels and tubing. The tubing also serves as a reinforcement for hydraulic cylinder supports. Gussets are also welded to those tubes at the point where the hydraulic cylinder support is fastened down.

The rear surface away from the sun is enclosed by sandwich construction of 0.5-inch plywood, 3.5-inch fiberglass insulation, 0.5-inch plywood, and another layer of 0.5-inch plywood, which is cemented and riveted to the steel frame. This sandwich provides good insulation and structural rigidity.

After fabrication, the panels were plated with one-half mil bright nickel followed by a black nickel selective coating. The finished plating was measured to have a 90-percent absorptance and a 10-percent emittance in the solar spectrum.

The absorber panels were attached to the inside plywood decking using aluminum standoffs. This ensured a suitable thermal barrier to the outside and minimized heat loss. Along the outside periphery of the channel urethane foam insulation strips minimized heat loss to the steel frame.

This collector assembly was glazed with a double layer of glass and Tedlar. The 3/16-inch thick tempered glass outer layer provides good transmissivity, maximum structural strength, and protection against weather and vandalism. The glass also shields the inner Tedlar layer from ultraviolet radiation. A low iron glass composition yielded a transmissivity of 92 percent versus 91 percent for the 4-mil Tedlar.

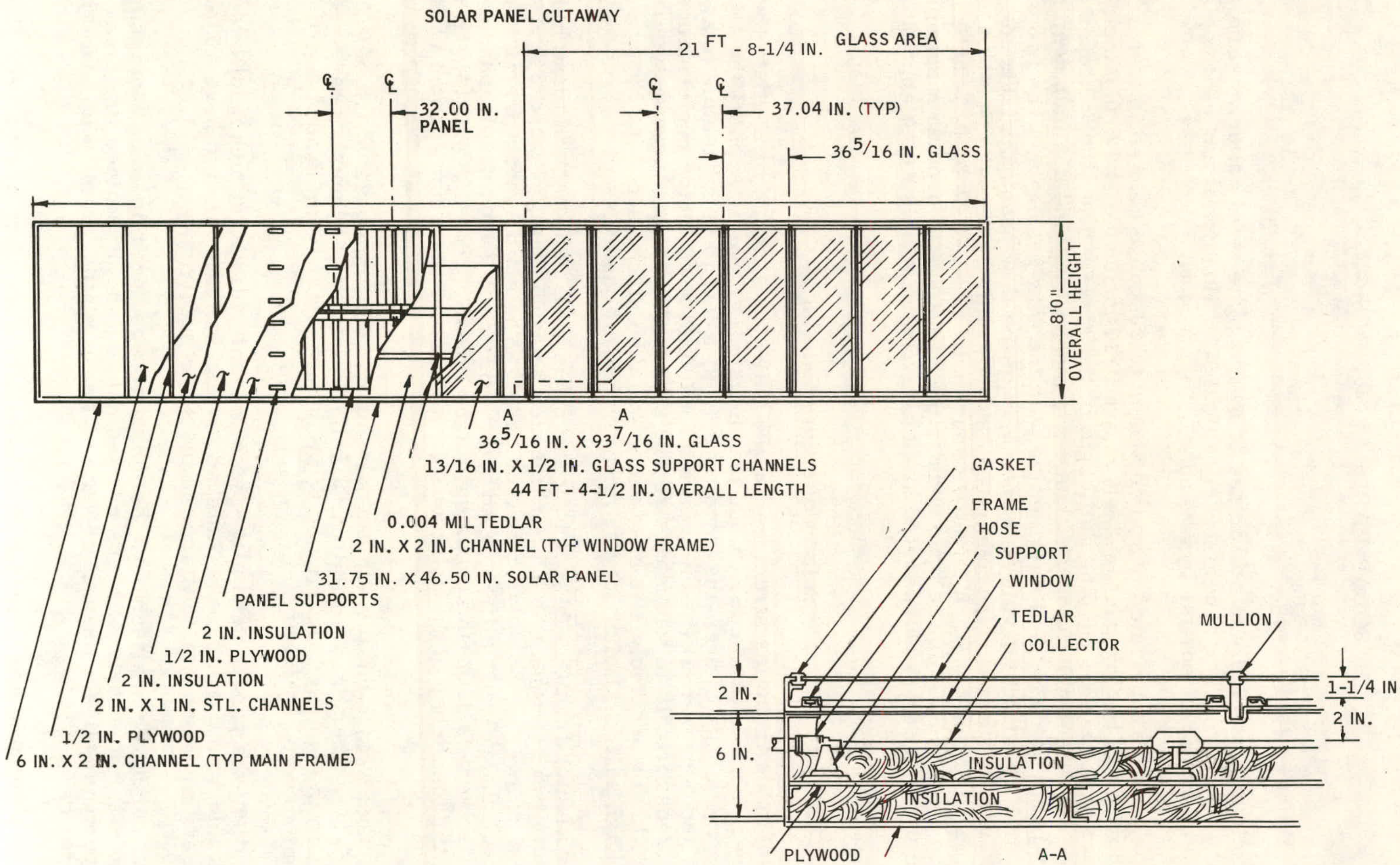


Figure 3-1. Solar Collector Frame Cutaway

Figure 3-2. Close-up Cutaway of Collector

The Tedlar is a DuPont polyvinyl fluoride film with good strength and approximately 91 percent solar radiation transmissivity. The Tedlar film used in the laboratory is 0.004-inch thick designated as 400 XRS-145 TR, Type 30. The Tedlar film was preshrunk by exposing each sheet to 300°F for five minutes before mounting it in a screen frame. Then the Tedlar was assembled in aluminum screen frames and held at the edges with a vinyl spline pressed into a groove in the frame. The spline is held in the groove by an aluminum angle fastened to the frame to press against the spline. The angle was added to prevent the Tedlar from pulling out of the groove as additional (about 80 percent was removed by preshrinkage) shrinkage occurs due to repeated exposure to the heat from the collector.

The glass is mounted in rubber-zipper-type frames approximately 1.25 inches away from the Tedlar. Each pane is held to the frame by a rubber zipper strip manufactured by Standard Products Corporation, Cleveland, Ohio under the trade name Stanlock.

Mounting structure for the windows was fabricated and installed by DeVac Inc., located in St. Louis Park, Minnesota.

Hydraulic Positioning System

The hydraulic system was designed taking into consideration the following design parameters:

- All three of the hydraulic cylinders must actuate at the same speed to prevent the possibility of warping the panel.
- The speed of actuation should be sufficiently slow to allow the operator to raise the panels so that they are close together and yet respond fast enough to prevent their smashing into each other.
- The system must be able to move the panels without exceeding 2000 psi hydraulic pressure.
- The operator must be able to control each cylinder individually.
- Care must be taken to prevent accidental lowering or raising of a panel.
- Since the operator may wish to work under the panel while it is extended, steps must be taken to prevent the panel coming down while someone is under it.
- Both panels must be able to swing approximately 60 degrees from the rest position.
- Since the system is to be capable of operating from an auxiliary power supply, the horsepower required should be kept as small as possible.

The two solar collector panels must be tilted away from the van so that the sun's rays normally hit them at noon. Both collector panels are hinged at the left edge of the roof (refer to Figure 3-3). The top panel folds down flat, and during transit it is supported by the three hinges and by six pad supports on the non-hinge side. Two over-center type locking clamps hold the top panel down during transit. The side panel folds in to the side of the van against three stops. During transit the side panel is securely locked to the side by three over-center type locking clamps. In addition, five bolts, which are threaded into three side supports, are screwed up against the steel frame of the side panel. These help to support the panel and relieve the hinge loads. These bolts are secured by five jam nuts.

Each panel weighs approximately 4000 pounds. During transit they are covered by tarps that weigh an additional 100 pounds each.

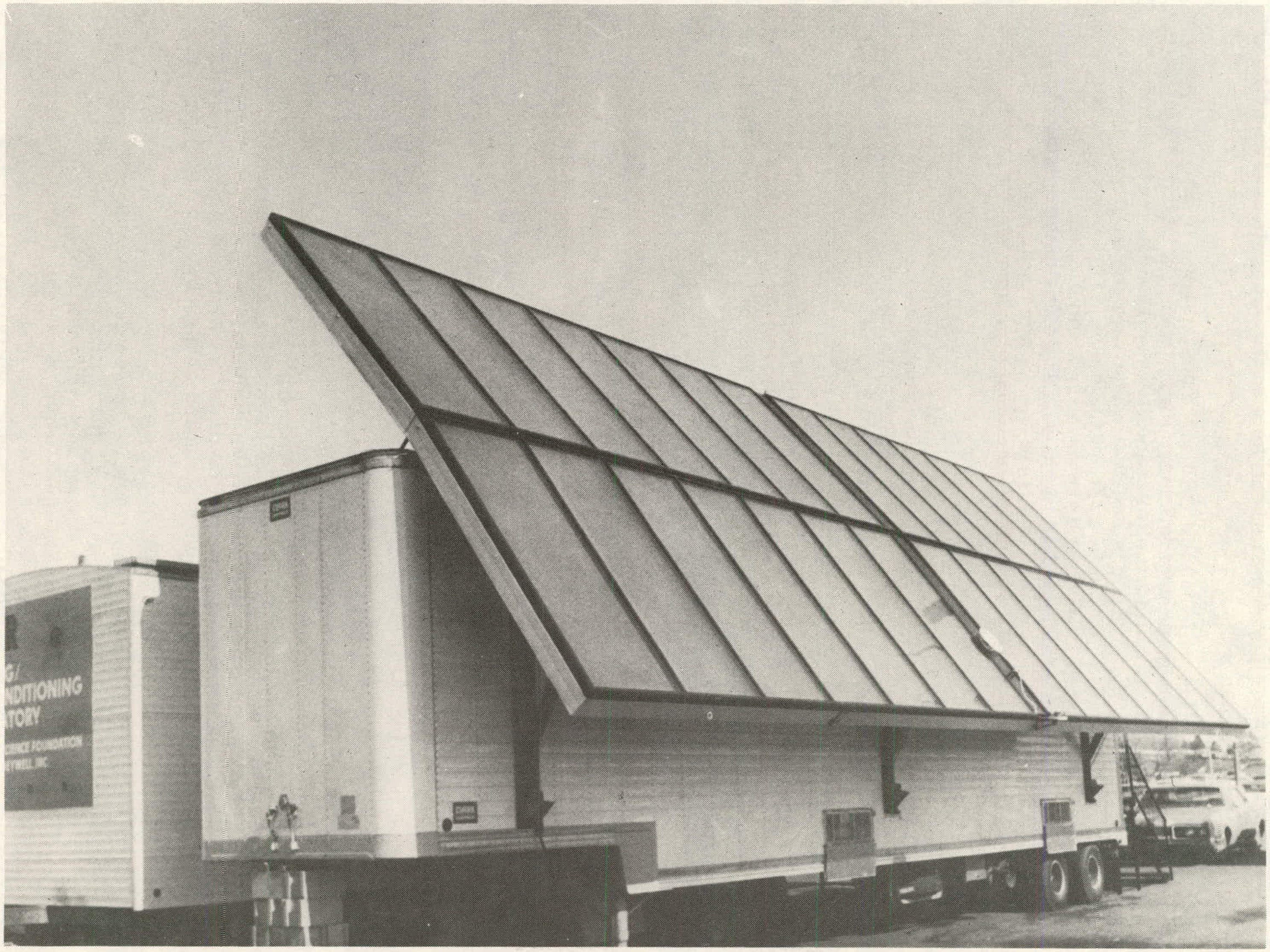


Figure 3-3. Solar Transportable Laboratory

Figure 3-4 shows the loads on the hydraulic cylinders for the top panel when it is being raised. The weight of 4100 pounds is equally distributed over the entire panel. The two end cylinders each have to lift 29 percent of the load, and the center cylinder lifts the remaining 42 percent. The loads are greatest when the panel is down because the rams are only 18 degrees from horizontal and the side force is greater than the lift force. In addition, a snow load could also possibly be on top of the panel. The loads on the front and rear cylinders are 4118 pounds each, and the load on the center hydraulic cylinder is 5964 pounds.

The maximum side solar collector hydraulic cylinder loads also occur when the panel is next to the truck. There are two reasons for this. First, when the panel is against the truck, the angle of the hydraulic cylinder to the panel is only 11.5 degrees; second, if there is a wind blowing from the side, it is greatest when the panel is next to the truck, and it is reduced as the panel swings out away from the truck. This occurs because the solar panel is basically a large flat plate and the drag coefficient on an inclined flat plate varies with the angle of attack of the wind on the plate. It was measured experimentally by A. Faye and F. Johansen and is shown in Figure 3-5. For this analysis a maximum wind velocity of 60 mph was selected, which gives a wind pressure force normal to the panel of 3500 pounds.

The required hydraulic force can then be calculated in a manner similar to that done for the top panel as is shown in Figure 3-6. The summation of the moments about the hinge point must equal zero. Therefore, when it is starting to raise, the load on the front and rear cylinders are 6290 pounds each and the load on the center hydraulic cylinder is 9110 pounds.

For uniformity of required stroke volume, the top cylinders are identical as are the side cylinders. Therefore, the maximum load must be used, and the largest rod available was chosen for stiffness with the long extension required. Table 3-1 shows the resultant cylinder specifications.

Table 3-1. Hydraulic Cylinder Specifications

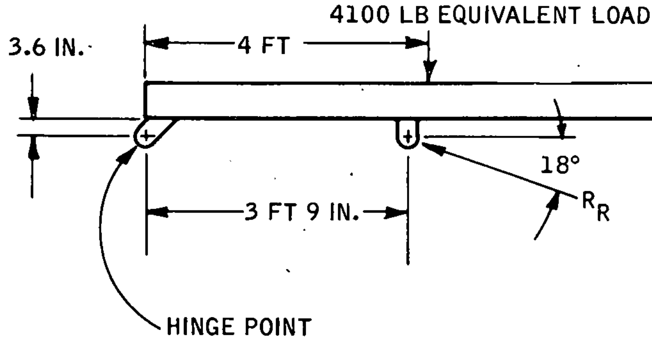
	Design Load (lb)	Rod Diameter (in.)	Cylinder Stroke (in.)	Cylinder Bore (in.)	Cylinder Pressure (psi)
Top	5964	1-3/8	42	2.0	1860
Side	9110	1-3/4	48	2.5	1830

Figure 3-7 is a schematic of the hydraulic system. The three identical pumps are all driven by timing belts at identical speeds from the 5 hp motor.

The pumps used deliver 1 gpm at 3450 rpm at 0 psi. This drops to 0.74 gpm at 2000 psi. These pumps require up to 1.25 hp each, depending on speed, pressure, and flow rate.

Figure 3-8 illustrates the control panel for the hydraulic actuation. The control panel is mounted on the end of a 50-foot cord so that the operator can walk around the truck and inspect the operation as he actuates the panel. The main power switch is detented and requires that the handle be pulled out to turn it on or off. The two push buttons on the right start and stop the motor. The operator picks top or side by pushing the toggle switch.

This switches all three of the two-position, four-way solenoid valves to the proper position. It also provides power to the control relays of the three 3-position, 4-way solenoid activated valves. The four other toggle switches are center off spring return switches. Switch 4 controls the relays for all three of the 3-position, 4-way solenoid valves. Pushing it to the up position drives all three hydraulic cylinders. The other three switches control each cylinder individually. The solenoid valves all use 110 volt, 60 Hz power. Relays are used so that only 24 volts are in the control box. This is to minimize the danger to the operator who may have to actuate the panels in wet conditions.



$M_{HINGE} = 0$

$4100 \times 4 = 3.75\ R_R \sin 18° = 0$

R_R = 14150 LBS TOTAL LOAD - ALL THREE RAMS

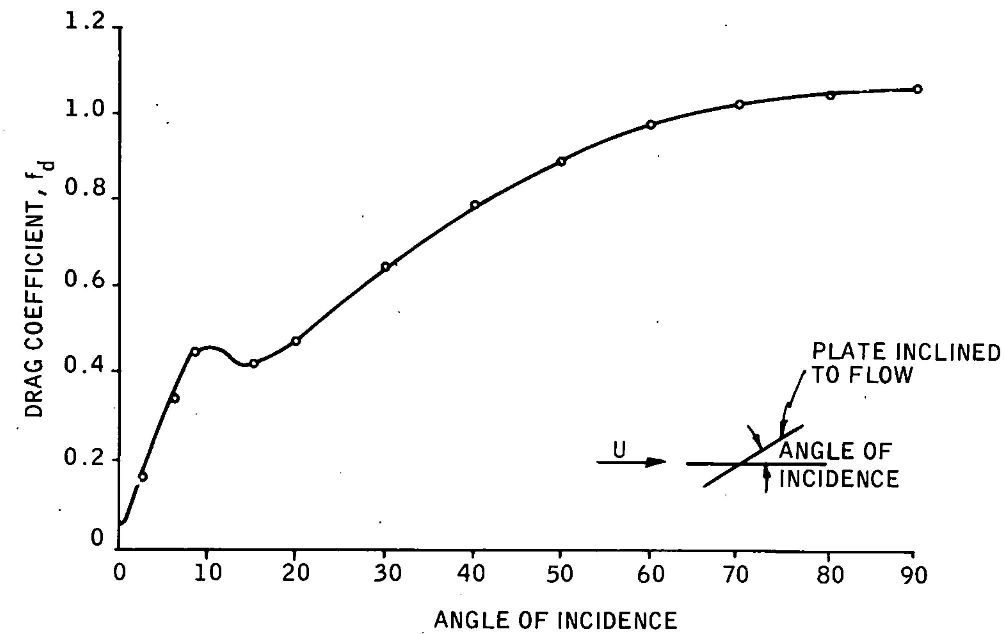
FOR FRONT AND REAR HYDRAULIC CYLINDERS

$R_{RF} = R_{RR} = 0.29(14200) = 4118$ LBS

FOR THE CENTER HYDRAULIC CYLINDER

$R_{RC} = 0.42(14200) = 5964$ LBS

Figure 3-4. Design Load to Lift Top Panel



THE WIND LOAD IS EQUAL TO:

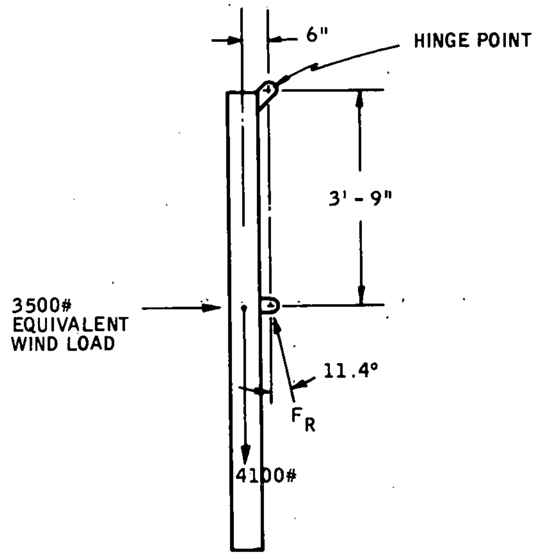
$$F_D = 1/2 \rho V^2 f_D A$$

WHERE F_D = WIND PRESSURE FORCE ON THE PANEL

ρ = DENSITY OF THE AIR = 0.002377 SLUGS/FT.^{3} AT STANDARD CONDITIONS AT SEA LEVEL

V = WIND VELOCITY IN FEET/SEC.

Figure 3-5. Drag Coefficients for Flow Past Inclined Flat Plates



$M_{HINGE} = 0$

$3.75\, F_R \sin 11.4° = (3500 \times 4) + (4100 \times 0.5)$

$F_R = 21700\#$ TOTAL LOAD - ALL THREE RAMS

FOR FRONT AND REAR HYDRAULIC CYLINDERS

$F_{RF} = F_{RR} = 0.29(21700) = 6290\#$

FOR THE CENTER HYDRAULIC CYLINDER

$F_{RC} = 0.42(21700) = 9110\#$

Figure 3-6. Design Load to Lift Side Panel

Since it is impractical to make a spool valve that never leaks, pilot-operated check valves with zero leakage are installed in the hydraulic line to the head end of each cylinder. Thus, the panel can only sag due to leakage across the piston in the cylinder. Adjustable panel supports are also provided to prevent panel sag and to minimize high pressures in the cylinders and lines due to wind gusts.

SOLAR ENERGY DISTRIBUTION SYSTEM

Collector Fluid Flow Paths

The fluid distribution system was designed with a redundant automatic or manual control capability. The manual system provides the flexibility to perform specific experiments as well as to provide a backup to the automatic control system.

The automatic system will operate in 20 heating modes and 24 cooling modes using only a two-stage heating/cooling thermostat and a differential return temperature, collector flow rate, and storage temperature as inputs.

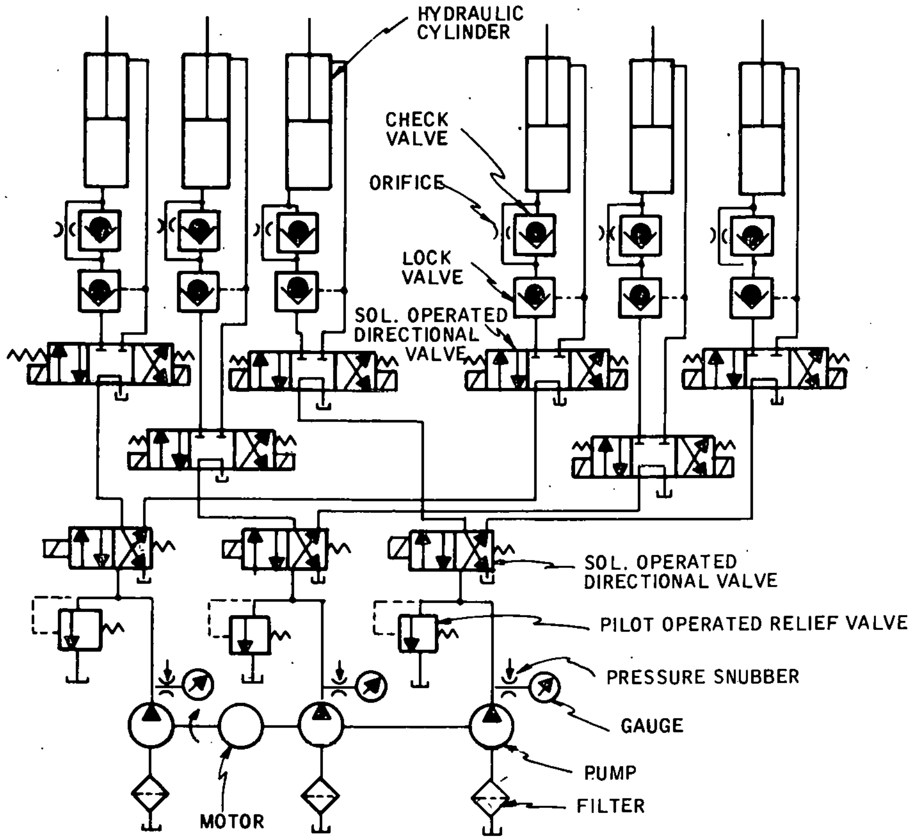


Figure 3-7. Hydraulic System Schematic

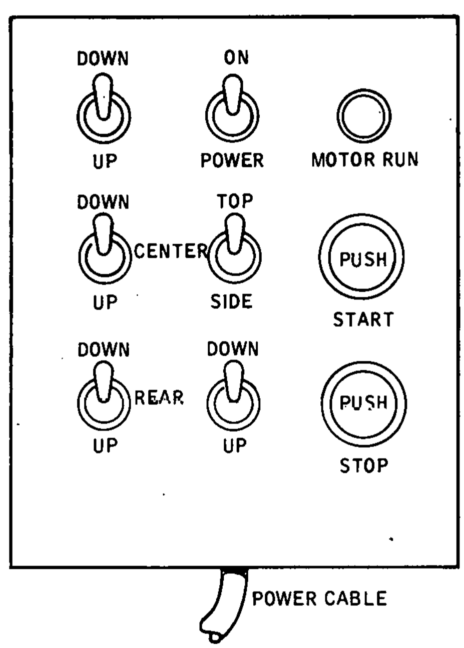


Figure 3-8. Hydraulic Control Panel

Most of these modes are shifts in the entry and exit of the fluid from the storage tanks. If the entering fluid is hotter than that in the top of the tank, it enters the top, otherwise it enters at the bottom to keep the tank stratified with the hottest fluid at the top. To help maintain the collector output at a constant temperature, the input flow rate is modulated and the rate determines whether the fluid comes from the top, center, or bottom of the tank. When no solar energy is available and storage heating is required, the hottest water from the top is used. These flow paths are designed to conserve energy by making the best use of that which is available. Figure 3-9 shows the fluid flow diagram for a system of this type. Figures 3-10 and 3-11 show the heating and cooling logic flow diagrams which control the system under automatic conditions.

Other than these changes in the storage handling, there are nine basic operational heating and cooling modes controlled by the two-stage thermostat and the control system logic. These are illustrated in Figures 3-12 through 3-20 and are as follows:

- Collect solar energy heat and send to storage.
- Heat space with solar energy collector.
- Heat space only with pre-stored heat.
- Second stage heating using solar energy collector augmented by boiler.
- Second stage heating using pre-stored heat augmented by boiler.
- Second stage heating with boiler only, bypassing storage.
- Cooling with solar energy collector.
- Cooling with solar energy augmented by boiler.
- Cooling with stored energy and/or boiler.

In the figures, the ones which have submodes of changing the flow path to and from the storage have these alternates shown dotted.

Heated Air Flow Path and Controls

The air from the blower enters a vertical 18-inch by 24-inch plenum chamber which contains two fin and tube exchangers. At the top of the plenum the air is divided with one duct leading to the center line of the equipment van and the other to a 12-inch diameter opening in the roof. Outside the roof an insulated flexible duct transfers the air to a similar opening in the top of the display unit. Above the drop ceiling in that unit the duct leads to the centerline and then branches toward the two ends. Along these ducts are located five adjustable air outlet registers. The wall thermostat in the display van has two stages for heating and two stages for cooling.

In the heating season making the first contact turns on the blower and diverts solar heated water to the collector heat exchanger. If the space temperature continues to go down, the second stage contact is made which turns on the boiler and opens the line leading to the auxiliary heat exchanger adding additional heat to the air in the plenum.

The branch duct in the equipment van also branches toward the two ends, each of which has two similar outlet registers. These registers are all adjustable for volume which allows balancing them to get the desired flow in each area. The thermostat in the equipment van is a single-stage heating and cooling type and controls the temperature as a subordinate second zone. It cannot turn on the heating system, but opens dampers in the two ducts if heating or cooling is required.

The return air from the display van comes through a large grill near the floor and is transferred back through a similar flexible duct through the subfloor of the equipment van entering the return air plenum under the blower. The equipment van return is through grills in the floor around the periphery, with the space under the removable floor acting

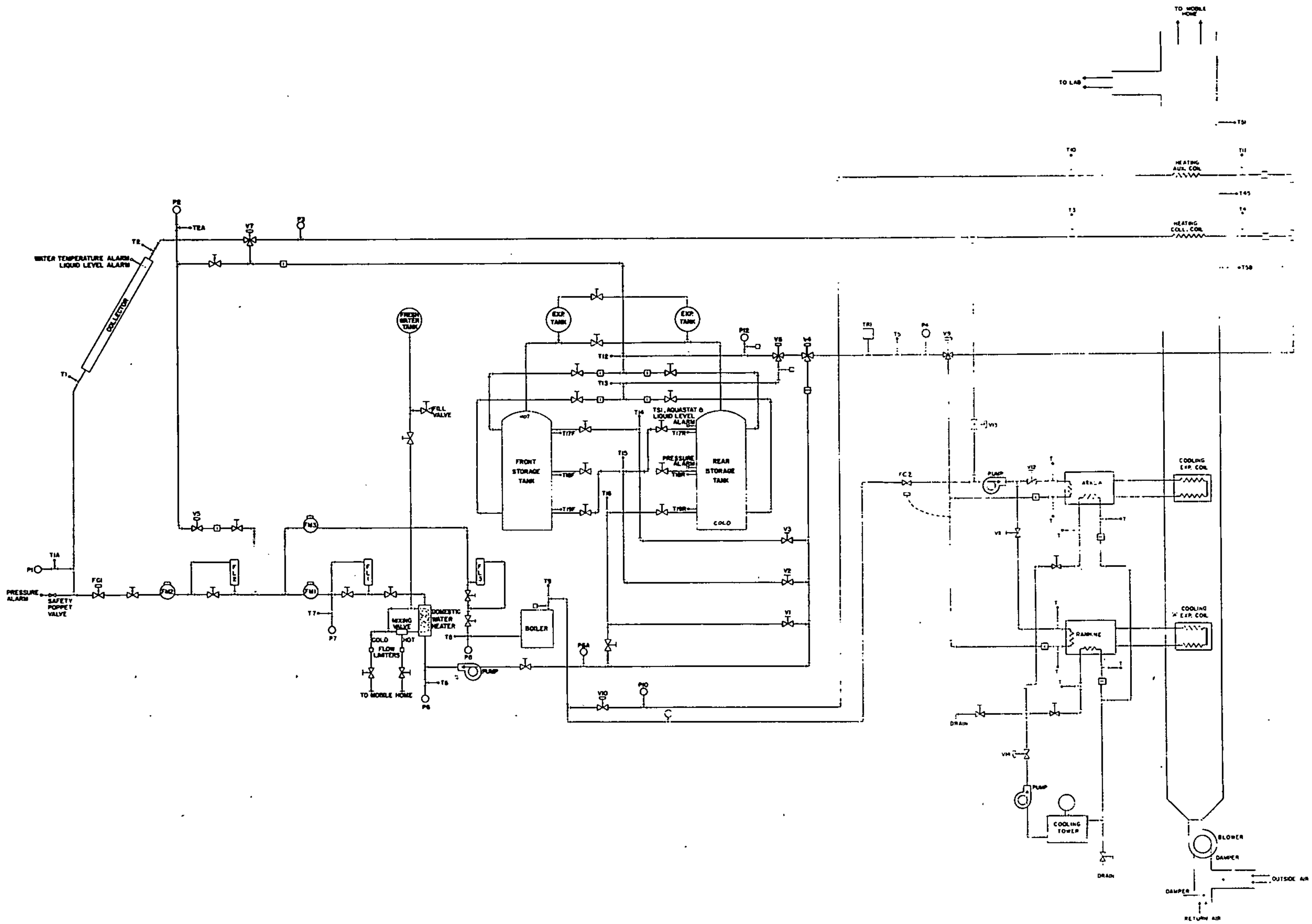


Figure 3-9. Fluid Flow Diagram Heating/Cooling

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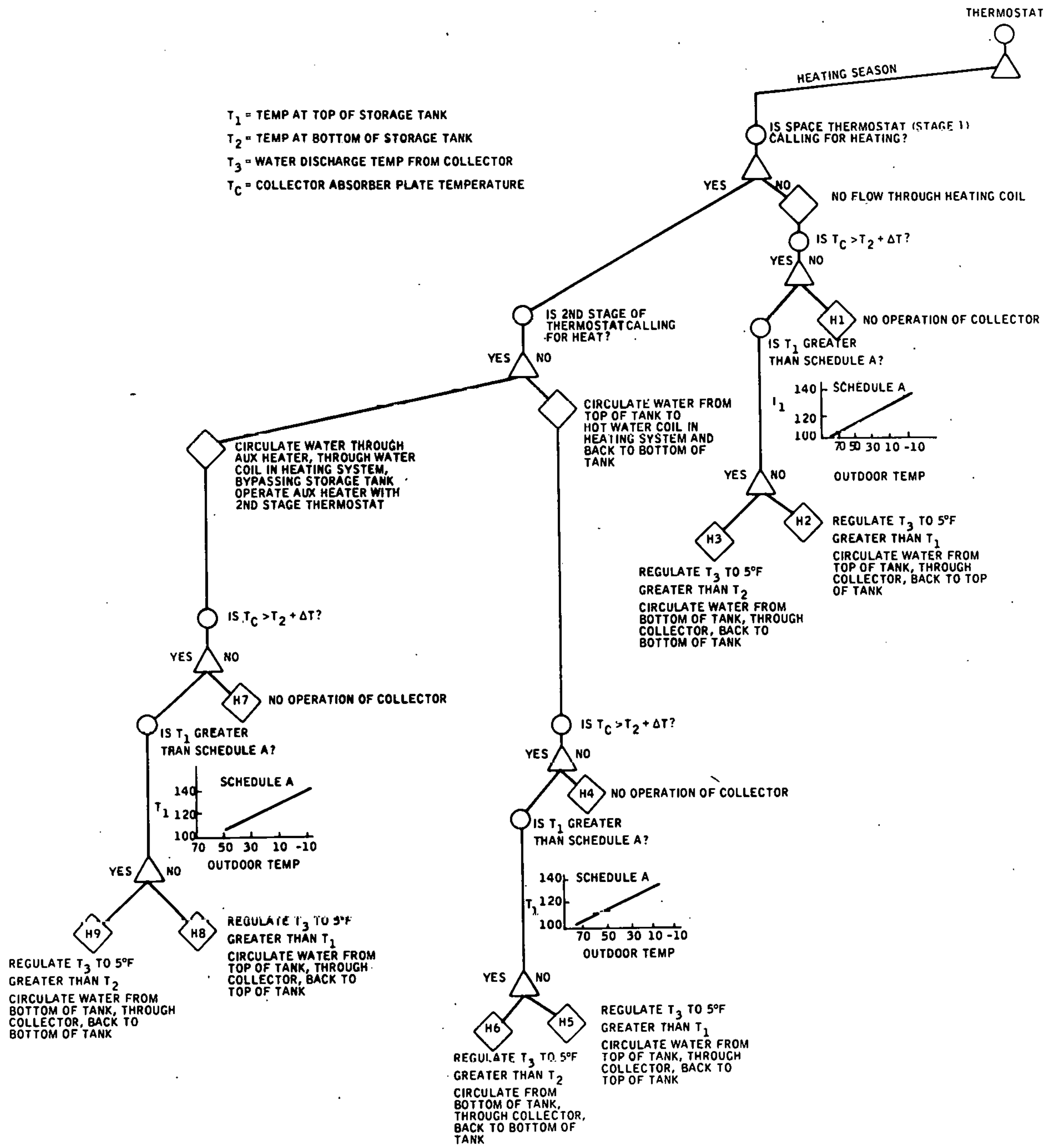

Figure 3-10. Logic Flow Diagram (Heating)

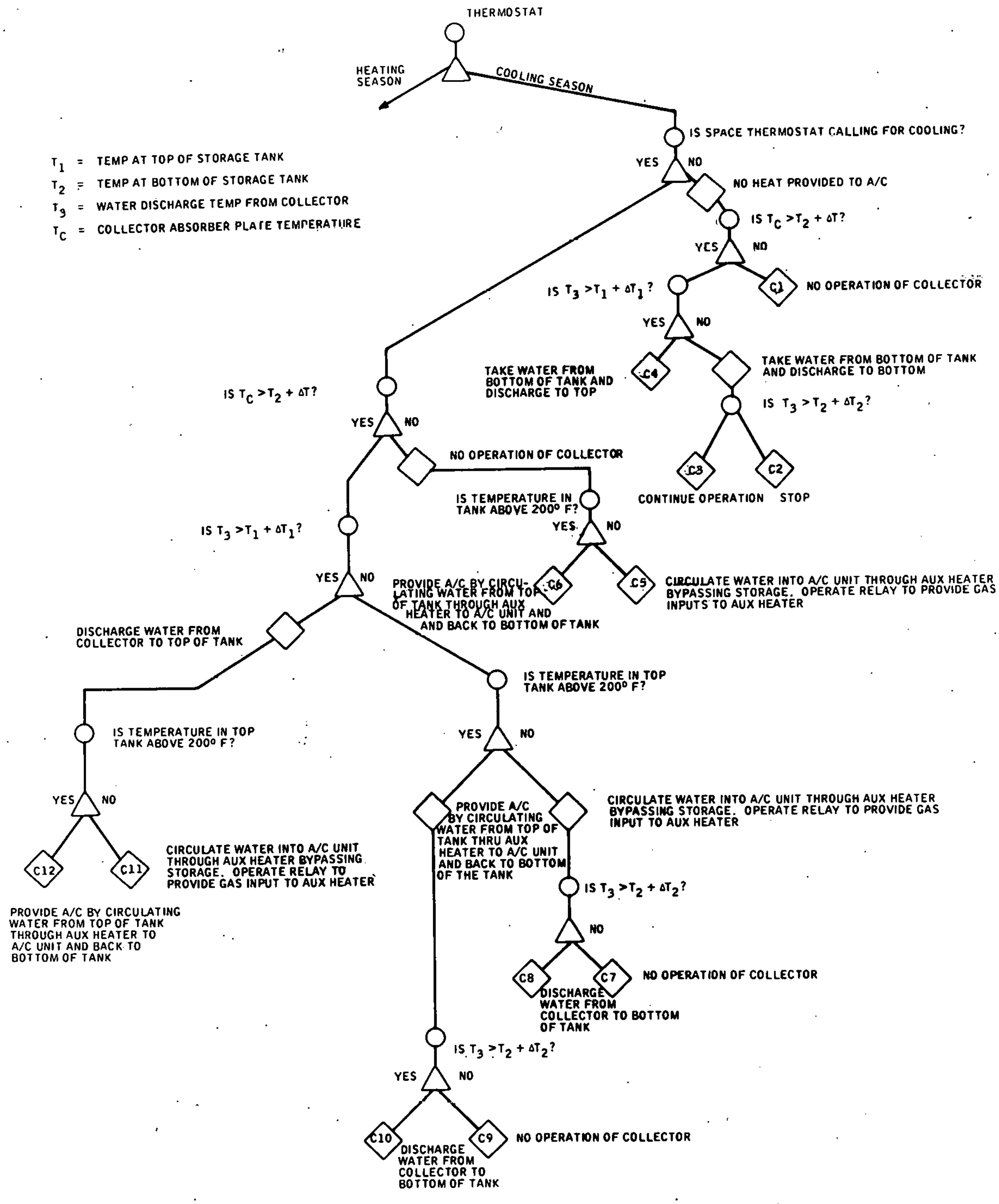

Figure 3-11. Logic Flow Diagram (Cooling)

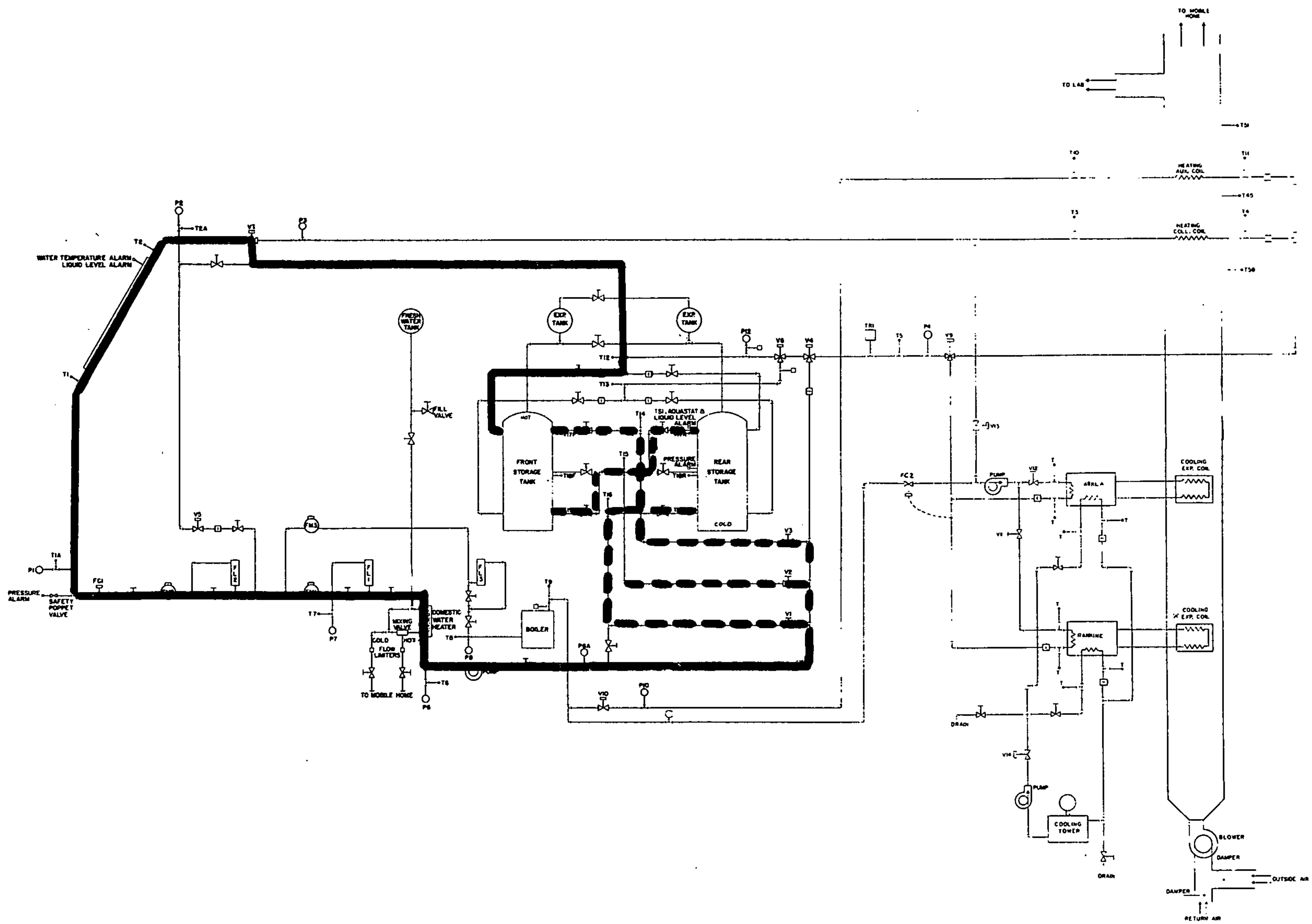

Figure 3-12. Collect and Store Solar Energy

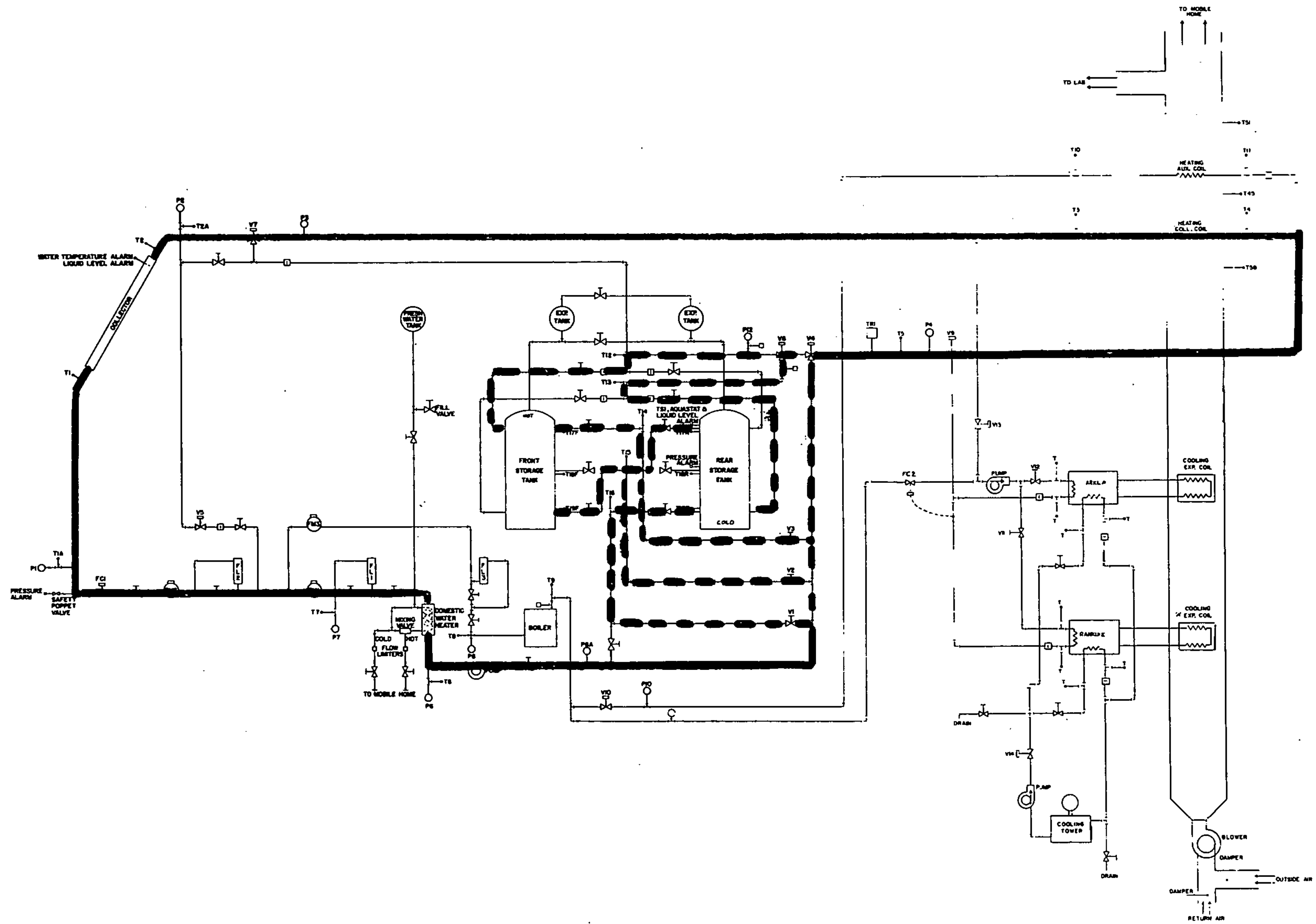

Figure 3-13. First Stage Heating - Collector Solar Energy.

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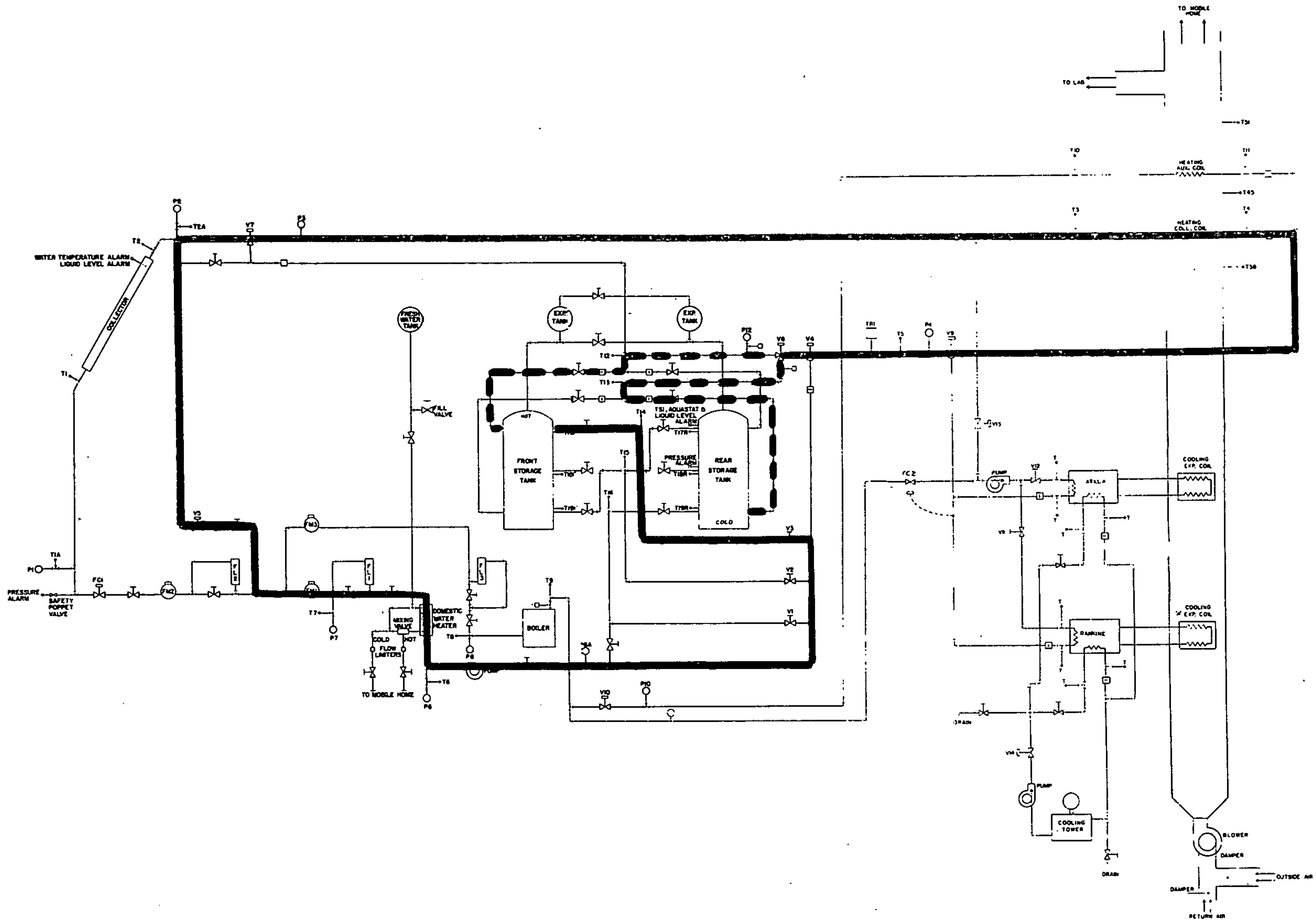

Figure 3-14. First Stage Heating - Stored Solar Energy

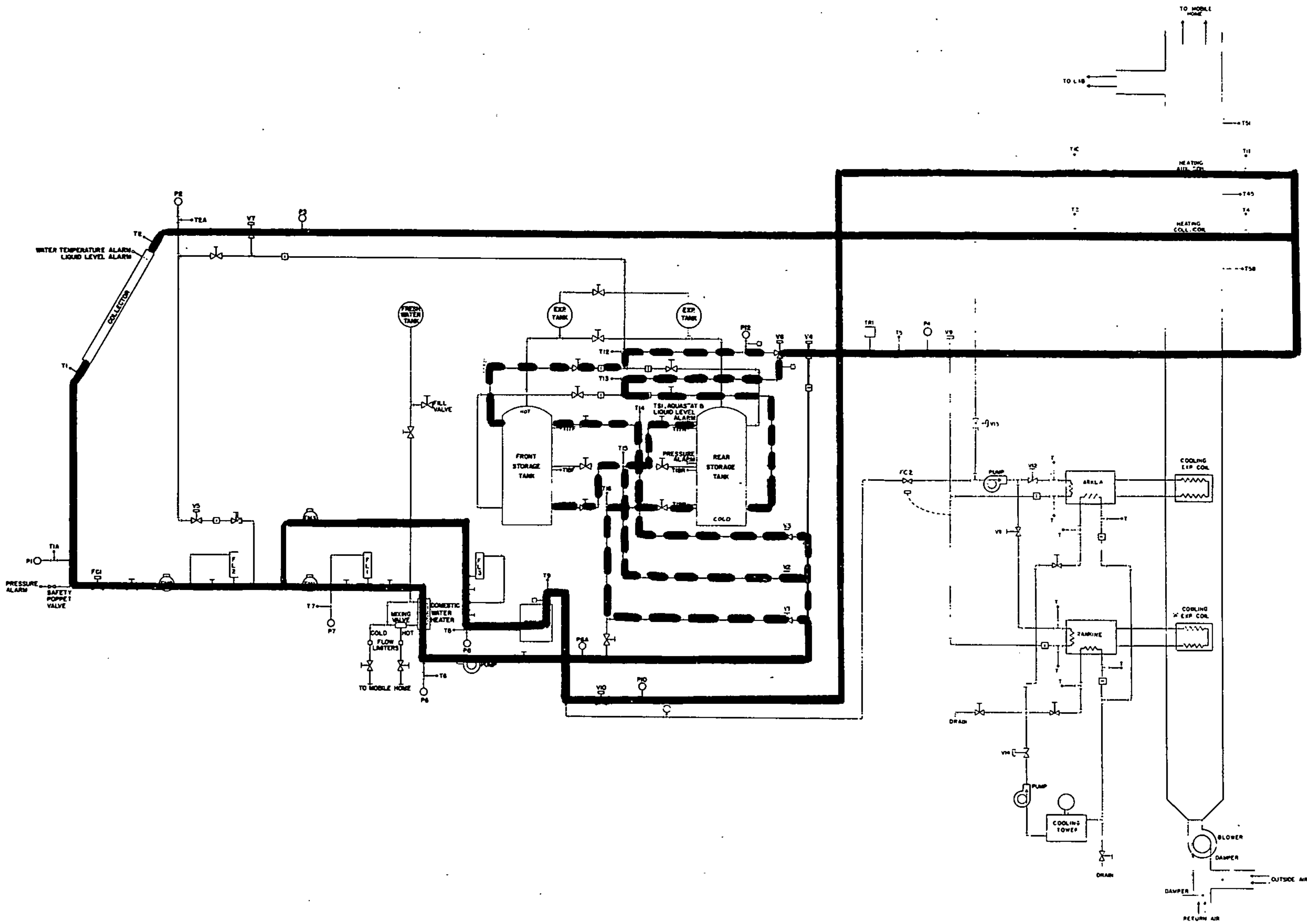

Figure 3-15. Second Stage Heating - Collector Plus Boiler

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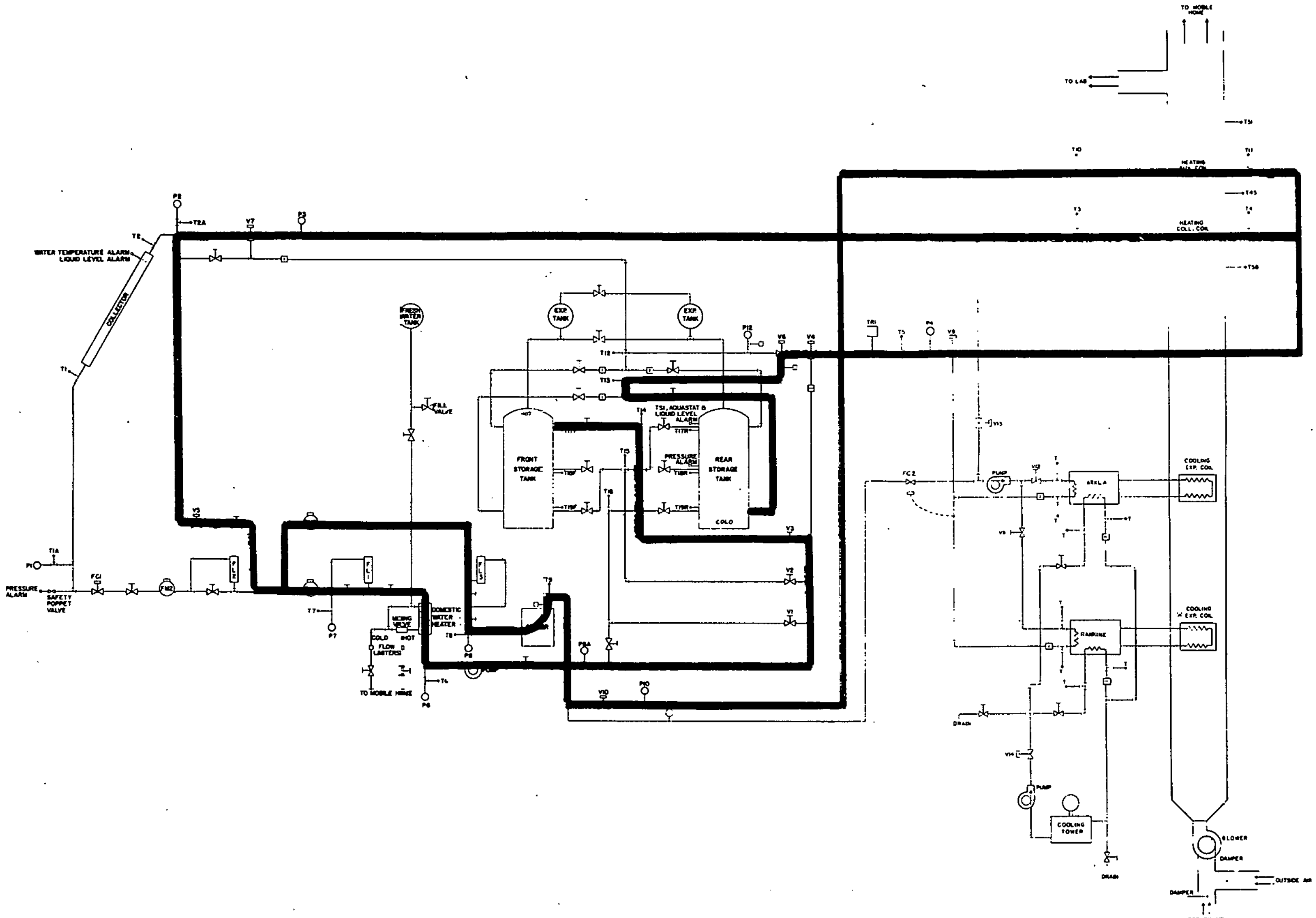

Figure 3-16. Second Stage Heating - Storage Plus Boiler

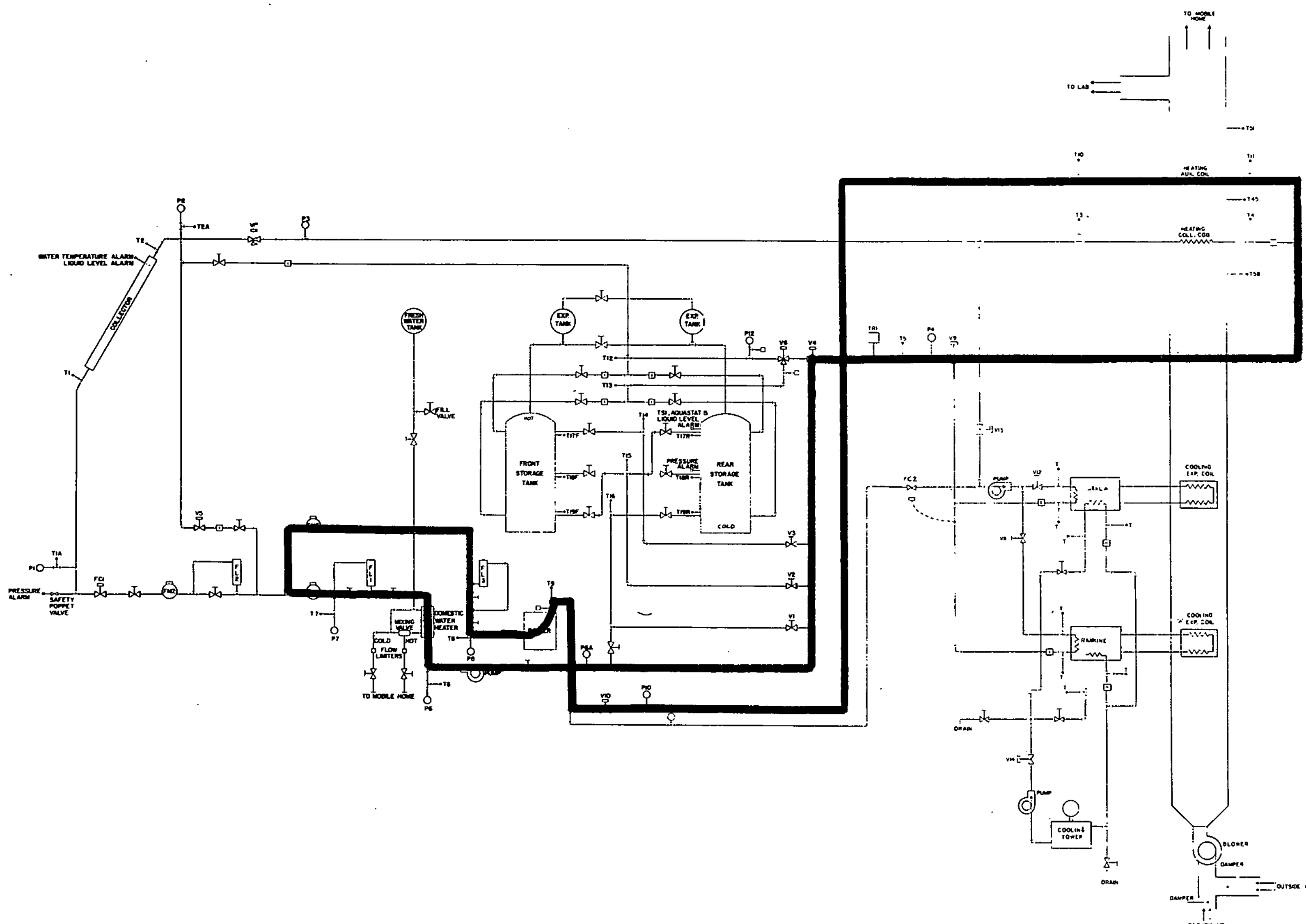

Figure 3-17. Second Stage Heating - Boiler Only

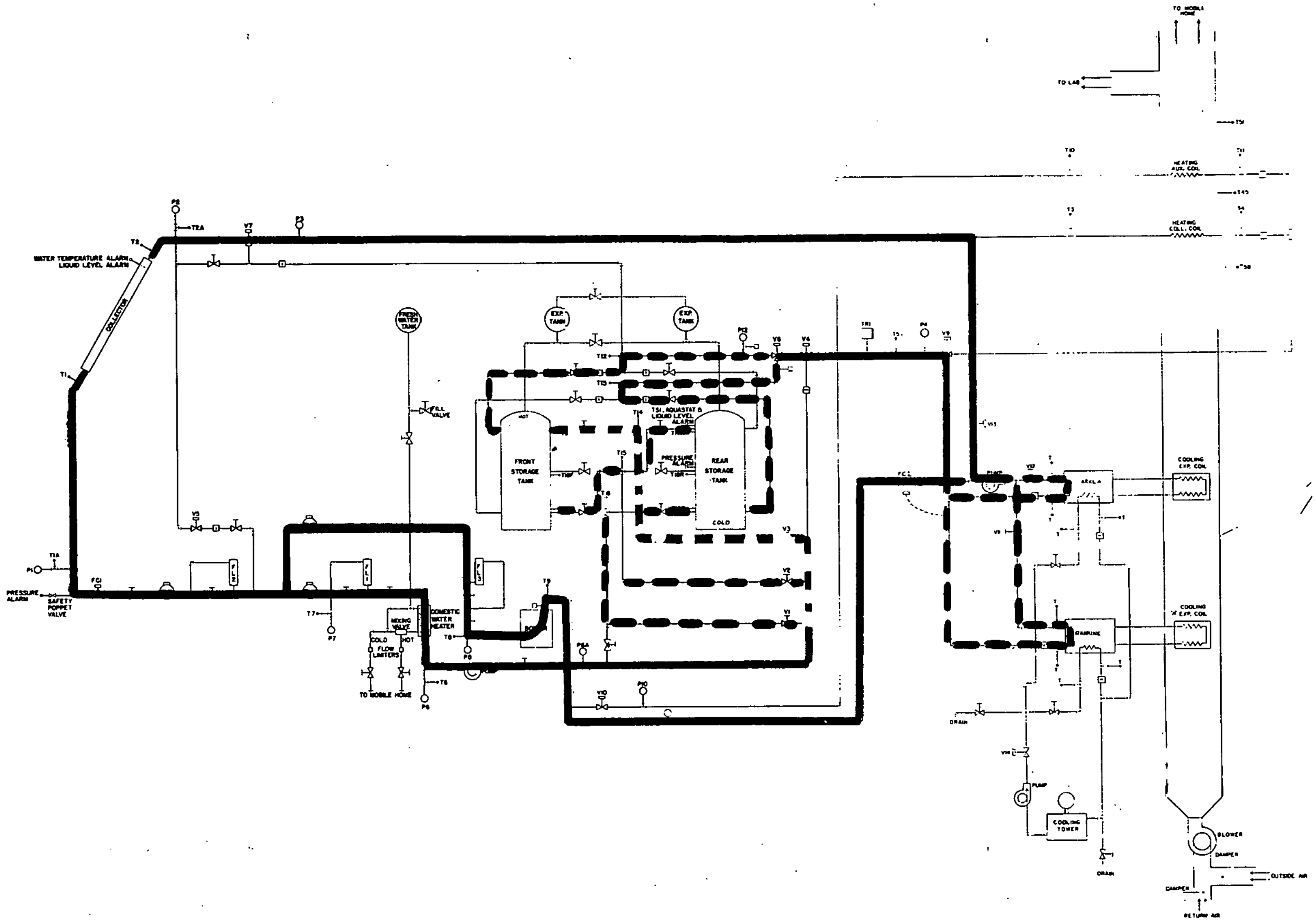

Figure 3-18. First Stage Cooling - Collector Solar Energy

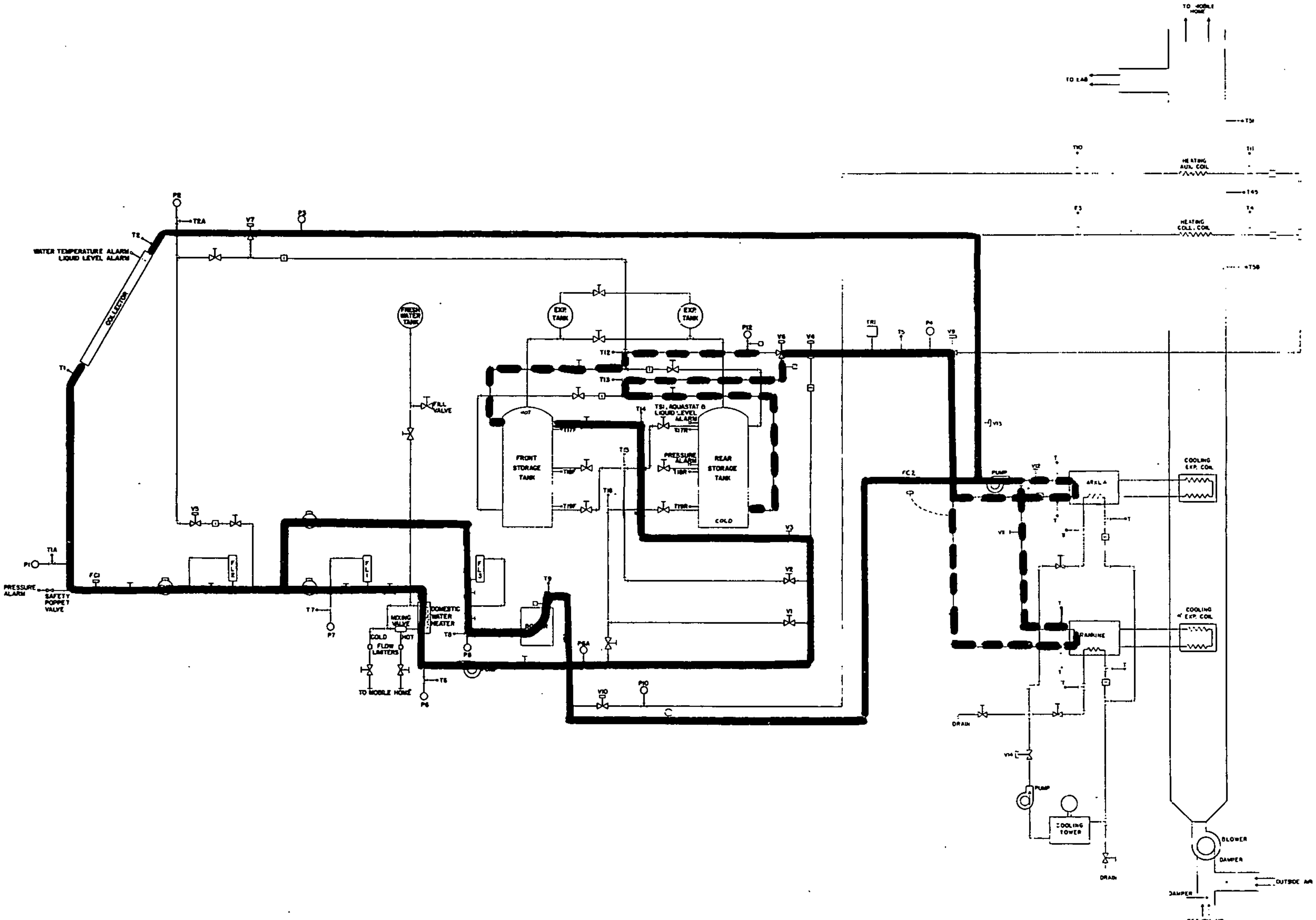

Figure 3-19. Second Stage Cooling - Collector Plus Boiler

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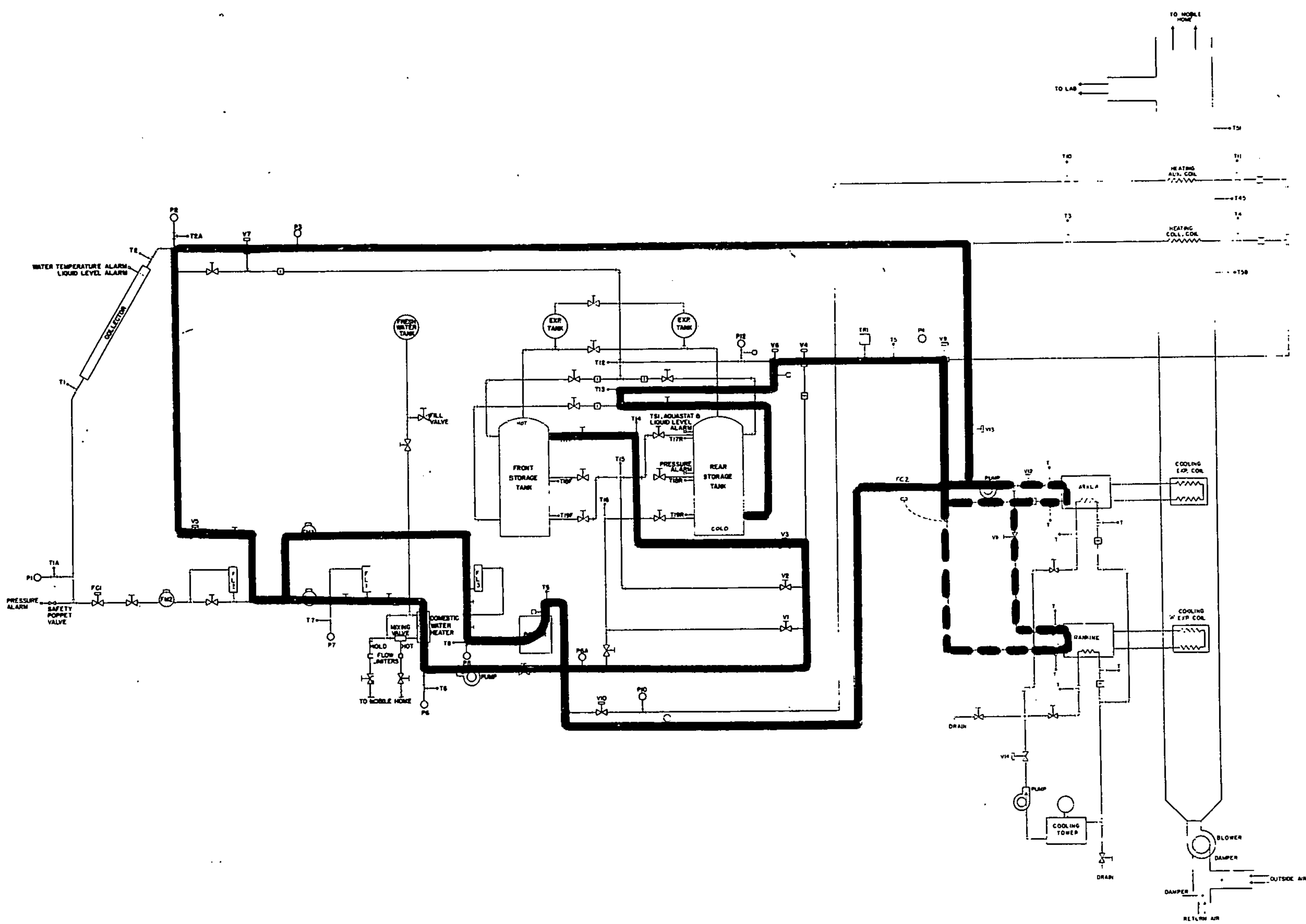

Figure 3-20. Second Stage Cooling - Storage Plus/or Boiler

as a large plenum. From there it enters the actual return air plenum through two manual sliding dampers covering openings on the plenum. From the plenum it returns to the blower through a washable metal air filter and an electronic air cleaner.

Heating Load

The two vehicles are well insulated to minimize heat loads. Both units have 3.5 inches of sprayed polyurethene insulation for a "U" factor of 0.07.

The following assumptions were used for calculating the heating load:

- "U" insulation rating is 0.07 for all exterior walls, floors, and ceilings.
- The external temperature is -30°F.
- The interior temperature is 72°F.
- Minimal air changes.
- 3.5 kw electric load: display, lights, control system, data ACB, etc.
- 12 persons in equipment van.
- 10 persons in display area.
- 4 persons in office area.
- The glass window areas have a "U" factor of 0.56.
- The exterior doors have a "U" factor of 1.13.

Using these assumptions, the heat load calculations are summarized in Table 3-2, which shows that at -30°F the total space loss is 36,240 Btu/hr. When the vans have people in them, outside air must be provided. This increases the heating load to 57,300 Btu/hr for approximately 26 persons. The trailer has a positive pressure to help prevent cold drafts from entering through the doors and windows.

Table 3-2. Heat Loads for Both Units

	Space Loss (Btu/hr)	AC/HR	CFM/OA	OA Load (Btu/hr)	CFM SA Used	CFM Exh. Used	Grand Total (Btu/hr)
Equipment Van	11,430	1.9	90	9,720	550	80	21,150
Display Unit Lounge	7,430	1.5	30	3,240	150	20	10,670
Display Area	14,920	1.37	75	8,100	320	65	23,020
Equipment Section	2,460						2,460
Totals	36,240		195	21,060	1020	165	57,300

Abbreviations used in table:

AC/HR - Air changes per hour
CFM - Cubic feet per minute
Btu/hr - British thermal units of heat per hour
OA - Outside air
SA - Supply air
EXH - Exhaust

Cooled Air Flow Paths and Controls

The air conditioning system is controlled by the thermostat in the display van, with the equipment van thermostat only opening and closing their dampers as required. When the first stage of the primary thermostat is made, the air economizer system is energized. This system opens a modulated damper in a duct leading from the return air plenum to the outside and modulates a damper in the return air duct. The amount these are opened or closed is determined by a temperature sensor in the return air plenum and an enthalpy sensor in the outside air duct. The purpose is to use outside air whenever possible to cool the space rather than operating the air conditioner. Thus, the enthalpy sensor measures how cool and dry the outside air is; if it is below the set point, the air is mixed with return air to hold the desired temperature.

If the outside air alone does not appear to be cooling the vans, either one of the air conditioners can be turned on with a manually operated push button. The system is still under automatic control as the first-stage thermostat contacts will operate the air conditioner automatically. The outside air economizer can then be manually closed to its minimum position at the operator's discretion. The minimum position is determined by the fresh air requirements and initially set within the air economizer actuator itself.

To allow outside air to enter, air must also be exhausted. This is done by a large dampered exhaust opening in the plenum under the floor of the equipment van and smaller openings above the ceiling at the rear of both vans. These dampers are weighted to open when the inside air pressure is raised, thus maintaining a positive pressure within the vans.

Provisions for two separate air conditioning systems have been made, which will be used individually and will allow comparisons of operation and efficiency of each of them for conditioning the same space over the differing climatic conditions. First-stage cooling will operate the air conditioners off solar energy. Second-stage cooling will fire off the auxiliary boiler to assist the solar energy or provide the entire energy for cooling.

The Absorption Cycle Refrigeration Subsystem

This subsystem will consist of a modified model 501 gas-fired, 3-ton air conditioning unit manufactured by ARKLA Air Conditioning Company, Evansville, Indiana. ARKLA, under subcontract to Honeywell, will replace the gas burner/generator with a hot-water, driven-heat exchanger.

Cycle of Operation

Single Coil -- Direct Fired Unit

The entire unit operates under a vacuum at all times. The absolute pressure within the generator and condenser, when on cooling, is of the order of 50 to 60 mm of mercury absolute, and the pressure within the cooling coil and absorber is 6 to 9 mm of mercury absolute. As a comparison, standard atmospheric pressure is the equivalent of 760 mm of mercury absolute. Therefore, it is quite evident that the entire system operates under a fairly high vacuum. This type of unit can also be used for a heating cycle, but for this usage and the modifications made it is used only for cooling.

Figure 3-21 shows the flow diagram for the cooling cycle. The system is charged with Lithium Bromide and water, the water being the refrigerant on the cooling cycles and the Lithium Bromide solution being the absorbent.

Referring to Figure 3-21 and considering the cooling cycle only, the generator contains a solution of Lithium Bromide in water. As heat is applied in the heat exchanger of the generator, it causes the refrigerant (water) to be boiled off. As this water vapor is driven off, the absorbent solution is raised by vapor lift action through tube (2) into the separating chamber (3).

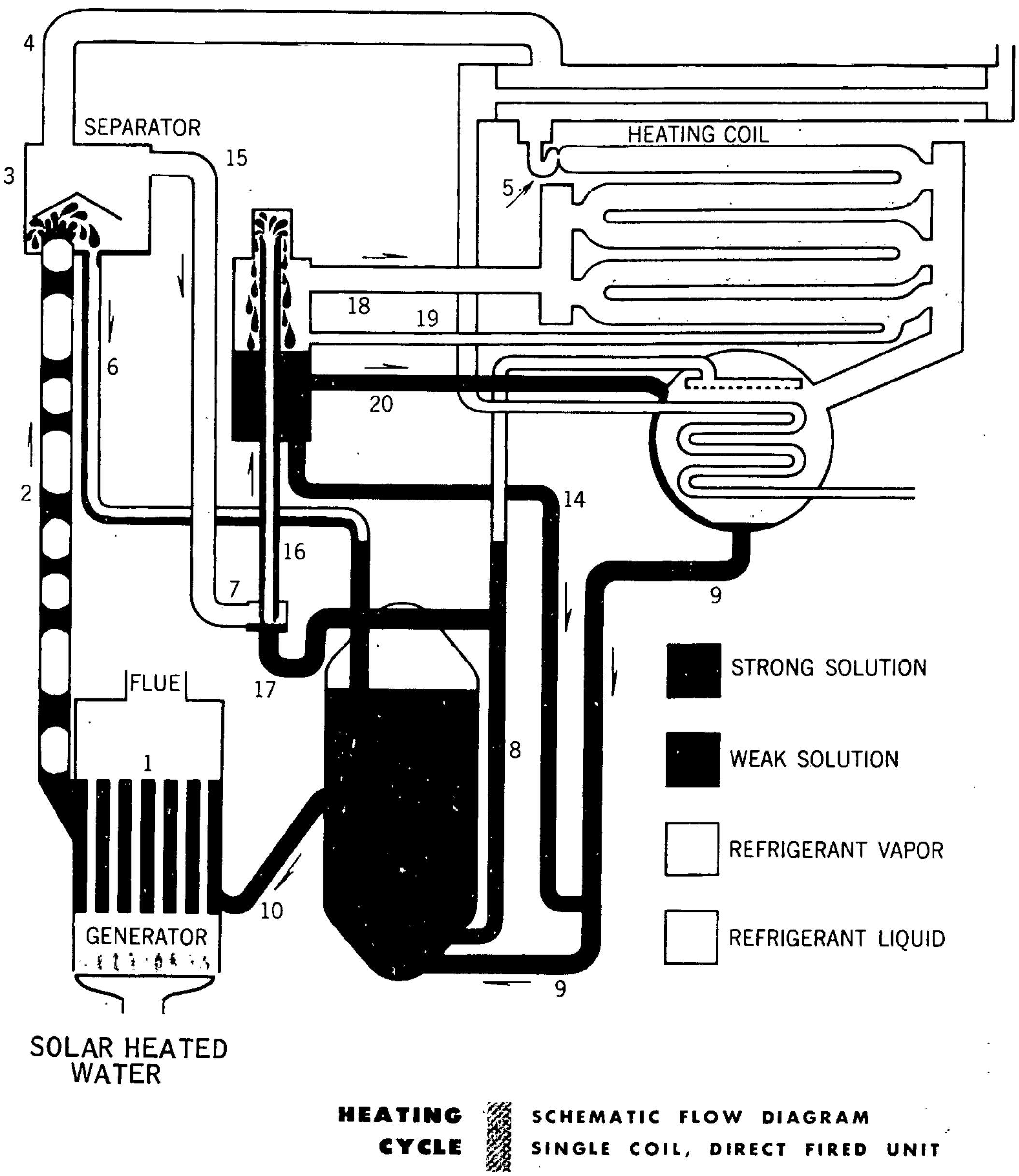

Figure 3-21. Arkla Air Conditioning Flow Diagram

Here the refrigerant vapor and the absorbent solution are separated by baffles. The refrigerant vapor rises through tube (4) to the condenser, and the absorbent solution flows by gravity through tube (6), through the heat exchanger, and then to the absorber. This circuit will be described in more detail; the refrigerant circuit will be described first.

The refrigerant (water vapor) passes from the separating chamber to the condenser through tube (4), where it is condensed to a liquid by the action of cooling water flowing through the condenser tubes. The cooling water is brought from an external source, such as a cooling tower.

The refrigerant vapor thus condensed to water within the condenser then flows through tube (5) into the cooling coil. Tube (5) contains a restriction which offers a resistance and therefore a pressure barrier to separate the slightly higher absolute pressure in the condenser from the lower pressure within the cooling coil. The refrigerant (water) entering the cooling coil vaporizes due to the lower absolute pressure (high vacuum) which exists within it. The high vacuum within the evaporator lowers the boiling temperature of water sufficiently to produce refrigeration effect.

The evaporator or cooling coil is constructed with finned horizontal tubes; and air being cooled flows over the coil surface. Evaporation of the refrigerant takes place within the cooling coil. The heat of evaporation for the refrigerant is extracted from the air stream, and cooling and dehumidifying are accomplished.

In the absorber, the solution absorbs the refrigerant vapors which are formed in the evaporator directly adjacent.

To explain the presence of the absorbent, it is necessary to divert attention back to the generator. The absorbent was separated from the refrigerant by boiling action. The absorbent then drains from the separator (3), down to the liquid heat exchanger, and thence to the absorber through tube (8). The flow of solution in this circuit can actually exist by gravity action alone, because the absorber is slightly below the level of the separating chamber. It is also aided by the pressure difference existing between the separator and the absorber.

The absorber is a cylindrical shell which contains a coil through which cooling water is circulated. The solution flowing into the top of the absorber is distributed over the entire outside surfaces of the coil so that a maximum area of absorbent solution is exposed to the refrigerant vapor which is flowing into this chamber from the evaporator.

It must be understood at this point that cool Lithium Bromide in either dry or in solution form has a very strong affinity for water vapor. It is because of this principle that the refrigerant vapor is absorbed back into solution again. The rate of absorption is increased at lower temperatures; therefore, a cooling water coil has been provided within the absorber shell. The resultant mixture of refrigerant and absorbent drains back through the heat exchanger through tube (9), to the refrigeration generator where it is again separated into its two component parts by boiling action, to repeat the cycle.

Because of the slightly higher absolute pressure in the separator, as compared to the cooling coil or evaporator, absorbent solution rises through tube (7), through the liquid trap (7), and up into tube (16), thereby forming a liquid seal so that refrigerant vapor cannot flow through tube (15) from the separator chamber. In this manner the cooling cycle is maintained.

The Rankine Cycle Subsystem

Barber-Nichols Engineering Co., Denver, Colorado, under contract to Honeywell, developed the 3-ton Rankine cycle solar powered air conditioner. This subsection describes the method of implementing a Rankine cycle engine to drive a mechanical air conditioner.

The Rankine Cycle subsystem is made up of two cycles, a Rankine power cycle and a refrigeration vapor cycle. The basic system concept is shown schematically in Figure 3-22. In the power cycle the working fluid is taken from the condenser in liquid phase and pumped to the maximum system pressure. It is then heated in the regenerator/heat exchanger, which is a cycle efficiency improvement device. The working fluid is then heated additionally by passing through the gear box and vaporized in the boiler using heat which is absorbed by the solar collector. After leaving the boiler, the working fluid is expanded and energy is extracted to provide the power to drive the compressor. After leaving the expander, the exhaust vapor is cooled in the regenerator while heating the liquid from the pump. From the regenerator the cooled vapor passes to the condenser where it is liquified before flowing to the pump.

The refrigeration vapor cycle shown in Figure 3-22 is of conventional design. The cooling is accomplished by evaporation of the low-temperature refrigerant in the evaporator. The vaporized refrigerant is then compressed by the compressor and passes onto the condenser where it is condensed to liquid form. After leaving the condenser, the refrigerant is expanded in the thermal expansion valve to reduce the pressure and control flow to the evaporator.

Air Conditioning Load

The air conditioning load for both vehicles is 3.34 tons of air conditioning. This is based on the following assumptions:

- "U" insulation rating of 0.07 for all exterior walls, floors, and ceilings.
- 7.5 CFM of outside air is provided for each person in the units.
- 12 persons are in the laboratory van.
- 10 persons are in the display portion of the van.
- 4 persons are in the lounge area.
- The glass windows have a "U" factor of 0.56.
- External doors have a "U" factor of 1.13.
- Both areas operate at a positive pressure so that there are no infiltration gains.
- No solar gain is figured on walls or glass.
- No solar gain is figured on the roof of the equipment van because the collector panel shades it.
- The outside temperature is 17°F higher than the inside temperature.
- The air supply has a 55°F dry bulb temperature.
- The dry bulb temperature of the supply air will rise 23°F.
- Assume that all people in the vehicles are quiet.

From these assumptions, the air conditioning load may be calculated as shown by the area breakdown in Table 3-3.

Table 3-3. Air Conditioning and Load Calculation*

	Total Trans. and Solar	Body Heat Gains		Equipment Heat Gains	Total Sensible Heat Gain	Total Latent Heat Gain	Total Heat Gain	Total Air Supply (CFM)	Vent Air Required (CFM)	A/C Load O.A.	Total A/C Load	
		Sensible	Latent								Btu/hr	Tons
Equipment Van	1448	2640	3360	9,738	13,826	3360	17,186	546	90	3420	20,610	1.7
Display Unit Lounge	1387	880	1120	1,468	3,735	1120	4,855	148	30	1140	5,995	0.5
Display Area	1732	2200	2800	4,161	8,093	2800	10,893	320	75	2850	13,743	1.14
Totals	4567	5720	7280	15,367	25,654	7280	32,934	1014	195	7410	40,348	3.34

*All heat loads are in Btu/hour unless otherwise specified.

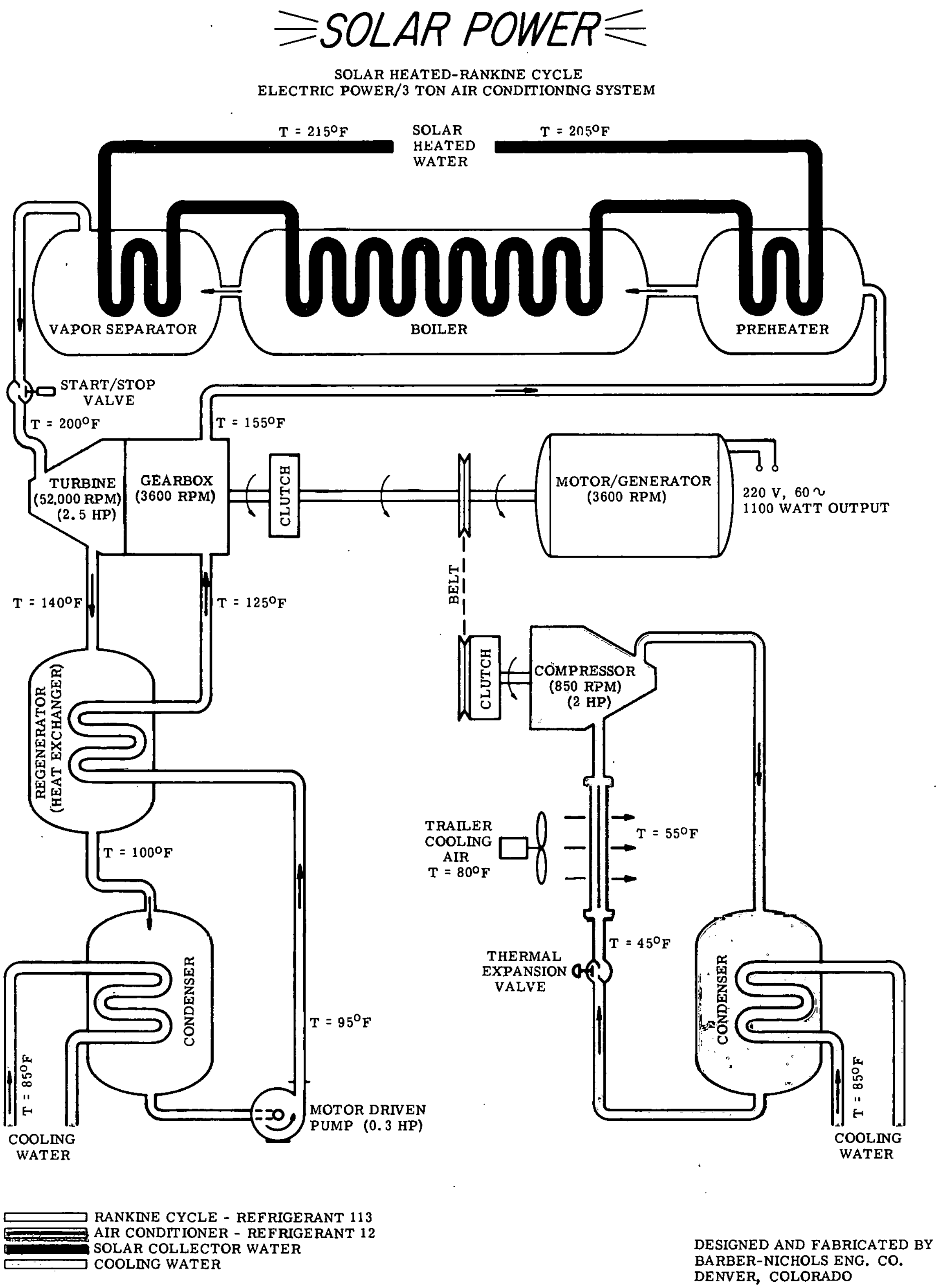

Figure 3-22. Rankine Cycle Schematic

SOLAR HEATED FLUID HANDLING SYSTEM

Fluid Selection

Water is the basic fluid for this system since a study of various heat transfer mediums is not a design objective. However, it must be kept from freezing, requiring the addition of an anti-freeze. A product of the Dow Chemical Company called Dowtherm SR-1 was chosen, which is an inhibited ethylene glycol-based product designed specifically for use in heat transfer systems. It is an available commercial product, gives better corrosion protection than automotive anti-freeze, and its physical characteristics are available for mechanical design and heat transfer analysis. SR-1 is not as good a heat transfer fluid as water, so it is desirable not to use a higher percent mixture than necessary; also, the differences are not as pronounced at higher temperatures (about 10 percent poorer at 200°F for the amount used).

Freezing protection to 0°F was the design compromise between the expected low temperature and the performance degradation of the system. Figure 3-23 shows the relationship between the freezing point of SR-1 and the percent by volume of the solution. With the system capacity of approximately 1000 gallons, 330 gallons (6 drums) of SR-1 were put into the system. Figure 3-24 gives the relationship between volume and weight for SR-1 showing that the 33 percent by volume is equal to 36 percent by weight.

Design Flow Rate

An average heat collection rate of 55,000 Btu/hr was selected at the start of the project to be the design goal for the system. Assuming water at 8.34 lb/gal to be the heat transfer medium and a ΔT of 10°F through the heat exchangers, the design flow rate was set at

$$\frac{55,000}{60 \times 8.34 \times 10} = 11 \text{ gpm}$$

After the vehicle design was firmed up, the heating load was recalculated as discussed in the previous section and found to be 57,300 Btu/hr at -30°F. Coupling this with the use of anti-freeze raises the question of the net result. From Figure 3-25 the specific heat of 36 percent by weight of SR-1 at a heating temperature of 160°F is 0.92 Btu/lb/°F, and Figure 3-26 shows the density to be 8.53 lb/gal. Then with the same temperature drop the heat output is

$$11 \times 60 \times 0.92 \times 8.53 \times 10 = 51,750 \text{ Btu/hr}$$

Due to the many factors involved, it was decided not to change the initial design.

Pipe Selection and Sizing

For ease of fabrication and potential modification, copper tubing was selected for use in this system. The pipe diameter is dependent upon the flow rate, flow velocity, and allowable pressure drop. Too high a flow velocity will frequently cause a noisy system due to turbulence and cavitation in the elbows and fittings, while too low a velocity will not carry along any trapped air and return it to the expansion tank. The recommended design range is a flow velocity between 2 and 4 feet/second. Charts such as shown in Figure 3-27 are available showing the relationship between flow rate, pipe size, flow velocity, and pressure drop for water and other fluids.

The chart used at the time was for water in copper tubing, which gave a pressure drop of 1.37 psi/100 feet at a velocity of 3.25 feet/second for a flow rate of 11 gpm in 1.25-inch tubing. The addition of anti-freeze does have a noticeable effect in raising the pressure drop at ambient temperature and below; but, at the working temperature of this system it is quite comparable to water.

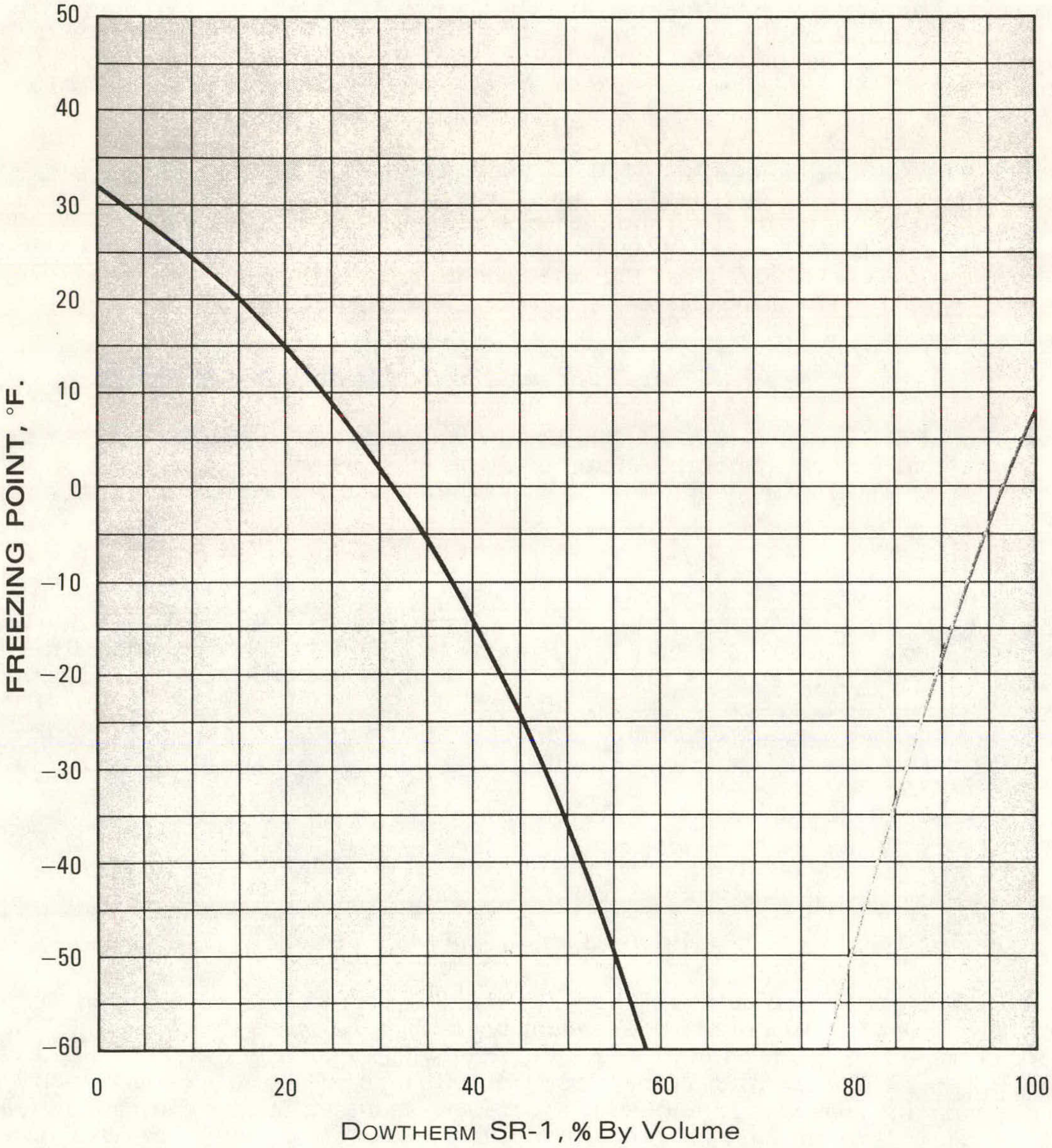

Figure 3-23. Freezing Point of Dowtherm SR-1

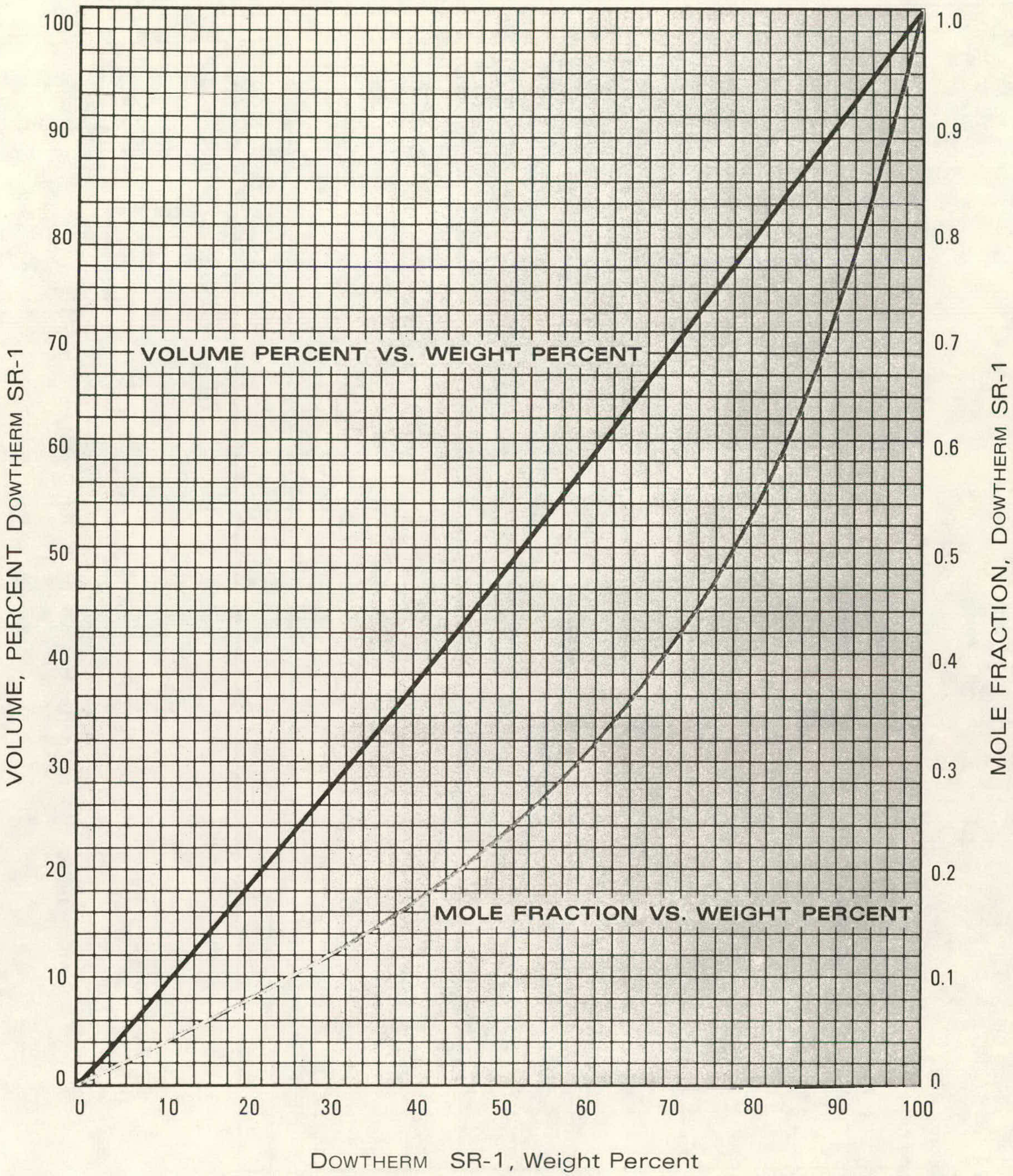

Figure 3-24. Conversion Chart for Dowtherm SR-1

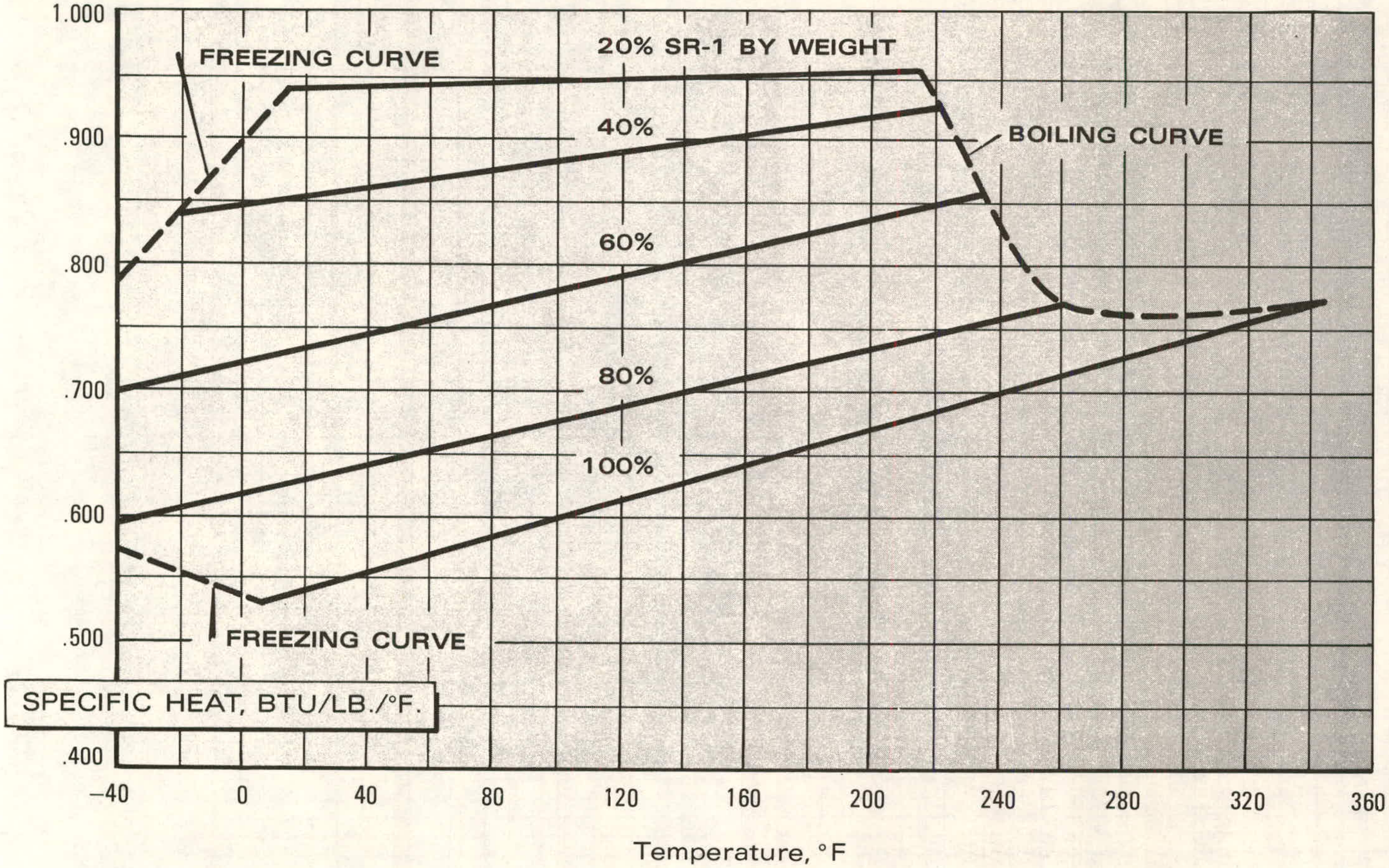

Figure 3-25. Specific Heat of Solutions of Dowtherm SR-1

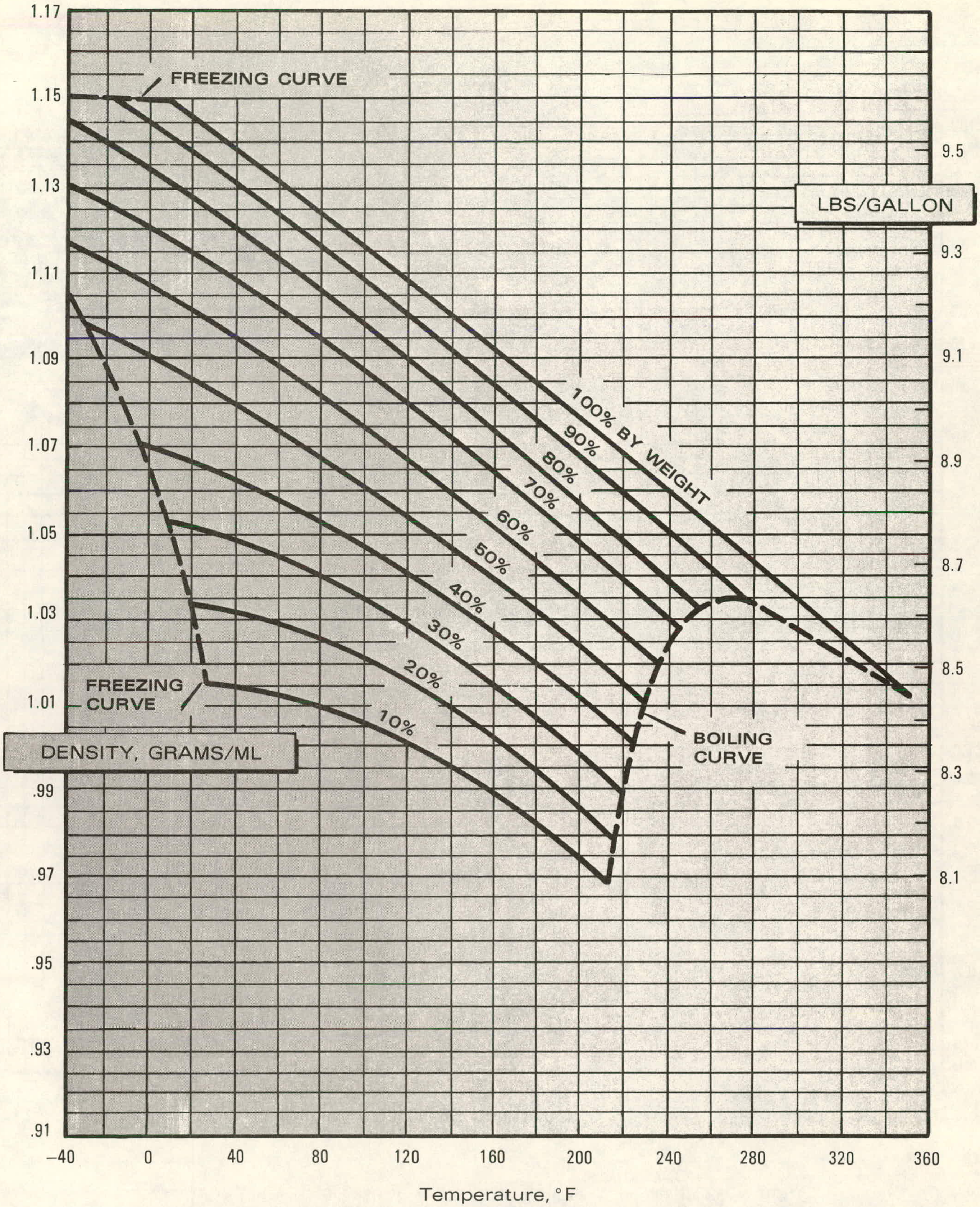

Figure 3-26. Densities of Solutions of Dowtherm SR-1

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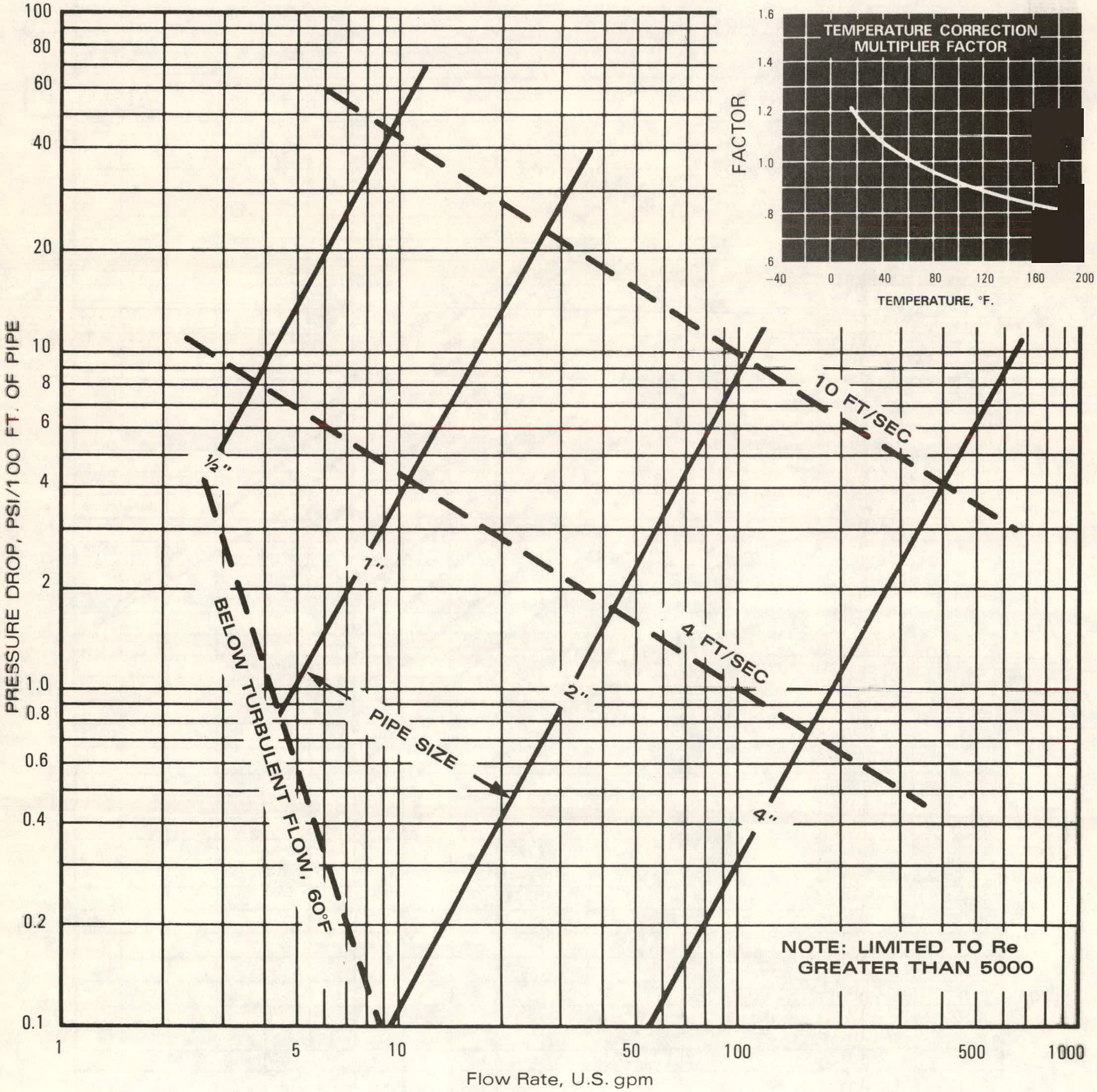


Figure 3-27. Pressure Drop versus Flow Rate - Dowtherm SR-1

Storage Tanks

Due to space limitations, the system fluid capacity was limited to 1000 gallons. One cylindrical tank standing on end would have been desirable, but the diameter was too large for the space available, and an oval cross section was not feasible for a pressurized system. The result was two tanks 42 inches in diameter with an overall height of 84 inches, as shown in Figure 3-28, which have a capacity of approximately 475 gallons each.

The steel tank has a wall thickness of 0.25 inch, and the ASME type dome ends are 5/16-inch thick. Multiple inlet and outlet openings are provided at the top, center, and bottom on opposite sides of the tanks by 1.25-inch diameter, female-threaded flanges. A row of 0.5-inch openings from top to bottom are for instrumentation temperature sensors, and an additional two at the top are for control system sensing. A 0.75-inch opening at the center of the bottom end provides for a drain, and a similar hole in the top end is for the pipe leading to the expansion tanks. To promote and maintain temperature stratification in the tanks, perforated metal baffles span the cross section. Two are just above and below the center outlet, and one each are below the top and above the bottom outlets. These are bolted in place to eliminate breaking welds from thermal expansion of the metal.

When the storage and expansion tanks were installed, they were interconnected by valves and plumbing so that they may be used independently or simultaneously.

Expansion Tanks

Every system such as this must have an expansion tank to accommodate the change in volume with changes in temperature. Theoretically it can be tied into the system at any location, but certain considerations must be taken into account. It can be vented to atmospheric pressure, but if so it must be located several feet above the highest point in the system, which is not practical for this installation. A vented tank also allows oxygen to enter the system which promotes corrosion and more rapidly depletes the inhibitors in the anti-freeze. Therefore, this is a closed system.

The junction point of the tank to the system becomes the point at which the pressure is constant whether or not the pump is running, which may cause undesirable pressure extremes at other locations such as the pump inlet and the safety blow-off valves. The pump inlet must always have a positive pressure head or it will not pump to capacity and may cavitate, leading to early failure, which could happen with the tank on the outlet side. In general, the best location is on the inlet side of the pump with as few fittings as possible between. This location also results in the smallest required size tank.

Another function of the expansion tank is to collect air trapped in the system. This is best done if the connection is at a tank, boiler, special air separation device, or other point of low flow rate. However, the tank connection must be located so that it is always in the operating line or it will not maintain constant pressure. This solar heat collection system has several separate flow paths that may or may not be open, but they all lead to the storage tanks. This is one of the reasons the storage and expansion tanks are located immediately upstream of the pump. One of the heating modes does bypass storage, but a small-diameter line still connects to storage to ensure that the pressure is equalized.

The ASHRA handbook contains a complete discussion of these factors and examples for calculating the required volume. The factors governing the sizing of expansion tanks are the same for all systems. They are:

- The system volume of water
- Maximum average water temperature
- The fill pressure of the system
- Tank location in the system
- Maximum operating pressure

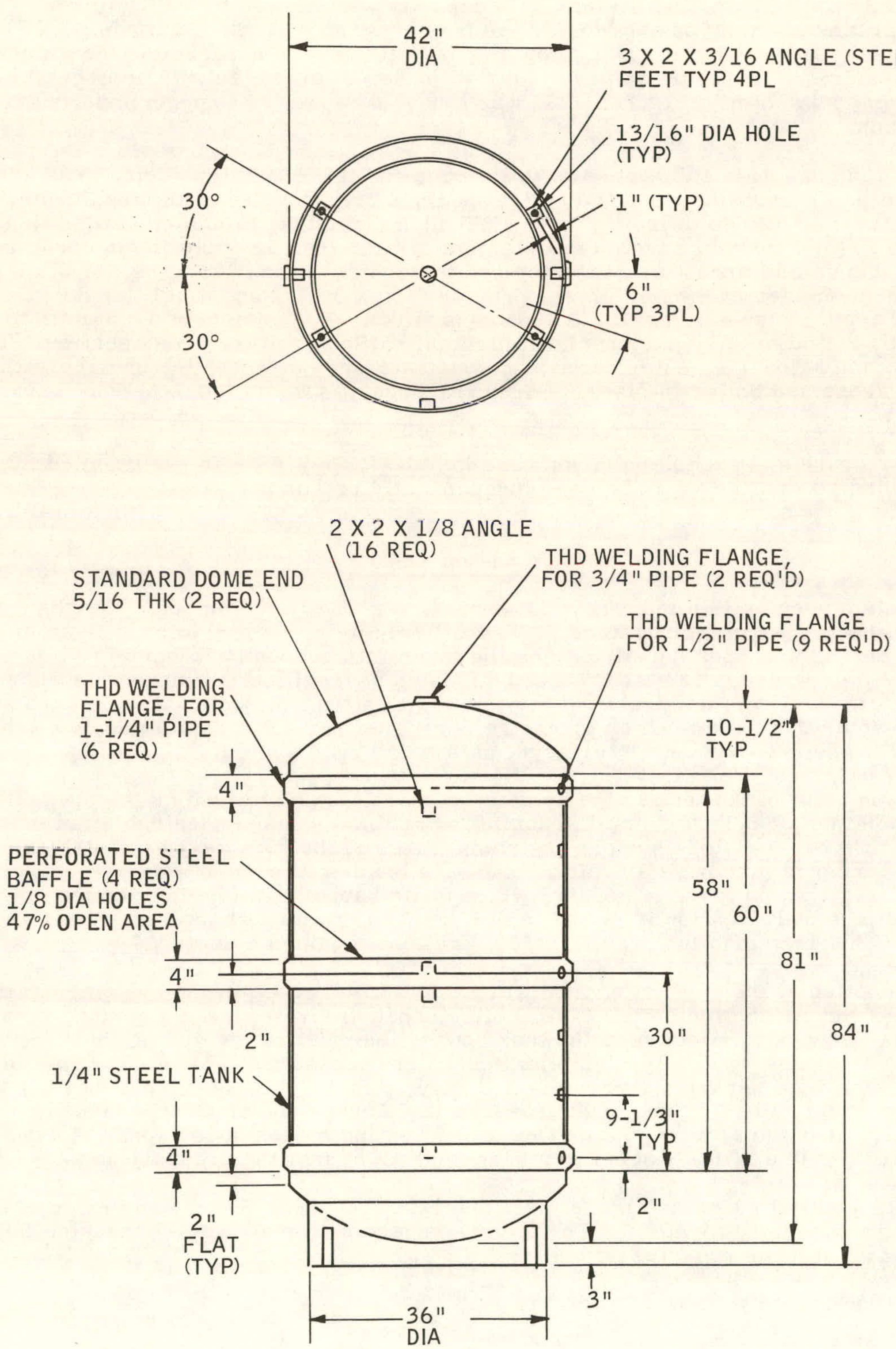


Figure 3-28. Storage Tank

The minimum required volume for the expansion tank in a closed system is determined by the following equation:

$$V_t = \frac{(0.00041t - 0.0466)}{P_a/P_t - P_a/P_o} \times V_s$$

where

V_t = Minimum gallonage of the tank

V_s = System volume in gallons

t = Maximum average water temperature in °F

P_a = Absolute pressure in tank when water first enters, usually atmospheric (14.7 psi at sea level)

P_t = Absolute pressure in tank created by initial fill, psig fill pressure to force the water to the highest point in the system plus atmospheric

P_o = Maximum absolute operating pressure, maximum psig pressure at tank plus atmospheric

For the design calculation, system volume is 1000 gallons, and the operating temperature of 200°F was used as the maximum average water temperature. To provide good venting and prevent boiling at the top of the panel, a 4-foot H_2O pressure head (1.73 psi) was assumed. Added to this is the height difference of 7.6 feet down to the expansion tank, making the initial fill pressure 5 psi gauge, or 19.7 psi absolute. The maximum operating pressure is determined by the pressure rating of the equipment, which is usually the safety valve setting which, for the boiler used is 30 psi. If the safety valve is not on the direct line to the expansion tank the pressure effects of the pipe, pump, and equipment must be taken into account, which in this case is a net lowering of the tank pressure of 5.9 psi. The height difference between the safety valve and expansion tank of 7.25 feet for this installation also lowers the pressure an additional 3.1 psi giving a total maximum operating pressure of 35.6 psi absolute. Then the equation becomes:

$$V_t = \frac{(0.00041 \times 200 - 0.0466)}{14.7/19.7 - 14.7/35.6} \times 1000$$

$$= \frac{0.0354}{0.747 - 0.413} \times 1000 = 106 \text{ gallons}$$

The thermal expansion of SR-1 anti-freeze, shown in Figure 3-29, is approximately 10 percent greater than water; therefore, two 60-gallon tanks were ordered and installed in the system. Due to the limited space available this is the largest capacity that can be installed without major modification. When it was found that the solar collector panels were to be run at a maximum pressure of 30 psi, it became necessary to operate it as a pressure monitored system or greatly enlarge the tank capacity, which was not feasible. With a pressure monitored system, the operating personnel must raise or reduce the system pressure as necessary to maintain it within the operating limits.

Auxiliary Heater

To provide continuing heat when solar energy is not available, an auxiliary heater, or boiler, is provided in the system. The boiler selected is an American Standard G-24 gas fired unit which has a net I=B=R rating of 62,600 Btu/hr water heating capacity.

For this installation, the boiler was fitted with the proper gas burners and ports for propane. Honeywell controls on the boiler allow setting the boiler water temperature to be maintained and a high limit gas shut-off control. An internal baffle arrangement near the top water outlet helps collect air driven out of the system by the boiler heat and vents it to the safety valve opening. The safety valve is set to the boiler rating of 30 psi. Manually opening the safety valve is the method used to release the air since the system design cannot lead it to the expansion tanks.

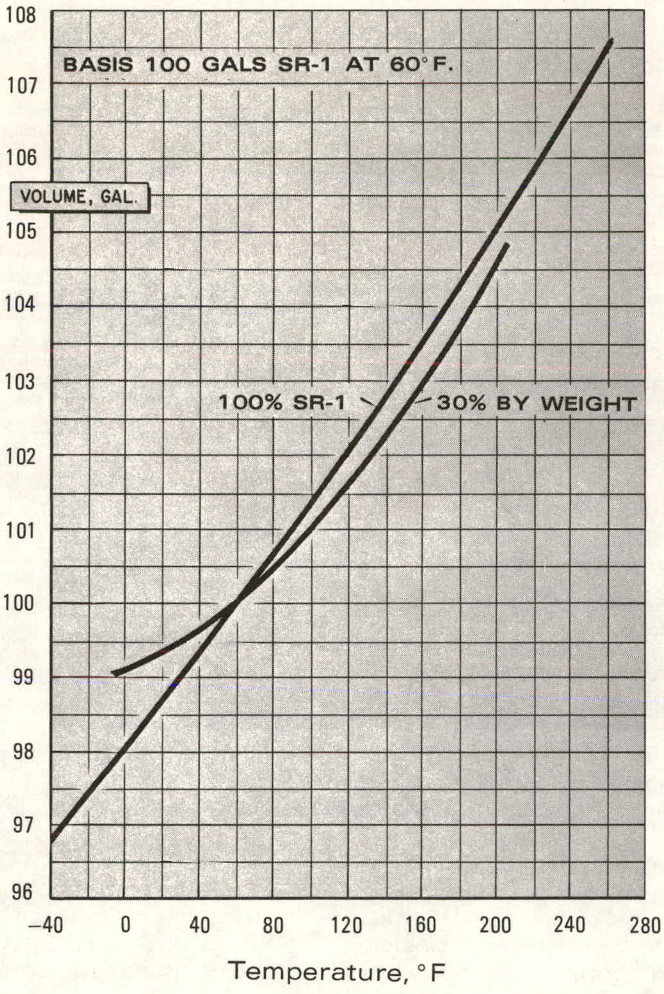


Figure 3-29. Thermal Expansion of Dowtherm SR-1

Water-to-Air Heat Exchangers

Two separate heat exchangers are located in the heating system to make the most efficient use of the available energy. The air enters the one utilizing solar energy first and is warmed to some degree, even if it is from only moderately warm storage. If this cannot maintain the space temperature, the second stage of the thermostat turns on, which starts the boiler and also opens the line to the second exchanger. If the return fluid is hotter than the storage tank, it is bypassed to avoid heating it from the boiler.

The heat exchangers used are 18 inches by 24 inches to fit the plenum chamber with 2-row, fin and tube construction manufactured by the Trane Company. This size and type construction was selected from the required air flow and heating load calculations.

Motorized Automatic Valves

The valves and motors to drive them used in this system to change between flow modes are Honeywell stock items (Figures 3-30 through 3-33). They are all 1.25-inch diameter to match the piping except for the solar collector flow control valve (FC1). This valve was sized to be 0.75 inch since it is a modulating valve, and the reduced size gives more linear control of flow rate through it at intermediate valve positions.

The control of the valves is from the electronic logic circuitry, and most of them are spring return. The spring return position was chosen to be fail safe to allow heat to be supplied and not cause flow blockage in case of power loss. The ones that are not spring return may be manually positioned, and will return automatically to the command position when power is restored.

Other Valves

Numerous other types of valves are installed in the system for specific functions. A balance valve is in each major flow circuit to be positioned such that the system flow is equal in all modes of operation. Gate-type shut-off valves are located where it may be desired to isolate certain parts of the circuit such as the storage tanks, and around the pump and flowmeters to allow removal for modification and repair. Air bleed valves are installed in all of the high points to facilitate removal of trapped air that would collect and block proper flow. Pressure safety valves are located on the collector line and on the boiler to relieve any excessive pressure that may be generated. The one on the boiler is also designed to be used for air bleeding since the boiler has a built-in air trap. The boiler safety valve is permanently set for 30 psi, and the collector valve is adjustable between 15 and 40 psi.

Pump Selection

The design system flow rate, pipe size, plumbing layout (including all valves, tees, elbows, etc.), and the pressure drop across each of the heat exchangers must be known before the pump can be selected. From various tables and curves the plumbing fittings are converted to equivalent lengths of straight pipe which added together allows calculation of the pipe friction pressure drop for the given size and flow rate for each flow path. To this is added the pressure drop through the heat exchangers and any other equipment to get the total pressure drop for the assumed flow rate.

Every pump has a performance curve similar to those shown in Figure 3-34. The point for the required head pressure for the design flow rate plotted on the pump curve determines the required pump size. For solar heating system, the Bell & Gossett 60-11 pump with a 5.125-inch impeller and 1/4 hp motor was chosen as being adequate for the system.

HONEYWELL **TRADELINE** CONTROLS

V5045A VALVE BODY

APPLICATION

The V5045A is a single-seat valve used with the V2045A Valve Motor to form an electric-motorized valve assembly. The combination motor-valve provides on-off control of the flow of low-pressure steam, or hot or chilled water.

Bonnet design of the V5045A allows the V2045A motor to be swiveled in any direction. The Teflon valve stem packing is practically leakproof, and can be replaced in minutes without leakage or shutdown of the system.

Rating of the V5045A Valve Body is 125 psig maximum; maximum temperature of the controlled medium is 250 F; valve stroke is 1/4 inch to 3/8 inch depending on the model.

Two models are available: straight-through or angle type, both have union outlets.

INSTALLATION

The V2045A Actuator and V5045A Valve Body are packed separately. They may be field assembled before or after installing the valve body to the system piping.

> **CAUTION**
>
> On an assembled device, do not rotate the actuator unless the valve is in the "OPEN" position. Manually turn the control shaft in either direction until the red "OPEN" appears in the indicator window.

The V2045A-V5045A can be mounted in any position from vertical (valve stem up) to horizontal. The shaft of the motor and gear train must be horizontal. Figs. 2A and 2B show the valve in a vertical position. The actuator may be rotated in any direction. Fig. 2C shows a valve mounted horizontally. The indicator window and manual control shaft face outward.

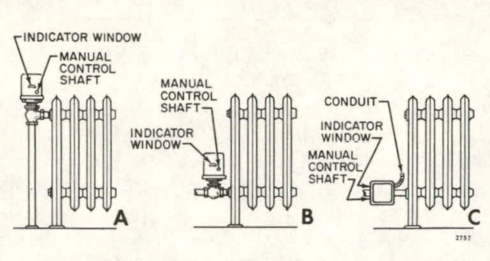
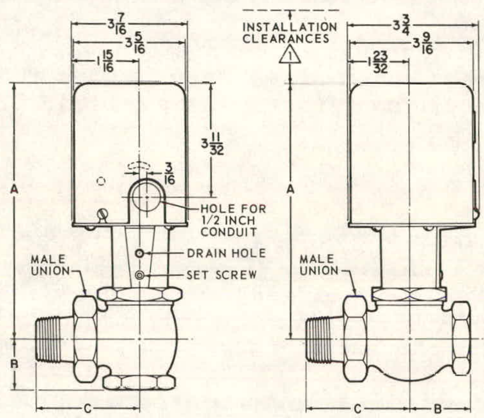


Fig. 1—Proper valve positions.



Valve Size	Angle Type			Straight-Through Type		
	A	B	C	A	B	C
1/2 in.	7- 3/8	1-1/8	2-13/16	7- 3/4	1-3/4	3-1/16
3/4 in.	7- 3/8	1-1/16	3-1/8	7-13/16	2-1/16	3-1/16
1 in.	7- 1/2	1-1/2	3- 1/8	7-15/16	1-13/16	3-1/8
1-1/4 in.	7-11/16	1-3/4	3- 9/16	8- 5/16	2-3/16	3-3/8
1-1/2 in.	7- 7/8	2	3- 3/4	8- 5/8	2-1/4	3-15/16
2 in.	8- 1/8	2-1/4	3- 5/16	9- 1/8	2-9/16	3-15/16

(1) ADD TO DIM A: 4 IN. FOR REMOVAL OF COVER, 2 IN. IF VALVE BODY INSTALLED FIRST, THEN ACTUATOR.

2756A

Fig. 2—Dimensions.

Be sure to allow enough clearance for the conduit and at least 4 inches for removal and service of the actuator.

INSTALL VALVE BODY

> **CAUTION**
>
> Install with fluid flow in direction of arrow cast on valve body to ensure proper valve operation and avoid damage to the system.

If space is limited at the valve location, the valve bonnet and disc assembly may be removed. Turn the bonnet counterclockwise to unscrew it from the body. Carefully lift the bonnet and disc assembly out of the body.

PREPARE PIPE

1. Use properly reamed pipe free from chips.
2. Do not thread or tighten pipe too far (see table). Valve distortion or malfunction may result from excess pipe within the valve.

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Figure 3-30. Two-Way Valve Body

HONEYWELL **TRADELINE** CONTROLS

V2045A VALVE ACTUATOR FOR RADIATOR VALVES

APPLICATION

The V2045A Valve Actuator is used with a V5045A Valve Body combination and a 3-wire spdt thermostat such as the Honeywell T26A to provide two position control of hot or chilled water, low-pressure steam, or air conditioning coils. The V2045A-V5045A combination is ideally suited for zoned systems. The valve actuator may be manually opened during power-off periods, and the controller will automatically resume command when the power is restored.

An auxiliary switch assembly, part no. 114191A (standard) or 114191B (Tradeline), can be field-added to provide control of an additional valve from a single thermostat.

The V2045A can be used to replace the valve-actuator on a V2045A-V5045A assembly and the V2008A Powerhead on a V2008A-V5008A assembly. With an adapter kit, part no. 126149B, the V2045A can also be used with the V5011A or V5011C.

INSTALLATION

CAUTION

1. Installer must be a trained, experienced serviceman.
2. Disconnect power supply before connecting wiring to prevent electrical shock and equipment damage.
3. The valve actuator must be mounted with shaft of motor and gear train horizontal.
4. On an assembled device, do not rotate the actuator unless the valve is in the open position. Manually turn the control shaft in either direction until the red OPEN appears in the indicator window.
5. Do not manually operate the valve to a partially open position in steam systems.
6. Always conduct a thorough checkout when installation is complete.

NOTE: If the control shaft cannot be turned, remove cover and lightly tap the motor lamination (see NOTE ⚠1 Fig. 1); this will free the rotor.

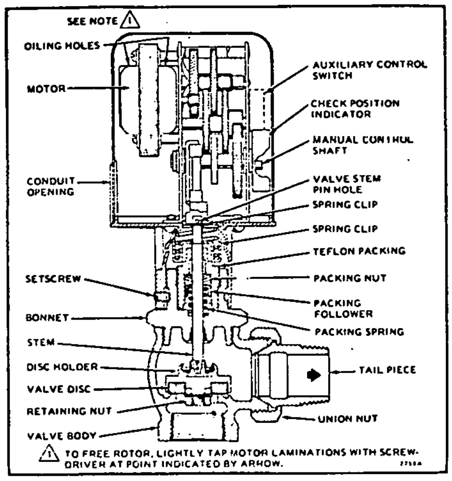


Fig. 1—Cross-section of V2045A-V5045A.

The V2045A-V5045A can be mounted in any position from vertical (valve stem up) to horizontal. The shaft of the motor and gear train must be horizontal. Figs. 2A and 2B show the valve in a vertical position. The actuator may be rotated in any direction. Fig. 2C shows a valve mounted horizontally. The indicator window and manual control shaft face outward.

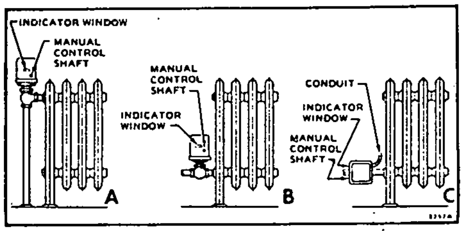


Fig. 2—Proper valve positions.

V2045A VALVE ACTUATOR

After mounting the V5045A Valve Body in the pipeline, push valve stem all the way down, then mount the valve actuator as follows:

CAUTION
The valve actuator must be mounted with shafts of motor and gear train horizontal.

1. Determine a position where (a) the indicator window is visible, (b) the opening for the manual control shaft is accessible, and (c) there is room for flexible conduit connections.
2. Check indicator window for red OPEN. If word OPEN does not show, turn manual control shaft (Fig. 1) with the wrench supplied until it does appear.

Figure 3-31. Two-Position Valve Actuator

HONEYWELL **TRADELINE** CONTROLS

V5013A THREE-WAY MIXING VALVE

APPLICATION

The V5013A Three-Way Mixing valve is used to control hot and cold water.

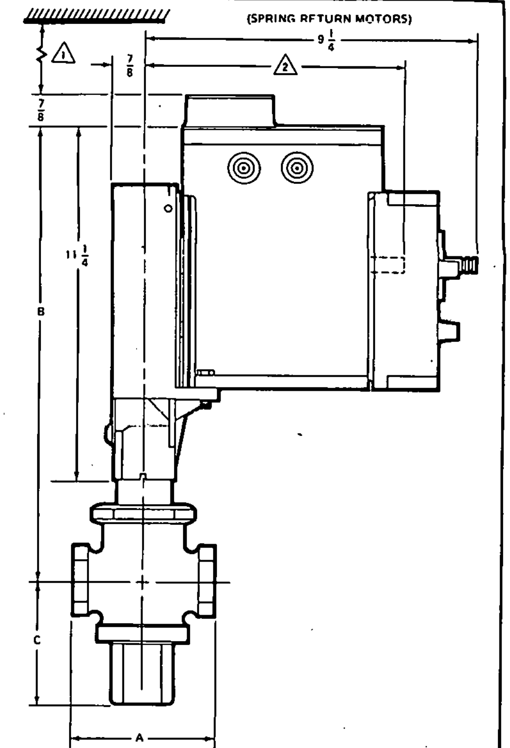
The V5013A requires either a Q601 or Q618 Linkage and a Modutrol Motor to form a motorized valve assembly.

The Tradeline valve will replace existing V5013A Valves with the same flow coefficient, Cv.

GENERAL SPECIFICATION THREE-WAY MIXING VALVE BODY [a]

VALVE SIZE (INCHES)	CAPACITY INDEX (C_v)	MAXIMUM PRESSURE DIFFERENTIAL FOR NORMAL LIFE OF DISC AND SEAT	PACKING LIMITATIONS: TEMP. LIMITS OF CONTROLLED LIQUID	PACKING LIMITATIONS: PRESS. LIMITS OF CONTROLLED LIQUID (GAUGE)	MAX. TEMP. DIFFERENTIAL IN ALTERNATE HOT-COLD USE
1/2	2.5 or 4.0	20 psi	Max. 240 F Min. 40 F	150 psi	140 F
3/4	6.3				
1	10.0				
1-1/4	16.0				
1-1/2	25.0				
2	40.0				

[a]Mixing valves are designed for mixing applications, and are not suitable for diverting applications.



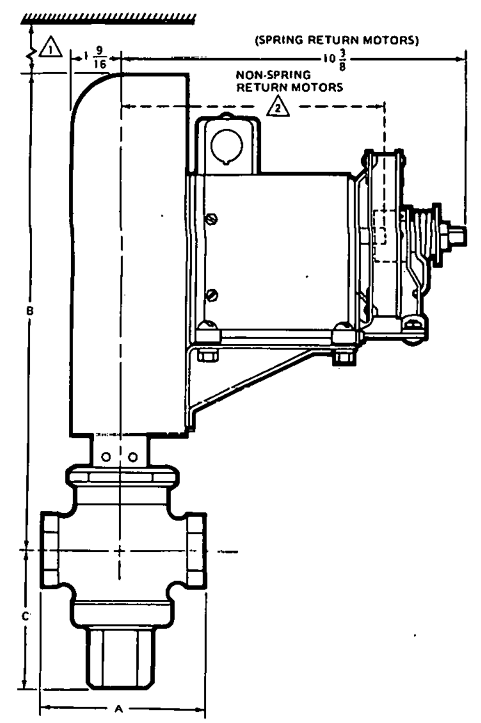
1 ALLOW APPROXIMATELY 4 INCH CLEARANCE FOR REMOVAL OF LINKAGE.

2 M634A, M934A, M7034 8 INCHES. OTHER NON SPRING RETURN MOTORS 7-3/4 INCH.

VALVE SIZE	DIMENSIONS IN INCHES: A	B	C
1/2	3-1/8	14-3/4	2-3/4
3/4	3-3/8	14-5/8	2-5/8
1	3-7/8	15	2-7/8
1-1/4	4-1/4	15-1/8	2-3/4
1-1/2	4-3/4	15-1/2	2-7/8
2	5-7/8	16-1/8	3-1/4

4411

Fig. 1—Mounting dimensions V5013A with Q618 Linkage.



1 ALLOW APPROX. 4 IN. CLEARANCE FOR REMOVAL OF LINKAGE.

2 M634A, M934A, M7034A—8-3/8 IN. OTHER NON-SPRING MOTORS 8-1/8 IN.

VALVE SIZE	DIMENSIONS (IN INCHES): A	B	C
1/2	3-1/8	17-3/4	2-3/4
3/4	3-3/8	17-5/8	2-5/8
1	3-7/8	18	2-7/8
1-1/4	4-1/4	18-1/8	2-3/4
1-1/2	4-3/4	18-1/2	2-7/8
2	5-7/8	19-1/8	3-1/4

4412

Fig. 2—Mounting dimensions V5013A with Q601 Linkage.

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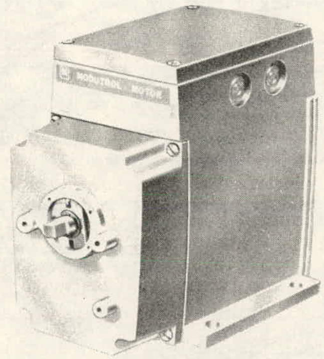
Figure 3-32. Three-Way Valve

MODUTROL MOTORS



M445A,B,C,D
M845A,B,C,E

The M445 and M845 are two position, spring return Modutrol motors. They are used to operate dampers or valves in applications where it is necessary or desirable to have the controlled element return to the starting position in the event of power failure or interruption.

The M445 operates on line voltage; the M845 operates on 24v ac. All models have a one minute, 160 degree stroke.

Models of these motors have an internal, adjustable spdt switch to actuate auxiliary equipment at a particular point in the motor stroke.

The M445C and M845B are equipped with internal, thermostatically controlled heaters for use in cold weather applications.

The M445B and M845E are designed for normally open valves; all other motors are normally closed.

☐ Sturdy, lightweight, die-cast aluminum case.

☐ Integral spring returns motor to normal position when power fails or is interrupted.

☐ Built-in spdt adjustable switch is available on some models for the control of auxiliary equipment.

☐ Oil immersed gear train assures long life and quiet operation.

☐ Thermostatically controlled internal heater on some models permits use at temperatures as low as minus 40 F.

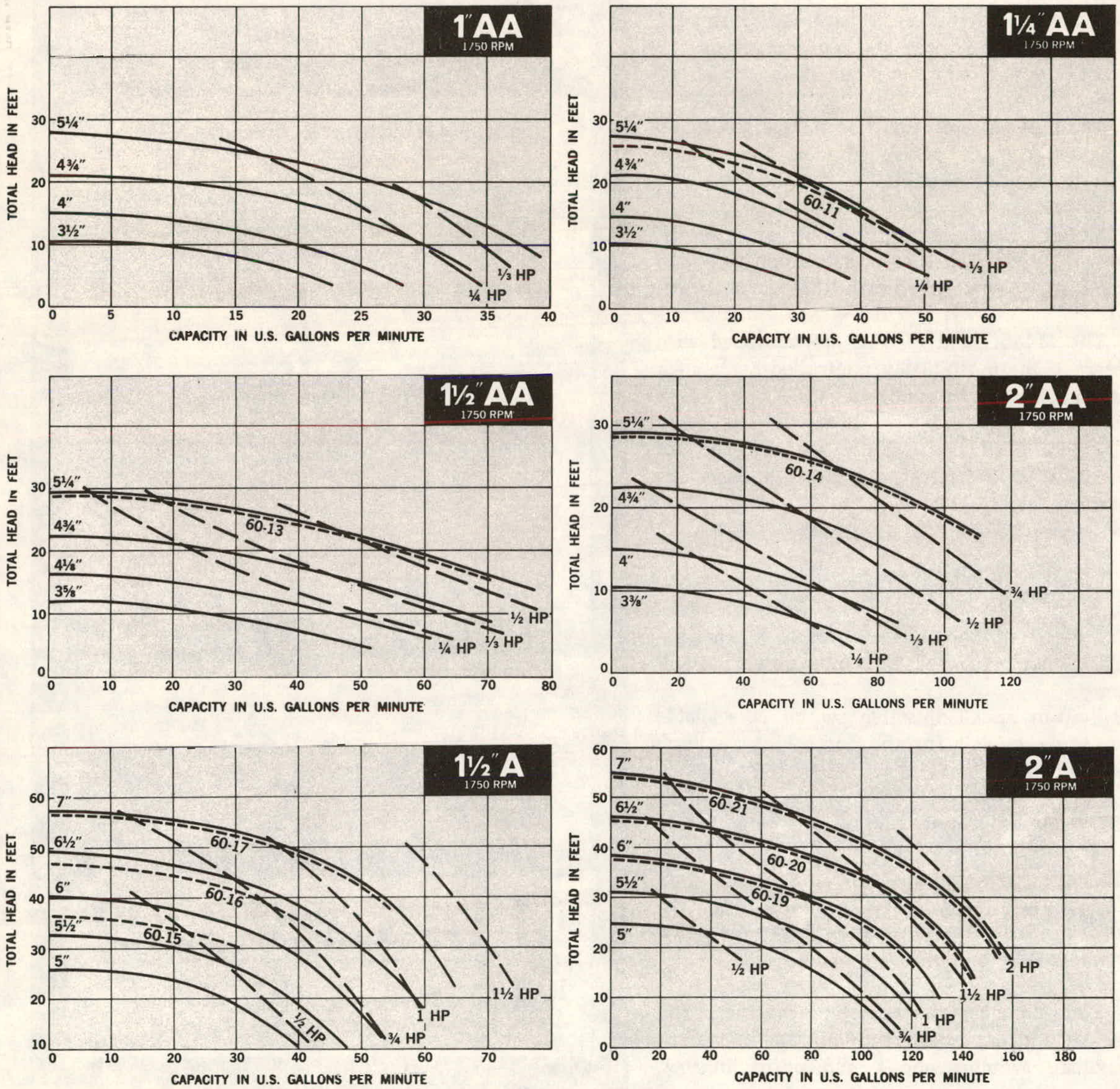
☐ Full line of accessories includes weatherproofing kit and explosion proof housing as well as auxiliary switches and a number of linkages.

☐ Tradeline M845A includes multitap transformer for 120/208/240v ac control circuits.

60-2037 11/71

Figure 3-33. Spring Return Valve Actuator

Series 60 Pumps can be furnished in bronze-fitted, all iron, or all bronze construction to suit your application



Curves based upon shop test using clear cold water at a temperature of not over 85° F.
Horsepower curves do not include motor service factor.

Figure 3-34. Pump Performance Curve

Control Panel

The control cabinet houses the power supplies and all of the electrical equipment required to operate the system in all of its various modes. The control panel located at the top of the cabinet has drawn on it a simplified flow diagram of the system. Push button switches are located on the diagram in the proper location for each of the motorized valves, pump, and blower. Lights in the switches signify when a valve is open in a line or a motor turned on. The printed circuit logic card is also accessible from the panel.

To operate the panel, the power switch must be turned on and the automatic or manual mode selected. In the manual mode, pressing and illuminating the desired switches will change the flow to any of the system circuits. The collector flow knob controls the amount of fluid flow through FC1 to the solar collector, and the indicator lights show the amount of flow.

On automatic operation the manual mode is completely disabled, and valve positions are controlled by the output of the logic circuit. The flow control valve is now controlled by the temperature of the fluid coming from the solar panel and the outdoor temperature. For lower outdoor temperatures, it is desirable to have hotter fluid for heating. Therefore, the valve partially closes, restricting the flow and thereby raising the temperature. This establishes the set point temperature which the fluid temperature sensor modulates the valve to hold. In addition, switches on the valve controller sense the amount the valve is open and select the proper outlet valve from the stratified storage tanks. When solar energy is readily available and the flow rate is high, V1 is open to take fluid from the cooler bottom of the tank. When the flow control valve closes part way, V1 is closed and V2 opens. Similarly, it closes and V3 opens for low flow to help maintain the desired higher temperature.

The fluid is returned to either the top or bottom of the storage tanks by V6, depending on fluid temperature. If the fluid is hotter than the top of the tank, it enters at the top; if the fluid is colder than the top of the tank, it enters at the bottom. The storage return control adjusts the deadband, or differential temperature, that the fluid must have before it will change the valve position. An exception to this is when no heat is being collected, the boiler is supplying auxiliary heat, and the return fluid is hotter than the tank. It would be a waste of energy to use the boiler to heat the storage tanks, so V4 diverts the fluid bypassing storage.

When no heat is being collected, signified by FC1 being closed, V5 is opened to allow heating from storage. The first stage of the thermostat controls the position of V7 to direct flow to the coil if required or return it to storage, and turns on the blower. It also turns off the pump if no heat is being collected or demanded. If the space temperature cannot be held, the second stage of the thermostat comes on which starts the boiler and opens V10 to also provide heat through the auxiliary coil.

INSTRUMENTATION

Temperature

Temperature measurements are made at approximately 100 different locations throughout the system. The temperature of the system fluid is measured at every component inlet and outlet from the time it leaves the solar collector until it returns. Temperature is also measured on the surface of a number of solar panels and at the outlet of each of the eight solar collector sections. Cooling tower water temperature is also measured in the distribution ducts before and after each heat exchanger plus inlet fresh air and return air from the vans. Wet bulb temperature are measured in the return air duct, before the Rankine evaporator coil and after the collector heat exchanger coil. Ambient air is also measured in the equipment van.

Outside wall temperatures of both the equipment and display vans are measured. All these temperature readings are recorded for analysis and performance evaluation of all the heating and cooling components.

Copper-Constantan thermocouples were used throughout the entire system. In the water or system fluid plumbing, Omega 1/8 inch well-type thermocouples were immersed in the liquid for good heat transfer and fast response readings. These were Omega model CPSS - 18G - 12 assemblies as shown in F gure 3-35. All the other thermocouples are made from Teflon-covered CU/C wire with lead junctions welded at the end. This same wire is used for lead wire to CU/C junction blocks in the auxiliary console where all the thermocouples are terminated.

Copper-Constantan leads are run from 40 selected thermocouples to two thermocouples selector switches (Figure 3-36). These switches are connected to an Omega digital thermocouple readout (Figure 3-37) where they can be read one at a time. Several selected thermocouples are also connected to the Data Logger for reading and analysis.

Water Flow (System Fluid)

Fluid flow is measured in three locations -- total flow, collector flow, and auxiliary flow. Two different types of flowmeters are at each location to provide visual and automatic readout. The visual flowmeter is a Brooks Rotometer which is a float in a ribbed tapered tube. The total flowmeter is calibrated in gallons per minute while the collector and auxiliary flowmeters are scaled in percent which is transferable to gallons per minute from a calibration curve.

The second type of flowmeters has an electrical output proportional to flow in gallons per minute. The basic flowmeter is a one-inch Brooks Disc type cast iron body, type 666, flowmeter as shown in Figure 3-38. A rotating disc produces a rotary shaft motion as an output. Attached to the top of the meter body and connected to the rotary shaft output is a high-frequency pulse generator (Figure 3-39) which generates a pulse proportional to the shaft revolutions. The output of the pulse generator is fed to a Brooks Pulse to analog converter (Figure 3-40) which converts the pulses to a voltage. This voltage is read out visually on a voltmeter in the auxiliary console and recorded on the Data Logger for analysis.

Air Flow

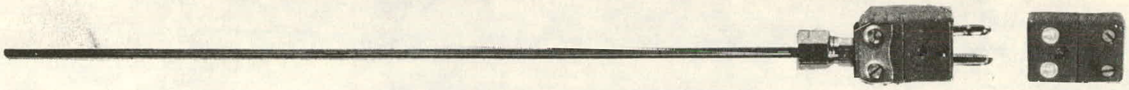
Air flow measurements are made in the blower cabinet and the fresh air intake duct. Several locations are provided at the outlet of the collector coil heat exchanger in order to plot a profile of the air velocity in the duct. Thermo-System Model 1610 velocity transducers (Figure 3-41) are used to measure this air velocity. These transducers are a hot wire anemometer type with an amplifier on the end of the probe. The output voltage (proportional to velocity in feet per second) is read out on a voltmeter plus being fed to the Data Logger for recording and analysis.

Weather Station Instrumentation

Weather instrumentation includes the following sensors manufactured by Weather Measure Corporation, Sacramento, California, and Kipp & Zonen, Holland.

- W101-P-HF/540 Skyvane I Wind Sensor with WTB101-HF/540 Translator -- Measures wind speed with a high-frequency tachometer. Tracking starts at wind speeds slightly less than one mile per hour.
- B242 Analog Output Barometer -- Measures barometric pressure over any 100 mb interval between 600 and 1065 mb (17.7 inches and 31.45 inches Hg). The gain and zero point can be set in the field to correspond to local elevations.

COMPLETE THERMOCOUPLE ASSEMBLIES



WITH QUICK DISCONNECT PLUG AND JACK

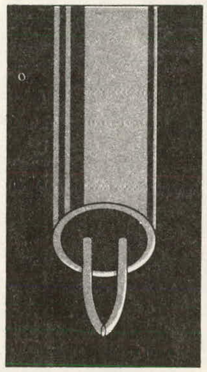
OMEGA'S QUICK DISCONNECT TYPE thermocouple assemblies come complete with color-coded, quick coupling male and female connectors having polarized contacts. Stock probes are nominally twelve inches long. They are available in 1/16, 1/8, 3/16 and 1/4 inch slzes.

SPECIFICATIONS *

DIAMETERS ■ Standard diameters: 1/16", 1/8", 3/16" and 1/4" with two wires.
LENGTH ■ Standard OMEGA thermocouples have 12 inch immersion lengths. Other lengths available.

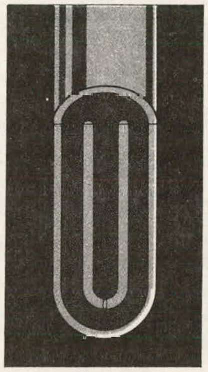
SHEATHS ■ Type 304 Stainless Steel and Inconel* are standard. Other sheath materials available; write for price and availability.
INSULATION ■ Magnesium Oxide is standard. Minimum insulation resistance wire to wire, or wire to sheath is 1.5 megohms at 500 volts D.C. in all diameters.
CALIBRATIONS ■ Iron-Constantan (J), Chromel-Alumel (K), Copper-Constantan (T) and Chromel-Constantan (E) are standard calibrations.
ACCURACY ■ The wires used in OMEGA thermocouples are selected and matched to meet ANSI Limits of Error.
BENDING ■ Easily bent and formed. Bend radius should be no less than twice the diameter of the sheath.
DELIVERY ■ Other sheaths available; write for price and delivery.

MEASURING JUNCTIONS



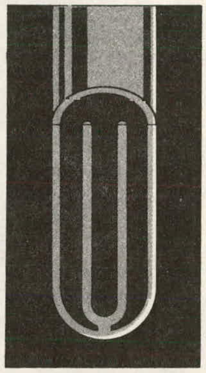
EXPOSED JUNCTION

This junction is recommended for the measurement of static or flowing non-corrosive gas temperatures where response time must be minimal. The junction extends beyond the protective metallic sheath to give fast response. The sheath insulation is sealed at the point of entry to prevent penetration of moisture or gas.



UNGROUNDED JUNCTION

This type is recommended for the measurement of static or flowing corrosive gas and liquid temperatures in critical electric applications. The welded wire thermocouple is physically insulated from the thermocouple sheath by a hard, high purity ceramic.



GROUNDED JUNCTION

This junction is recommended for the measurement of static or flowing corrosive gas and liquid temperatures and for high pressure applications. The junction of this thermocouple is welded to the protective sheath giving faster response than the ungrounded junction type.

RECOMMENDED THERMOCOUPLES

CALIBRATION AND USE	POSITIVE (+)	NEGATIVE (−)	SYMBOL	COLOR CODE
COPPER-CONSTANTAN (VERY LOW TEMPERATURE)	COPPER (YELLOW METAL)	CONSTANTAN (SILVER METAL)	T	BLUE
IRON-CONSTANTAN (REDUCING ATMOSPHERE)	IRON (MAGNETIC)	CONSTANTAN (NON-MAGNETIC)	J	WHITE
CHROMEL-ALUMEL (OXIDIZING ATMOSPHERE)	CHROMEL (NON-MAGNETIC)	ALUMEL (MAGNETIC)	K	YELLOW
CHROMEL-CONSTANTAN (HIGH OUTPUT)	CHROMEL (NON-MAGNETIC)	CONSTANTAN (NON-MAGNETIC)	E	BROWN

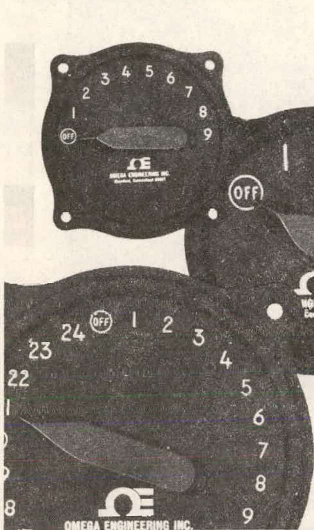
TWO PROTECTION TUBE MATERIALS

MATERIAL	RECOMMENDED MAXIMUM TEMP. °F	APPLICATION ATMOSPHERE				APPLICATIONS
		OXIDIZING	HYDROGEN	VACUUM	INERT	
304SS	1700	VERY GOOD	GOOD	VERY GOOD	VERY GOOD	Recommended for general chemical applications, food applications, oil refinery use, and steam lines.
INCONEL 600	2100	VERY GOOD	GOOD	VERY GOOD	VERY GOOD	Recommended for gas furnaces, lead baths and bath mixtures containing cyanide. Do not use in salt baths contaminated by sulphur.

* SEE TABLE ON PAGE D-12 FOR PROBE RESISTANCE.

Figure 3-35. Thermocouple Assemblies

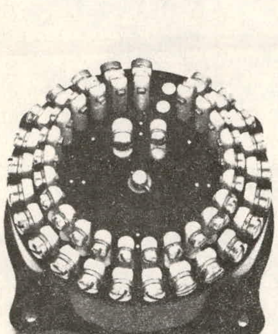
[redacted]



Number of Dual Circuits in Addition to "Off"	Size	Omega Catalog Designation	Price of Switch
2	2″	YSTR2-2	$ 50.00
3	2″	YSTR2-3	52.50
4	2″	YSTR2-4	55.00
5	3″	YSTR3-5	57.50
6	3″	YSTR3-6	60.00
8	3″	YSTR3-8	62.50
9	3″	YSTR3-9	65.00
10	3″	YSTR3-10	67.50
12	3″	YSTR3-12	70.00
16	3″	YSTR3-16	85.00
18	3″	YSTR3-18	95.00
20	3″	YSTR3-20	100.00
24	4″	YSTR4-24	105.00
28	4″	YSTR4-28	110.00
40*	4″	YLTR4-40	112.50

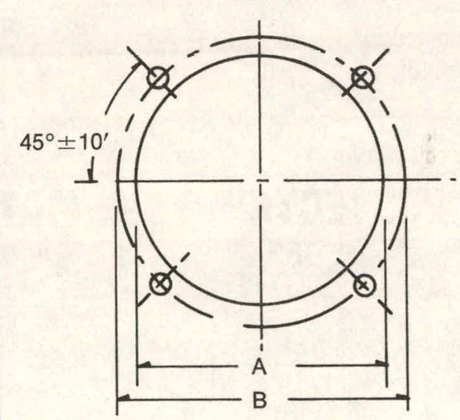
*This switch has soldering-lug terminals.

[redacted]



Rear View 24-Point Switch

NO. 18 (.1695) DRILL 4 HOLES EQUALLY SPACED

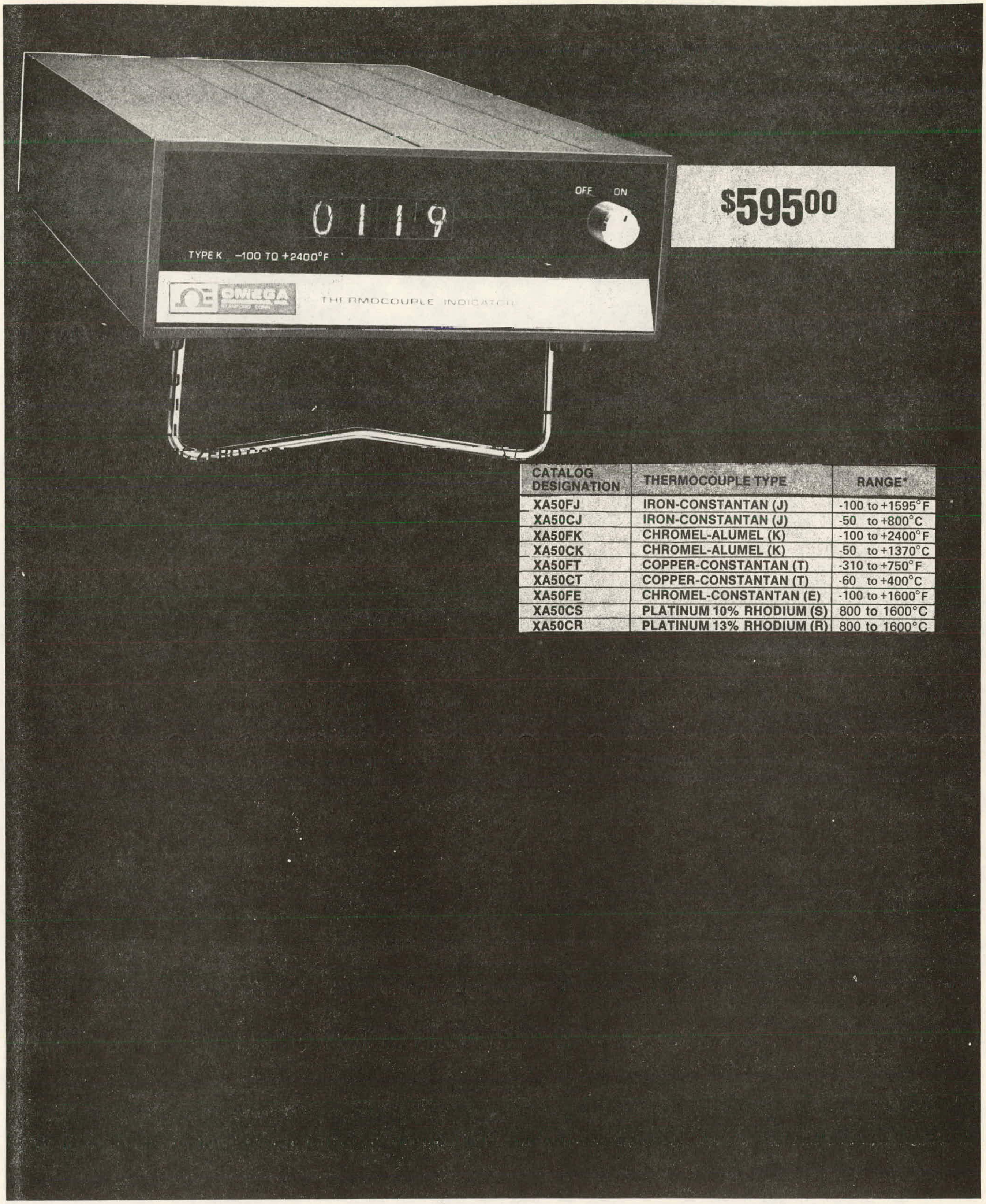


DRILLING DIMENSIONS

For Flush Mounting

Nominal Case Size	Dim. A.	Dim. B.
2″	2.255	2.625
3″	3.130	3.500
4″	4.130	4.750

Figure 3-36. Thermocouple Switch Assembly



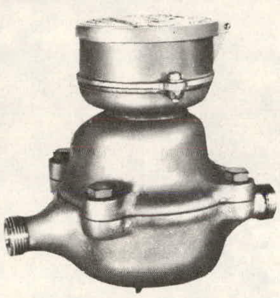
CATALOG DESIGNATION	THERMOCOUPLE TYPE	RANGE*
XA50FJ	IRON-CONSTANTAN (J)	-100 to +1595°F
XA50CJ	IRON-CONSTANTAN (J)	-50 to +800°C
XA50FK	CHROMEL-ALUMEL (K)	-100 to +2400°F
XA50CK	CHROMEL-ALUMEL (K)	-50 to +1370°C
XA50FT	COPPER-CONSTANTAN (T)	-310 to +750°F
XA50CT	COPPER-CONSTANTAN (T)	-60 to +400°C
XA50FE	CHROMEL-CONSTANTAN (E)	-100 to +1600°F
XA50CS	PLATINUM 10% RHODIUM (S)	800 to 1600°C
XA50CR	PLATINUM 13% RHODIUM (R)	800 to 1600°C

Figure 3-37. Digital Thermocouple Readout

BROOKS-DISC

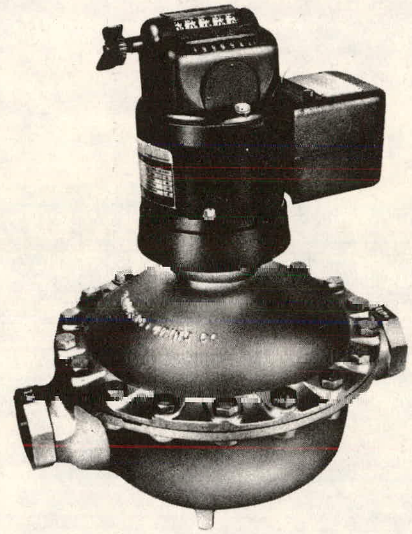
ALL BRONZE

CHEMICALS, OILS, WATER
FLOW RATES: 1 TO 160 gpm



1" METER WITH
SERIES "10" REGISTER

1-1/2" METER
WITH BRODIMATIC COUNTER



2" METER WITH IMPULSE CONTACTOR
AND SERIES "30" REGISTER

Designed especially for quality minded meter buyers, the Brooks-Disc (666) Industrial meter costs less initially. Used primarily for measuring and batching chemicals, oils and water, the Industrial Disc Meters are engineered to run longer and measure more. They give additional savings because their practical accuracy is sufficient to meet everyday industrial requirements.

Exact dimensional control during machining assures pricise interchangeablility of component parts, high initial accuracy and lower maintenance cost.

The Brooks-Disc (666) Bronze meter accepts a variety of Brooks Counters; among which are Series 10 Counters with optional integral impulse contactor, Series 30 Counters, Brodimatic Counters and Brodimatic Quantrol Counters with micro switch. An explosion proof Impulse Contactor can be utilized as a separate stack-up accessory for electrical control of counters, samplers, injection devices or other electrically-operated equipment. Page 4 gives a brief description of these accessories.

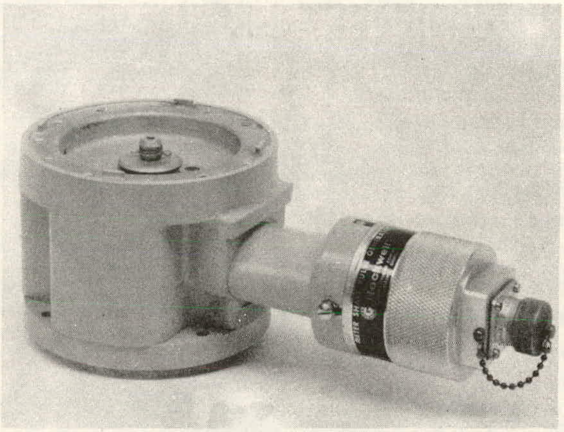
Figure 3-38. Brooks Disc-Type Flowmeter

DESIGN SPECIFICATIONS

BROOKS INSTRUMENT DIVISION
EMERSON ELECTRIC CO.
STATESBORO, GEORGIA 30458

EMERSON

HIGH FREQUENCY PULSE GENERATOR



FEATURES

* Converts small increments of throughput into pulses
* Adapts to most meters
* Self contained, sealed transistorized circuitry
* Explosion-proof and weather tight
* No calibration adjustment required
* Maintenance-free
* Temperature: Minus 30 to 200 F.
* Long distance operation
* Can serve several meters
* Low output impedance
* Low torque and low speed

The Brooks High Frequency Pulse Generator is a device for obtaining a high resolution, digital signal for meter throughput. It is designed to fit new and existing installation using Brooks meters and with the proper adapter those manufactured by others.

A companion to the Pulse Generator is an adapter which mounts permanently in the meter stackup. Depending upon the nature of the installation, the Pulse Generator can be permanently attached or merely plugged into the adapter.

Although designed primarily as an accessory for meter proving, the Pulse Generator has an application wherever precise throughput or rate-of-flow measurement is required. Providing as many as 10,000 pulses per throughput unit, the Generator and adapter supply digital throughput resolution necessary for electronic counters used with high speed, piston provers.

Instantaneous flow rates can also be accurately determined with oscilloscopes or time base electronic counters or displayed as a continuous indication on a deflection meter by converting the pulse rate to proportional voltage or current signals. It provides the pulse output required for control of digital blending systems.

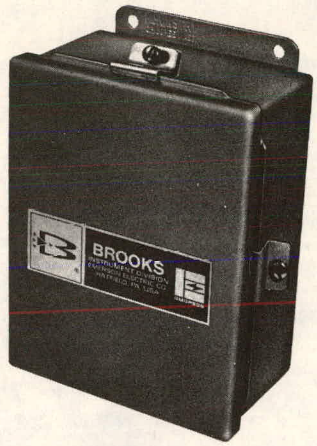
A disc with precisely spaced opaque and transparent slots is rotated between an incandescent lamp and photo-voltaic cell. The rapid interruption of the light source causes the photocell to emit pulses into a self contained transistorized circuit for shaping and amplifying. The resulting pulse from the Pulse Generator is a positive square wave.

Figure 3-39. Brooks High-Frequency Pulse Generator

BROOKS PULSE-TO-ANALOG CONVERTERS
Models 4010 and 4011

Description

The Brooks Models 4010 and 4011 Frequency-to-Analog Converters will accept a contact closure input or a logic input signal and produce an analog output. The Model 4010 provides a voltage output and Model 4011 provides a current output. Both units can be supplied with one, two, or three control relays, each with SPST and SPDT contacts which may be used to operate an alarm, valve, etc. Each control relay has its own comparator circuit which contains a set point adjustable from 0 to 100% (Full Span), and a dead band adjustable to approximately 5% (Full Span), which eliminates relay chatter or oscillation. The converter utilizes integral power supplies, and is completely self-contained in a metal enclosure suitable for wall mounting.



Model 4010, 4011
Frequency-to-Analog Converter

Design Features

- Accepts contact closure or logic input signals
- Wide range of input frequency capability
- Outputs corresponding to standard process instrument inputs
- Printed circuits, solid state components
- Modular design for more efficient maintenance
- Integral power supplies
- Optional integral control relays available

Specifications

POWER REQUIREMENTS
115 vac ± 10 vac 50/60 Hz (Standard)
220 vac ± 20 vac 50/60 Hz (Optional)

INPUT SIGNAL TYPE
Contact Closure
Logic: 0-5 volt pulse (min 25 % of duty cycle)
Photopulser: 0-10 volt, 3-10 volt
Signal Generator: sine or square wave (2 volt RMS min.)

INPUT SIGNAL FREQUENCY RANGE

*Minimum	**Max. (FS) (Adjustable)
3 Hz	8 - 60 Hz
3 Hz	16 - 120 Hz
3 Hz	70 - 580 Hz
3 Hz	330 - 2500 Hz

* 3 Hz min. when quick response is required, min. lower frequency may be achieved resulting in slower response time.

** These ranges adjustable for 100% (FS) at factory or by user.

OUTPUT RANGES $225
Model 4010: 0-100 mv, 0-5 v, 0-10 v
Model 4011: 1-5 ma, 4-20 ma, 10-50 ma, 0-5 ma, 0-20 ma, 0-50 ma

NOTE: Above output ranges are standard. Other ranges can be supplied upon request. Consult factory.

OUTPUT CONTROL (OPTIONAL)
One, two, or three control relays providing SPST and SPDT contacts

CONTROL RELAY CONTACT RATING
5 amperes at 115 vac

OUTPUT LOAD RESISTANCE
0-5 ma, 1-5 ma: 0 to 3400 ohms
0-20 ma, 4-20 ma: 0 to 850 ohms
0-50 ma, 10-50 ma: 0 to 330 ohms

LINEARITY
Better than ± 0.1% (FS)

REPEATABILITY
Better than ± 0.25% (FS)

ACCURACY
Zero ± 0.5% (FS); Span ± 0.5% (FS)

RESPONSE TIME
Less than 5 seconds for any input change in range from 8 to 15 Hz and less than 3 sec. for any input change above 25 Hz.

OPERATING TEMP. RANGE
0° F to 150° F

STORAGE TEMPERATURE RANGE
−30° F to 180° F

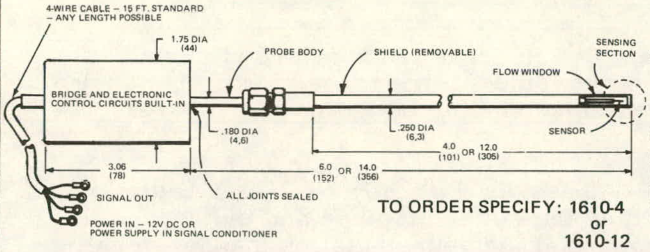
PHYSICAL DIMENSIONS
Models without control relays:
7" x 9½" x 3¾" (Overall)
Models with control relays:
9" x 11½" x 4¼" (Overall)

INTERCONNECTING CABLES (User Furnished)
Input: 2 conductor shielded, Belden 8422 or equivalent
Output Signal: No. 22 AWG
Control Relay: No. 18 AWG

Figure 3-40. Brooks Pulse-to-Analog Converter

MODEL 1610 VELOCITY TRANSDUCER

PROBE PACKAGE FOR 12V DC POWER SOURCE—NON-LINEAR—NO READOUT



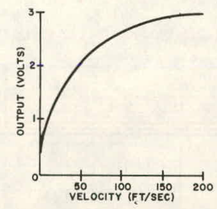
TO ORDER SPECIFY: 1610-4
or
1610-12
Depending on Length

SPECIFICATIONS

Velocity Range: 0 to 200 ft/sec (60 mps). (Data is furnished to give good accuracy throughout the above range. Any specific smaller range can be utilized. Calibration is at standard conditions of temperature and pressure.)
Output: Approx. 0.5 to 3V DC. Impedance – 100 ohms
Accuracy: ±0.1% of FS and ±2% of Reading
Repeatability: less than 0.1% (repeated conditions)
Response Time: 100 milliseconds
Environmental Operation Conditions-Sensor Section: Temperature–20°C to 100°C Pressure 0.01 to 500 psia
Electronic Section: 0-70°C
Cable: 15' Standard (specify otherwise)
Power Supply: 12V $^{+3.0}_{-0.2}$ V DC (+15V DC optional). 0.5 Amp. Max.

PRINCIPLE OF OPERATION

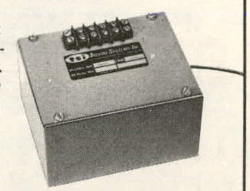
The 1610 measures velocity (actually "standard" veocity based on calibration at 70°F and 1 atmosphere or mass flow) by sensing the cooling effect of a moving flowstream over the heated sensor surface. The sensor is heated electrically by current from the control amplifier. The sensor is one leg of a bridge which is continuously balanced by the unique, PDM amplifier system, requiring no adjustments. Another leg of the bridge serves as a temperature sensor to compensate for temperature changes. The output signal is proportional to the electrical power dissipated in the sensor or about the square root of velocity. If desired a linear output is provided by a very accurate curve-fitting amplifier that is part of the Signal C nditioner packages shown below.



TYPICAL CALIBRATION CURVE-VT SERIES

MODEL 1605 POWER SUPPLY

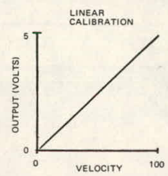
Converts 115/230V DC power to 12V DC for operation of VT Series Probes. Convenient terminals serve as junction point for signal leads.



SIGNAL CONDITIONERS – LINEAR OUTPUTS, READOUT OPTIONS

MODEL 64 LINEAR SIGNAL CONDITIONER-ANALOG READOUT

PROVIDES LINEAR OUTPUT, ANALOG METER, CONVENIENT PORTABLE PACKAGE



SPECIFICATIONS

(Signal conditioner includes power supplies, linearizing circuits, output amplifiers and analog meter.)
Analog Output: Rear Terminals; 50 ohm impedance; 0 – 5V DC full scale range; 100 millisecond response time; and total linear accuracy ±3% of reading in 50 to 1 range.
Analog Meter: 0 – 100% and 0 – 10% of full scale. ±1% F.S.
Cabinet: Aluminum, portable–8.5 x 3.5 x 11". (216 x 89 x 280 mm)
Power Required: 115/230 V AC, 50-400 Hz.

Each Model 64 is set up to match a given Model 1610 Transducer.

Ranges:	Model No.	Range
	64-1	0 – 1000 fpm
	-3	0 – 10 mps
	-5	0 – 100 fps
	-6	0 – 10,000 fpm
	-8	0 – 100 mps

(other ranges may be specified as needed)

MODEL 67 LINEAR SIGNAL CONDITIONER-DIGITAL READOUT

PROVIDES LINEAR OUTPUT, DIGITAL METER, CONVENIENT PORTABLE PACKAGE

SPECIFICATIONS

Digital Display: 3½ digit, neon; 100% over range; 2 second response time, 0.2 second/reading display, and ±1 digit resolution. BCD output optional.
Analog Output: Rear Terminals; 50 ohm impedance; 0 – 5V DC full scale range; 100 millisecond response time; and total linear accuracy ±3% of reading in 50 to 1 range.
Cabinet: Aluminu.n, portable– 8.5 x 3.5 x 11". (216 x 89 x 280 mm)
Power Required: 115/230 V AC, 50-400 Hz.

Each Model 67 is set up to match a given Model 1610 Transducer.

Ranges:	Model No.	Range (direct reading)
	67-2	0 – 2000 fpm
	-4	0 – 20.00 mps
	-7	0 – 200.0 fps
	-8	0 – 100.0 mps

(other ranges may be specified as needed)

Accessories For Signal Conditioners: Model 10166 Rack Mounting Kit – contains hardware for clamping 2 units side by side and mounting in standard 19" rack – 3.5" high.
Model 10165 Multi Channel Kit contains hardware for clamp ing either 2 or 4 units together – with convenient handles.

Figure 3-41. Thermo-System Velocity Transducer

- H352 Solid-State Relative Humidity Sensor -- Measures relative humidity between 20-95 percent at temperature ranges of -20 to 120°F. Unit contains probe and output meter.

- T621 Remote Temperature Indicator -- Records temperature with an accuracy of ±0.3°F. The unit has two temperature ranges, -40°F to 60°F and 30°F to 130°F. These are switch selectable on the panels. Panel contains readout meter.

- Kipp & Zonen CM5 Solarimeter -- Measures the amount of solar radiation. Two are employed -- one mounted on the collector plane and the second mounted in a horizontal position for total incident-radiation measurement.

Table 3-4 outlines pertinent characteristics of these sensors.

Table 3-4. Weather Station Sensor Characteristics

Sensor		Signal Output	Accuracy	Power	Size (in.)	Weight (lb)
I	W101-P-HF/540 & WTB101-HF/540 (wind direction and speed)	Modified 0-10 mv	±1 mph ±5% above 25 mph	115 vac 60 Hz	29 x 31 x 13 11 x 6 x 6.5	12 4.5
II	B242 (barometric pressure)	Modified 0-10 mv	±0.5 mb	115 vac 60 Hz	9 x 5.5 x 5.5 6 x 11 x 6.5	9
III	B352 (relative humidity)	0-10 mv	±4%	115 vac 60 Hz	12 x 6.75 x 6	5
IV	T621 (temperature)	Modified 0-10 mv	±0.3°F	115 vac 60 Hz	12 x 6.5 x 6.75	4
V	Kipp and Zonen CM5 Solarimeter	0-10 mv			12 x 12 x 6	8

Basic criteria used to select weather sensors included price, accuracy, reliability, and compatibility. All instruments surveyed were comparable with respect to price range and accuracy. Few offered output signals that were compatible with the Kaye Data Logger. Weathermeasure was agreeable to modify signal output levels to the required 0-10 mv.

DATA ACQUISITION/PROCESSING SYSTEM

Operation

The Data Acquisition/Processing System has been designed to collect all data necessary to monitor and assess the operational efficiency of the transportable laboratory and provide timely results of resultant data (Figure 3-42). The system consists of sensors and a data recording/terminal system with the following modes of operation:

- Data Collection -- Data gathered from weather sensors, thermocouples, and flow meters are periodically scanned by the Data Logger. The data logger converts the analog voltage reading to digital signals and outputs the data to the Techtran 4100 Communication Terminal (Magnetic Tape Cassette). The periodic scan interval is determined by two factors. First, a switch on the front panel is used to set the scan interval (nominally 30 minutes). Secondly, 10 valve switch positions which determine

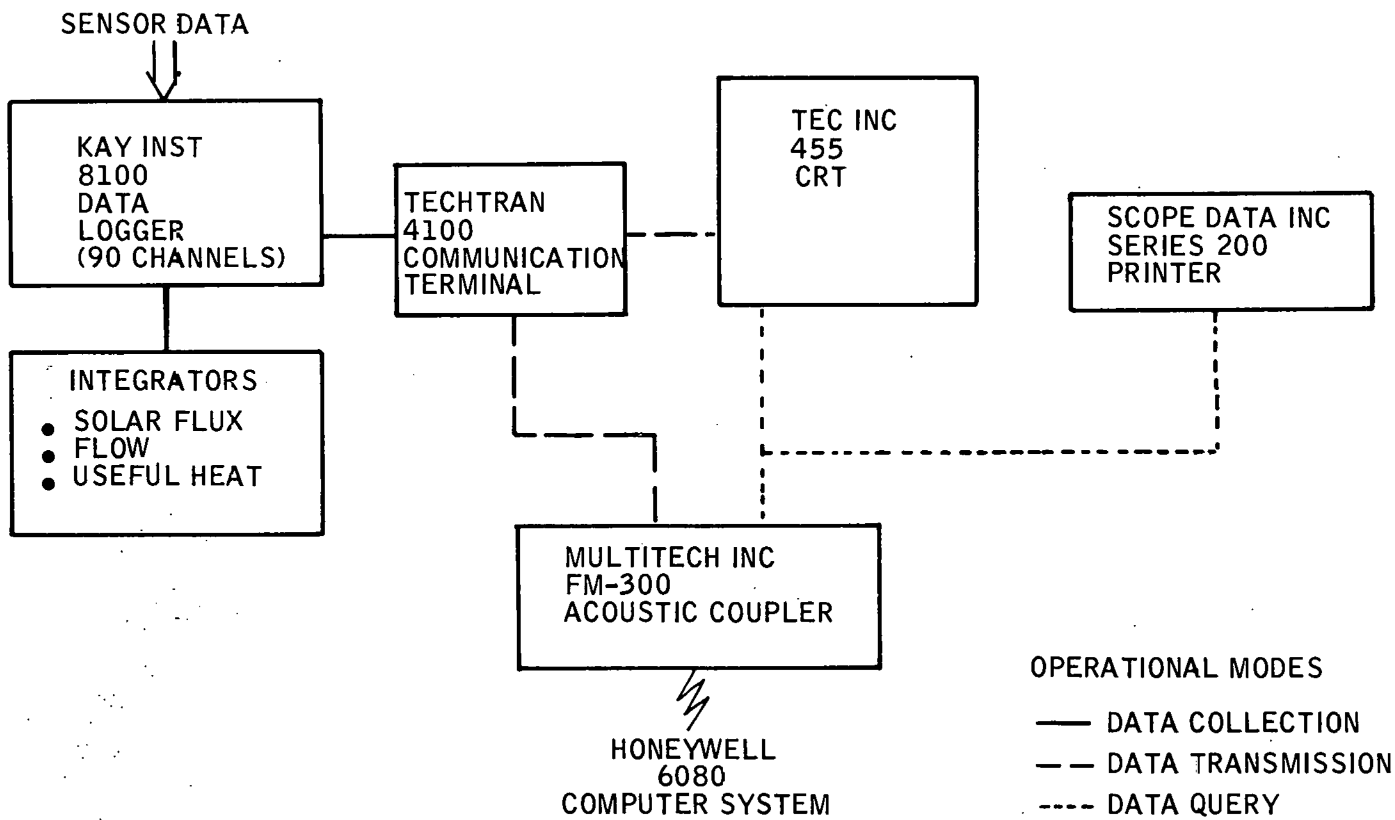


Figure 3-42. Data Acquisition System

the operational mode of the system are monitored. When any one of these changes state (on to off, off to on, open to close, or close to open), the data logger is tripped to initiate a scan. This information is required for computation of heat gain/loss for the various modes of operation.

- Data Transmission -- In this mode of operation, data are transmitted over the phone lines to the Honeywell H-6080 Computer System in Minneapolis, Minnesota. In this mode, contact is made with the computer via the keyboard on the TEC 455 CRT Terminal. Data are transmitted from the Tectran 4100 Communication terminal at a rate of 30 characters per second. The terminal system operates in a time share mode of operation.

- Data Query -- This mode of operation allows the terminal operator to query the data base established in the H-6080 computer system. Data available for query consists of the data collected for the previous 7 days and some calculated daily data such as heat collected and heat available. Request for data are made via the CRT keyboard. The results of the data query are plotted as a function of time on the CRT screen. Alternately, a hardcopy of the query results can be obtained via the printer.

Equipment Description

A brief description of the particular equipment that comprises the data recording terminal subsystem includes:

- Kaye Instruments Inc. 8000 Data Logger -- The data logger accepts low-voltage signals from the instrumentation and weather subsystems (i.e., wind speed, thermocouples, flowmeters, etc.), multiplexes these signals, converts them to digital form, and outputs the digital signals to the magnetic tape recorder.

  - RS 232 Interface to Cassette
  - Day Printout
  - External Step and Print Control
  - 90 Channels

    60 thermocouple -- Type T

    30 low-level voltage, 0 to 20 mv

- Techtran 4100 Communications Terminal -- The Techtran 4100 is a magnetic tape cassette digital recorder used to record data from the data logger. This unit also interfaces with the acoustic coupler to permit recorded data to be transmitted over the phone lines.

  - Transmission speed selectable (110-1200 band)
  - RS 232 interface to modem
  - Philips type cassette (70,000 characters)
  - Remote control

- TEC 455 CRT -- The TEC 455 CRT is a teletype compatible terminal used to interact with the central site Honeywell 6080 computer. The CRT operates in a teletype compatible mode to provide visual output of plotted variables at the remote site.

  - Transmission speed selectable (110-1200 band)
  - Hardcopy adapter
  - 1920 characters/display
  - RS 232 interface to modem

- Scope Data Inc. 200 Series Receive Only Printer -- The Scope Data Printer interfaces directly with the CRT to provide a means to obtain hardcopy output of the current contents of the display screen.

  - RS 232 interface to CRT
  - Receive only printer
  - Selectable print speed (30-120 characters/second)

- Multi-Tech FM300 Acoustic Coupler -- The acoustic coupler provides a means to transmit and receive digital data to/from the central site computer.

  - Transmission speed (300 band)
  - Full/half duplex
  - Acoustic/direct mode

- Integrators -- Three integrators were designed and built by Honeywell to measure total flow solar radiation and useful heat collected.

Selection Criteria

The objective of the Data Acquisition System was threefold. First, the system was to collect data with minimal manual operation. Second, the system was to have the capability to process the data. Third, the system had to have the capability to store, selectively retrieve, and present the data at the remote site.

One obvious solution was to design a mini-computer system to collect, process, and store the data. This was a viable solution to the problem, and hindsight would suggest that it would have been the most optimal. The major reasons for not considering this option were the procurement costs (approximately $50K) and development time (6 months for hardware integration and software development).

Another solution, and the one finally adopted, was to develop a terminal system. This type of system would meet all three design objectives while minimizing costs and development time. Since the terminal basically operates as a time-share terminal, no software development was required to support the basic operation of the terminal. Further, these individual components are essentially "off-the-shelf" items and could be easily integrated into a workable system.

There are numerous manufacturers of terminal equipment. The application dictated few criteria to facilitate a selection process. In addition, all terminals were found somewhat price compatible, and the final choice was primarily based on availability and previous experience with the equipment. Selection criteria included:

- CRT -- Hardcopy adapter to permit printed copies of contents of CRT screen. A large screen capacity is required to facilitate plotting of data.

- Printer -- Compatible with CRT.

- Cassette Terminal -- Buffer recording capability. Remote control of operation. Compatibility with CRT and data logger.

- Acoustic Coupler -- 300 band transmission.

- Data Logger -- Compatibility with cassette recorder. Wide range of voltage, thermocouple, and external inputs. Printed output.

Software

Software was developed to support the application of the Data Acquisition System. This included:

- Analysis of Heating/Cooling Data -- Analysis and reporting of the heat balance of the heating/cooling system.

- Data Storage -- Organize and store collected raw data and calculated data.

- Data Retrieval -- Provides capability to query the data base and plot results on CRT.

No software development was necessary to support the operation of the terminal, since the terminal basically operates as a time-share terminal. The cassette system simulates a paper tape reader.

Figure 3-43 outlines the software programs, file system, and flow for the data acquisition system. The software is written in FORTRAN IV for operation on the H-6080 computer system. The file system is organized to take advantage of the capabilities and peculiarities of the H-6080 file software. Programs developed include:

- HEAT -- This program takes raw sensor data from temporary file 29, determines the mode of the heating system, calculates the associated heat balance for that mode, prints the results of the heat balance calculations for the day, stores the raw data into the temporary file TEMST, and stores calculated daily data into temporary files CAD20 through CAD23. Figure 3-44 illustrates the output from this program.

- COOL -- Program COOL operates in a fashion similar to that of program HEAT; only the calculations are run for the cooling system. The program uses the same file system.

- MAGTAP2 -- This program takes the raw data from temporary file 29 and copies the data onto 0.5-inch magnetic tape for permanent storage.

- QUESTION -- This program allows the user to query the data base of raw sensor data. The program allows the user to specify a channel or calculated data for a specific day. The program calculates hourly averages for daily data, automatically scales the ordinates, and plots the data on the CRT screen. A sample output is illustrated in Figure 3-45.

No program is required for data input. This process is automatically handled under the H-6080 File System. Since the cassette unit simulates a paper tape reader, this software allows the user to create an input file that contains raw sensor data.

File Description

The following data files have been established to hold raw and calculated data. The files were organized to take advantage of the H-6080 File System.

- File No. 29 -- This file contains raw sensor data for a period of one day. This file is created daily and supplies data to the heat and cooling mode programs.

- File No. 19 -- This file contains the titles for the plots of channel and calculated data.

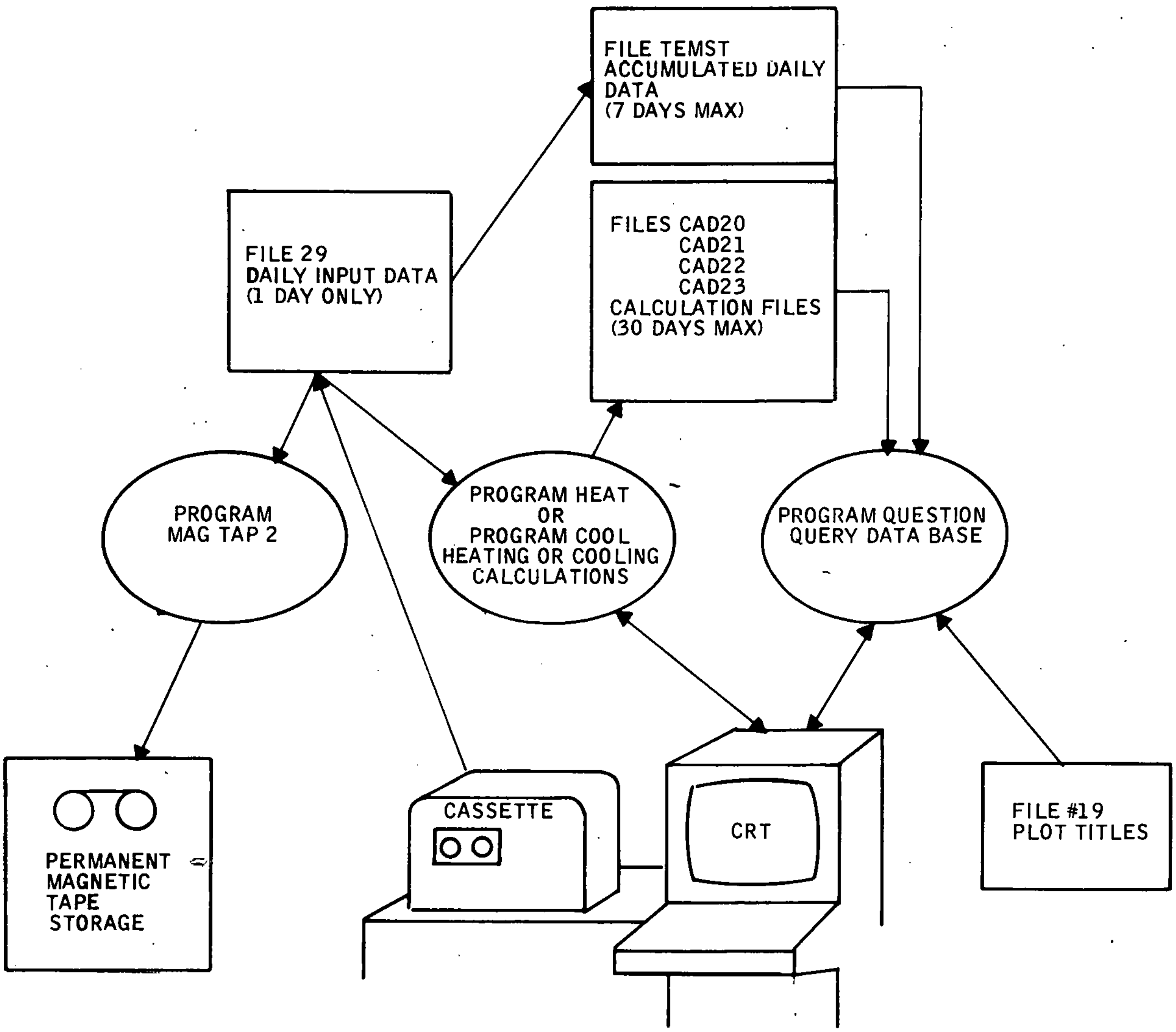


Figure 3-43. Software Flow Diagram

PAGE 1

DAY NO. 53

TIME IN EACH MODE

MODE 1	TIME	0.	HRS
MODE 2	TIME	0.	HRS
MODE 3	TIME	0.	HRS
MODE 4	TIME	0.	HRS
MODE 5	TIME	0.	HRS
MODE 6	TIME	0.	HRS
MODE 7	TIME	0.	HRS
MODE 8	TIME	0.	HRS
MODE 9	TIME	0.	HRS
MODE 10	TIME	0.	HRS
MODE 11	TIME	0.	HRS
MODE 12	TIME	0.	HRS
MODE 13	TIME	0.	HRS
MODE 14	TIME	0.	HRS
MODE 15	TIME	0.	HRS
MODE 16	TIME	0.	HRS
MODE 17	TIME	0.	HRS
MODE 18	TIME	0.	HRS
MODE 19	TIME	2.37	HRS
MODE 20	TIME	0.	HRS

TOTAL SOLAR ENERGY AVAILABLE FOR DAY,	352537.93	BTU
TOTAL HEAT COLLECTED FOR DAY	224410.04	BTUS
DAILY COLLECTER EFFICIENCY	63	PERCENT
TOTAL SPACE HEAT REQ,	5012.74	BTUS
AVE DAILY SOLAR CONTRIBUTION	032	PERCENT
DAILY TOTAL ENERGY EXTRACTED FROM TANK	-4031.80	BTUS

Figure 3-44. Heat Program Output Data

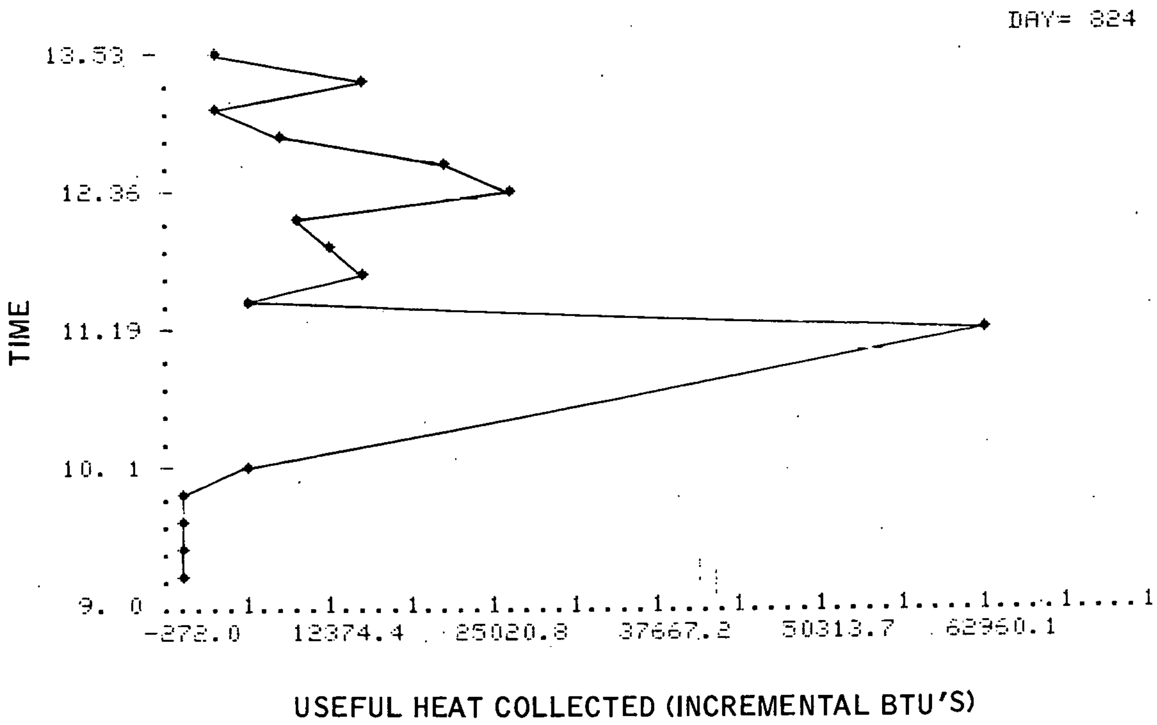


Figure 3-45. Sample Query Data Output

- File TEMST -- Contains accumulative raw sensor data for a period of seven days. This file supplies data for data base queries.
- Files CAD20 - CAD23 -- These files contain calculated data for data base queries. Data contained includes incremental solar energy, daily solar energy available, incremental heat collected, and daily heat collected, respectively. These files may be expanded to include other calculated data of interest.

VEHICLE DESCRIPTION AND MODIFICATIONS

Equipment Van

The equipment van was a Clark Model 45 DVE-2 Electronics Van, 8 feet wide, 45 feet long, and 13.5 feet high. The new empty weight was approximately 12,000 pounds. The inside floor steps up 18 inches at a distance of 9 feet from the front to accommodate the "fifth" wheel.

As originally purchased the van had conventional leaf spring suspension. However, because of the large glass areas supported by the body, air suspension was substituted.

Major modifications were made to the body and interior to make it suitable for use. The primary changes are as follows:

- The structure was reinforced to carry the weight of the solar panels mounted on the roof and left side, and hinges and supports added.
- The structure was reinforced to prevent twisting and warping due to wind loads on the extended solar panels.

- Cutouts were made in the left side and the top of the body, and housings and mounting supports were added for the six hydraulic cylinders used to raise and lower the two solar panels.
- An 80-inch by 42-inch cargo door was added to the right hand side.
- A 30-inch by 70-inch personnel door was added in the left hand rear cargo door.
- Various openings were cut in the ceiling, walls, and floor for hot and cold air ducts, combustion air, economizer air, etc.
- The entire interior was insulated with 3.5-inch polyurethane foam insulation.
- A false suspended ceiling was installed.
- The floor space was divided into an engineering office, a water storage tank room, a boiler room, a vestibule, and a laboratory.
- All of the walls were finished with paneling except for gypsum board in the engineering office and the boiler room.
- In the laboratory and vestibule a computer-type floor was installed even with the floor forward of the step. The space under this floor houses the hydraulic system, the cold air return ducts, and much of the control wiring.
- 120-volt, 60-Hz outlets were provided in all of the walls at convenient intervals. The circuits in the laboratory are split, making an outlet easy to convert to 240 volts if necessary.
- Recessed fluorescent lamps are used to light all rooms in the van.

After completion of all modifications and loaded with the solar equipment, the total van weight is 57,000 pounds, has a width of 9.3 feet, and 14 feet high. It is parked along side the 50-foot display van. The two are connected by a stairway and a weather proof cover. This allows easy access from one unit to the other.

Conference Area and Display Van

This unit was custom built for comfort and durability by Whitley Manufacturing Company, Inc., South Whitley, Indiana. The construction is better than the average similar unit in order to make ti capable of traveling long distances without damage or deterioration.

The van is 50 feet long, 12 feet wide, and 11 feet 8 inches high. The floor height is 43 inches and its ground clearance is 33 inches. The structural design and all materials specified conform to or exceed the current standards of the Mobile Home Manufacturers' Association and National Fire Protection Association as set forth in ANSI Standard A119.1 and NFPA-501B. The electrical and mechanical design conform to the National Electrical Code (USAC CI-1968) except as otherwise permitted by A119.1 and NFPA-501B.

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 predictions may be invalidated by such conditions 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.

Literature on the analysis of solar energy collection dates from the 1880's. One of the well known treatments of solar energy collection is by Hottel and Woertz[1] in 1942. Hottel and Whillier[2] (1958) and Bliss[3] (1959) presented certain plate efficiency factors, which simplified calculating 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 major factor in analyzing and designing 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, there are short-term fluctuations due to variable cloud cover. A comprehensive analysis of a collector must consider this time-dependent aspect 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.

[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 flat plate program was developed with the following goals:

- The transient nature of the problem must be adequately treated.
- 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., in order 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 subsection describes the analytical procedures and main features of the program. (See Appendix A for more detail.)

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 3-46.

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_i 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 repeated until the desired time range is covered.

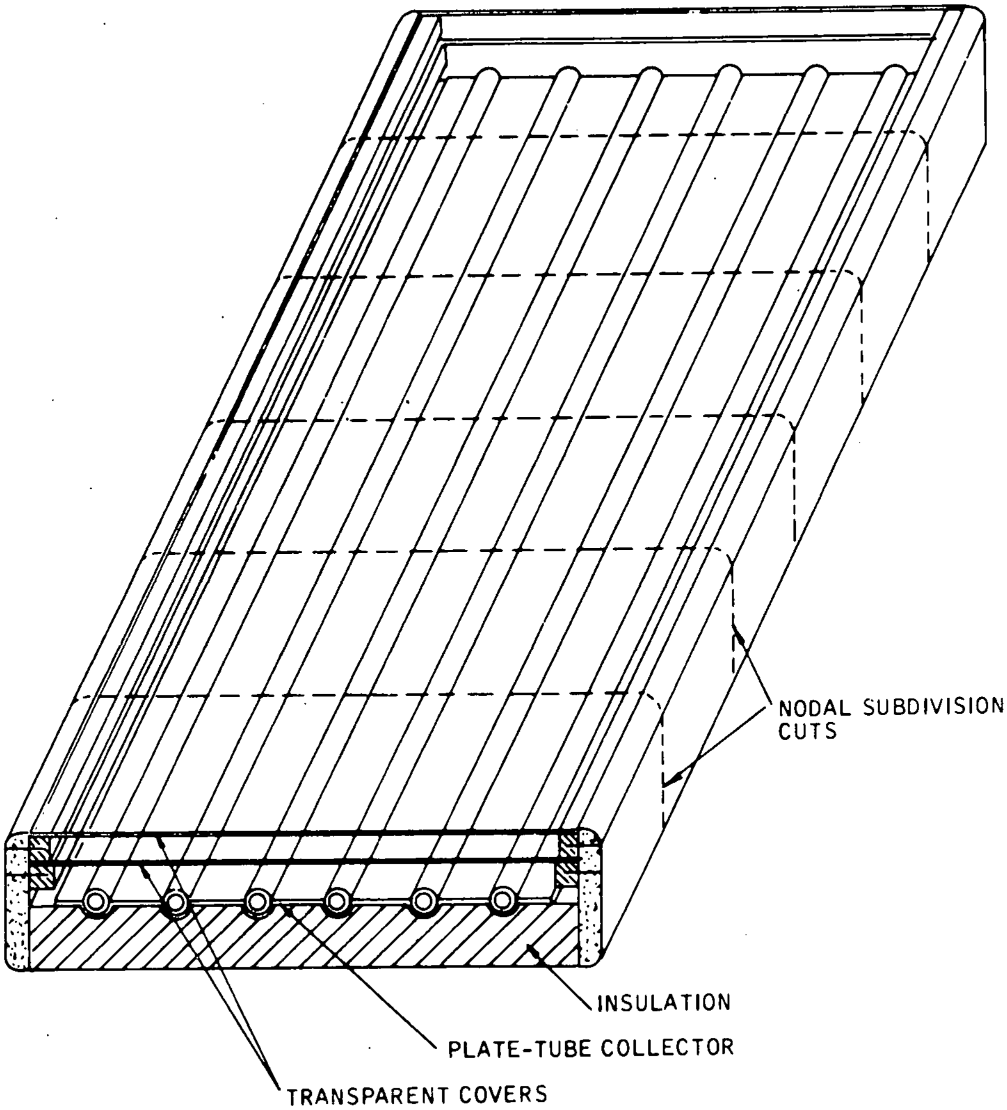


Figure 3-46. Flat Plate Solar Collector

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 considered 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:

- 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 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. Details on the pertinent heat transfer equations and their application in the prediction of the heat losses are given in Appendix A.

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.

Connected with the absorber-fluid heat transfer is the question of the temperature drop in the absorber because 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 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 useful in optimizing 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 resulst. (Special attention was given to this subject in the Honeywell program.)

In its present form, the input to the program is by 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 rearranged by the computer, and put in a convenient format for future reference. Figure 3-47 is a sample output of the geometrical, physical, and environmental parameters of a typical case.

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 3-48 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 specular or diffuse absorber.

The next output (Figure 3-49) 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 3-50) 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 3-51). This information indicates the magnitude of the various heat flow mechanisms and their relative importance.

The temperatures of each element at each time interval, as well as the rate of change of temperatures (Figure 3-52) are also given in the output.

The final output is an energy balance summary (Figure 3-53). 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.

The program is modular to provide for adding new features and modifying or replacing such routines as those used to compute heat transfer coefficients. The program consists of a main, executive routine, plus 17 subroutines, and is written exclusively in Fortran to economically implement it on any computer.

Sample Calculations -- Performance of a flat plate collector exposed to winter and summer solar fluxes is presented in Figures 3-54 and 3-55. The solar data -- sun position, direct normal flux, and diffuse flux -- were obtained from the ASHRAE Handbook of Fundamentals[4] for 40 degrees north altitude, a representative latitude in the United States. From this data the incident angle and total flux were computed and are plotted in Figures 3-56 and 3-57. February 21 was selected as representative of the available winter solar energy

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

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DYNAMIC ANALYSIS OF FLAT PLATE SOLAR COLLECTOR SYSTEM

PHYSICAL DATA

   COLLECTOR
      LENGTH, M                                    .12192E 01
      WIDTH, M                                     .12192E 01
      NO. COVERS, 1
      ABSORBER TO COVER NO.1 GAP, CM               .25400E 01
   STORAGE TANK
      LENGTH, M                                    .10000E 02
      DIAMETER, M                                  .10000E 03
      VOLUME, CU M                                 .78540E 05

   ABSORBER
      MATERIAL, ALUMINUM
      SURFACE CONDITION, SPEC
      SOLAR ABSORPTANCE,                           .90000E 00
      INFRARED EMITTANCE,                          .10000E 00
      THICKNESS, CM                                .15240E 00
      DENSITY, KGM/CU M                            .27130E 04
      SPECIFIC HEAT, W-SEC/KGM-C                   .96300E 03
      THERMAL CONDUCTIVITY, W/M-C                  .17100E 03
      NO. TUBES, 12
      TUBE ID, CM                                  .95250E 00
      TUBE OD, CM                                  .12573E 01
      TUBE LENGTH, M                               .12192E 01
      TUBE SPACING, CM                             .10160E 02

   COVER NO.1(INNER)
      MATERIAL,    GLASS
      INDEX OF REFRACTION                          .15260E 01
      EXTINCTION COEFFICIENT, 1/CM                 .78700E-01
      INFRARED EMITTANCE                           .90000E 00
      THICKNESS, CM                                .30400E 00
      DENSITY, KGM/CU M                            .87000E 04
      SPECIFIC HEAT, W-SEC/KGM-C                   .80000E 03
      THERMAL CONDUCTIVITY, W/M-C                  .76000E 00

   INSULATION
      MATERIAL,GL FIBER
      SOLAR ABSORPTANCE,                           .90000E 00
      INFRARED EMITTANCE,                          .90000E 00
      THICKNESS, CM                                .76200E 01
      DENSITY, KGM/CU M                            .12810E 02
      SPECIFIC HEAT, W-SEC/KGM-C                   .67000E 03
      THERMAL CONDUCTIVITY, W/M-C                  .40270E-01


ENVIRONMENTAL DATA

   AMBIENT CONDITIONS
      TEMPERATURE, C                               .32220E 02
      PRESSURE, BAR                                .10133E 01
      WIND VELOCITY, M/SEC                         .33528E 01
```

Figure 3-47. Physical and Environmental Data

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	.81213	.02585	.00000	.00000	.00000
	PARALLEL	.16413	.83587	.81030	.02557	.00000	.00000	.00000
	PERPENDICULAR	.15992	.84008	.81423	.02549	.00000	.00000	.00000
18.0	TOTAL	.16215	.83785	.81157	.02628	.00000	.00000	.00000
	PARALLEL	.17081	.82919	.80294	.02625	.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	.80641	.02745	.00000	.00000	.00000
	PARALLEL	.20470	.79530	.76756	.02772	.00000	.00000	.00000
	PERPENDICULAR	.12757	.87243	.84445	.02798	.00000	.00000	.00000
45.0	TOTAL	.17468	.82532	.79637	.02895	.00000	.00000	.00000
	PARALLEL	.23944	.76056	.73154	.02872	.00000	.00000	.00000
	PERPENDICULAR	.10992	.89008	.86021	.02916	.00000	.00000	.00000
54.0	TOTAL	.19574	.80424	.77410	.03015	.00000	.00000	.00000
	PARALLEL	.29575	.70425	.67446	.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	.58125	.03266	.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	.03266	.00000	.00000	.00000
81.0	TOTAL	.58986	.41014	.37878	.03136	.00000	.00000	.00000
	PARALLEL	.72874	.27126	.24034	.03142	.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	.74670	.02695	.00000	.00000	.00000
	PARALLEL	.29719	.70286	.67420	.02867	.00000	.00000	.00000
	PERPENDICULAR	.14756	.85250	.82020	.02231	.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	.81800	.02605	.00000	.00000	.00000
	PARALLEL	.15596	.84404	.81800	.02605	.00000	.00000	.00000
	PERPENDICULAR	.15596	.84404	.81800	.02605	.00000	.00000	.00000
9.0	TOTAL	.15596	.84405	.81788	.02617	.00000	.00000	.00000
	PARALLEL	.15629	.84372	.81555	.02616	.00000	.00000	.00000
	PERPENDICULAR	.15362	.84638	.82020	.02618	.00000	.00000	.00000
18.0	TOTAL	.15613	.84386	.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	.02898	.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	.72379	.03126	.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	.03156	.00000	.00000	.00000
	PARALLEL	.54238	.45763	.42570	.03093	.00000	.00000	.00000
	PERPENDICULAR	.18393	.81606	.78366	.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	.21996	.78010	.75109	.02901	.00000	.00000	.00000
	PARALLEL	.30057	.69949	.67080	.02869	.00000	.00000	.00000
	PERPENDICULAR	.13935	.86071	.83139	.02933	.00000	.00000	.00000

Figure 3-48. Analysis of 1 Cover System Plus Absorber with Solar Absorptance = 0.90

TIME, HR = .1200E 02, INTEGRATION INTERVAL, HR = .2717E-02, CONVERGENCE FACTOR = .9000E 00

SOLAR HEAT ABSORPTION, W

NODE NO.	SPECTRAL	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 3-49. Solar Heat Absorption

TIME =12.00 HR

INTER-NODAL CONDUCTANCES, W/C

NODE NO.

1	.87088E 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	.15869E 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	.15869E 01
	.15869E 01	.16090E 02	.33763E 00	.00000E 00	.00000E 00	.00000E 00	
4	.87966E 00	.29154E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.15869E 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	.15869E 01
	.00000E 00	.15466E 02	.33763E 00	.00000E 00	.00000E 00	.00000E 00	
6	.00000E 00	.00000E 00	.87088E 00	.27437E 00	.61601E 01	.18096E 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	.18333E 01	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
8	.00000E 00	.00000E 00	.87613E 00	.28737E 00	.40949E 01	.18445E 01	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
9	.00000E 00	.00000E 00	.87966E 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	.36308E 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	.16090E 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 3-50. Internodal Conductances

TIME =12:00 HR

INTER-NODAL HEAT FLOWS, W

NODE NO.							
1	-.51546E 02	-.16739E 02	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.64305E 01	-.14706E 03	-.11342E 02	.00000E 00	.00000E 00	.00000E 00	
2	-.52866E 02	-.17114E 02	.00000E 00	.00000E 00	.00000E 00	.00000E 00	-.64305E 01
	.41770E 01	-.13557E 03	-.11949E 02	.00000E 00	.00000E 00	.00000E 00	
3	-.54277E 02	-.17503E 02	.00000E 00	.00000E 00	.00000E 00	.00000E 00	-.41770E 01
	.36909E 01	-.13482E 03	-.12359E 02	.00000E 00	.00000E 00	.00000E 00	
4	-.55683E 02	-.18454E 02	.00000E 00	.00000E 00	.00000E 00	.00000E 00	-.36909E 01
	.30904E 01	-.13227E 03	-.12723E 02	.00000E 00	.00000E 00	.00000E 00	
5	-.56890E 02	-.19016E 02	.00000E 00	.00000E 00	.00000E 00	.00000E 00	-.30904E 01
	.00000E 00	-.12770E 03	-.13129E 02	.00000E 00	.00000E 00	.00000E 00	
6	.00000E 00	.00000E 00	.51546E 02	.16239E 02	-.57999E 02	-.17038E 02	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
7	.00000E 00	.00000E 00	.52866E 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	-.54615E 02	-.24602E 02	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
9	.00000E 00	.00000E 00	.55683E 02	.18454E 02	-.54706E 02	-.26528E 02	.00000E 00
	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	
10	.00000E 00	.00000E 00	.56890E 02	.19016E 02	-.54854E 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	.11949E 02	.00000E 00	.00000E 00	-.11946E 02	
13	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.00000E 00	.12359E 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	.13129E 02	.00000E 00	.00000E 00	-.13022E 02	
16	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00	.00000E 00
	.00000E 00	.14706E 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 3-51. 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	.34782E-01
7	.44301E 02	.71668E-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 3-52. Temperature, C, and Rates, C/H

TIME =12.00 HR

ENERGY SUMMARY

	INSTANTANEOUS W	ACCUMULATED 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 3-53. Energy Summary

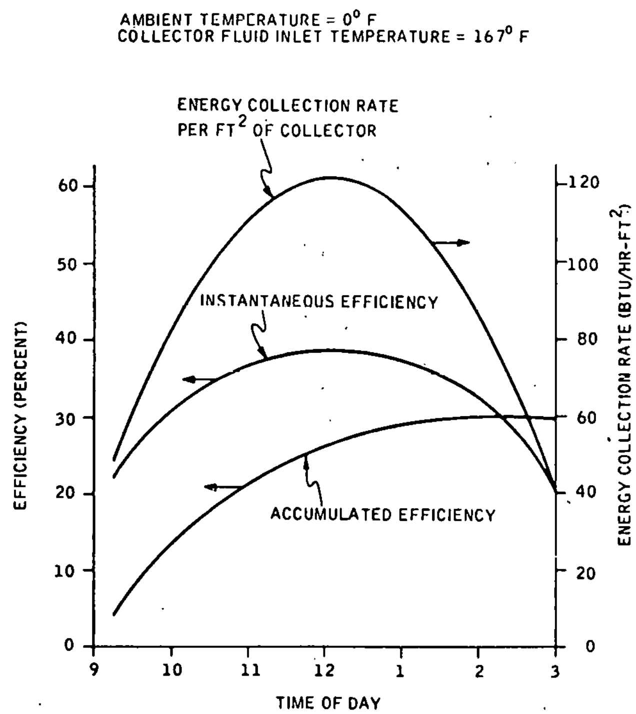


Figure 3-54. Flat Plate Collector Performance at 40° N. Latitude on 21 February

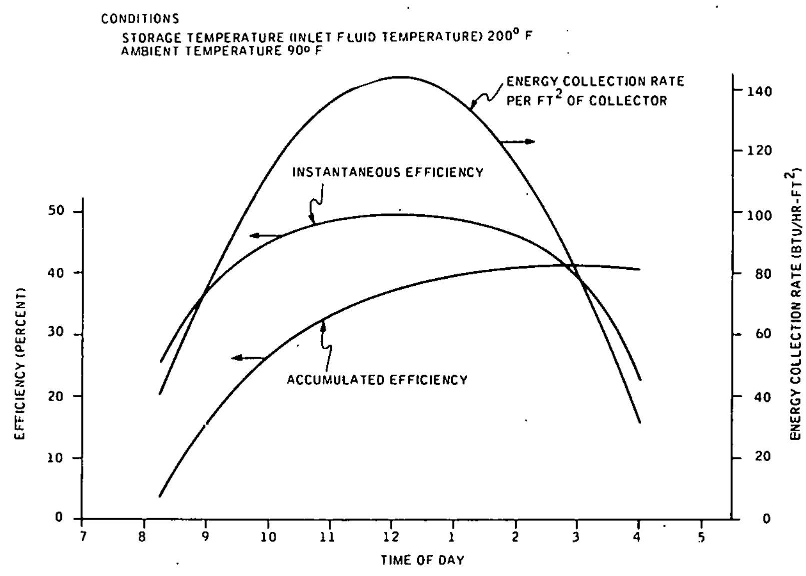


Figure 3-55. Flat Plate Collector Performance at 40° N. Latitude on 21 July

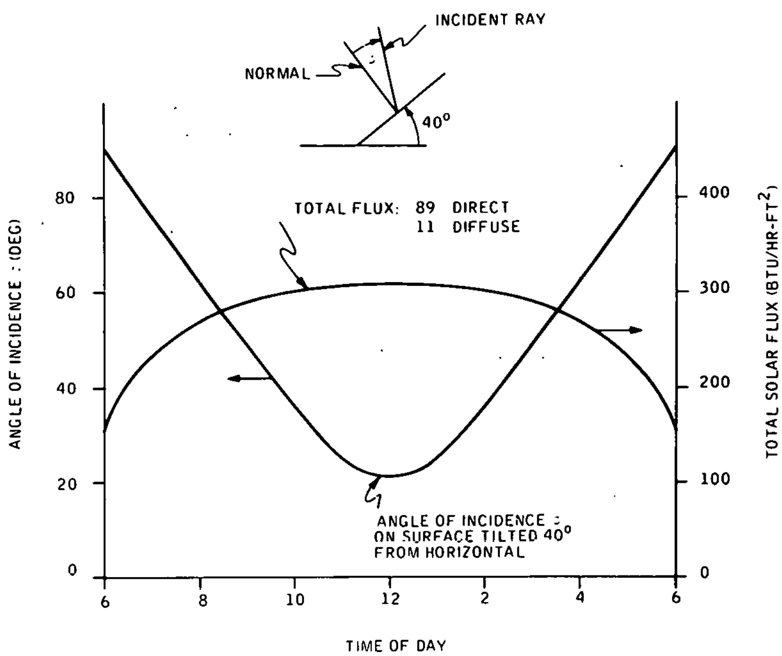


Figure 3-56. Solar Flux and Angle of Incidence at 40° N. Latitude on 21 February

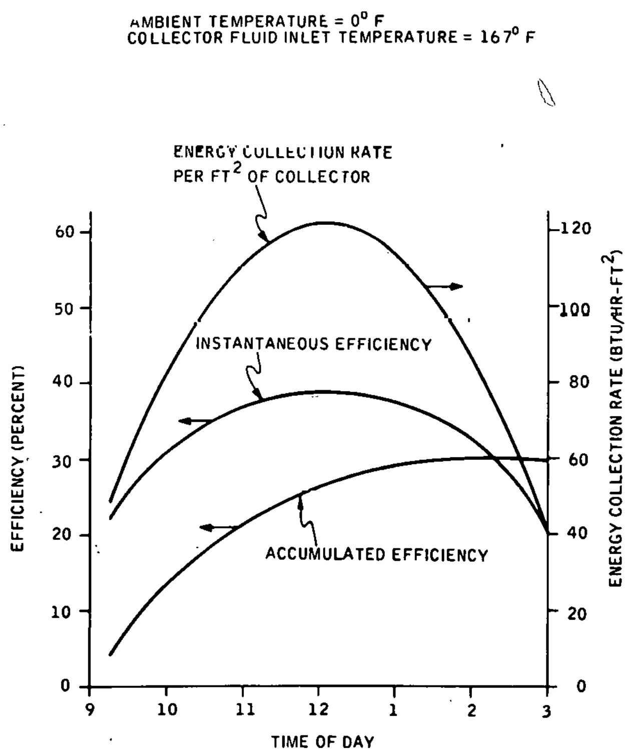


Figure 3-57. Solar Flux and Angle of Incidence at 40° N. Latitude on 21 July

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 winter and summer collection. The pertinent physical and environmental parameters assumed are listed in Table 3-5.

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.

Table 3-5. 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 mph in winter
Collection Fluid	
Material:	Specially inhibited 50/50 ethylene glycol and water
Flow Rate:	0.25 gpm

Figure 3-54, 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 122 Btu/hr-ft^2 at noon. The integration of this curve gives the daily collected energy, which is 605 Btu/ft^2.

The summer collector performance (Figure 3-55) 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 instantaneous 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.

Aqua-Ammonia Absorption Refrigeration Cycle Analysis

Program Description -- The absorption refrigeration cycle is a natural choice for the air conditioner in a solar powered system since it requires no electrical power (other than a small, fractional horsepower circulating pump). The system consists of a heat-actuated cycle in which heat is removed from two reservoirs, the source of heat for the generator and the cooling load, and is added to a third reservoir, the condenser cooling air. The effectiveness of the cycle, expressed as a coefficient of performance, is strongly affected by the temperature of the heat source. Since the solar flux and resulting collection fluid temperature vary throughout the day, the generator temperature in the absorption cycle and the resulting coefficient of performance (C.O.P.), will also be time dependent. If the hourly and integrated daily performance of a solar collector-absorption cycle air conditioner are to be predicted, performance of the absorption cycle as a function of its generator temperature must be calculated.

A computer program written to perform this task has two major sections: the calculation of the thermodynamic properties of aqua-ammonia mixtures in the liquid and vapor state; and the calculation of the performance of the absorption cycle. (See Appendix B, Analysis of the Absorption Refrigeration Cycle.)

The simplified cycle is shown schematically in Figure 3-58. The system consists of two fluid loops. The outer loop (1-2-3-9-4-10-6-8-12) can be termed the refrigerant loop. In the absorption system, the vapor is absorbed into a liquid absorbent, pumped to a higher pressure, and then desorbed to enter the condenser at this pressure. The inner loop (3-9-4-5-7-13) is the absorbent loop.

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 in Figure 3-59, 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 the supply header and the collection header at the tube ends. The header pressures will vary along their length due to (1) wall friction losses and (2) momentum flux changes in the headers as fluid is withdrawn or added. The pressure in the supply manifold will fall in the flow direction due to wall shear stress but the drop will be reduced by pressure recovery at each cross-tube due to extraction of fluid and subsequent loss of momentum

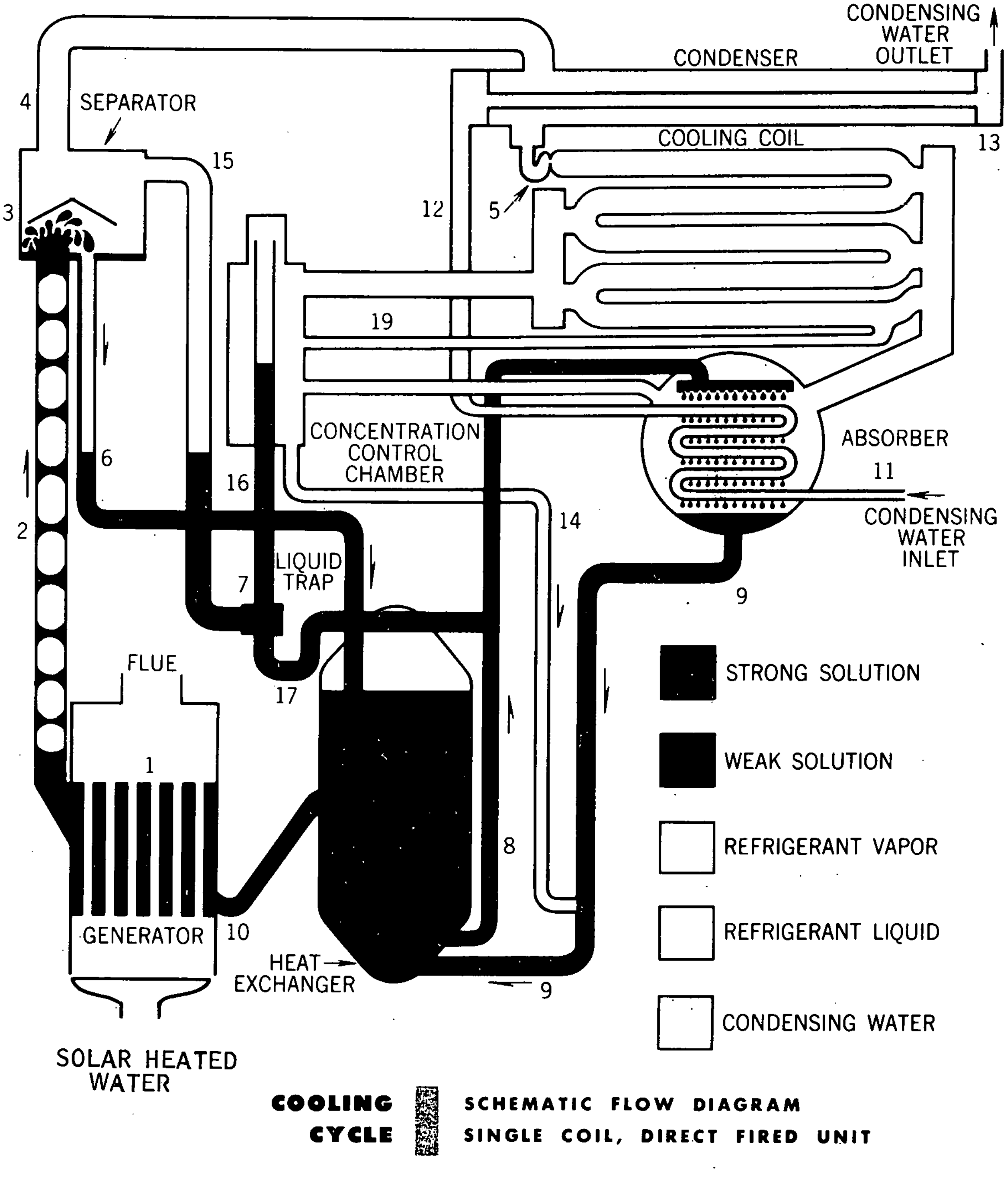


Figure 3-58. Absorptive Air Conditioning System

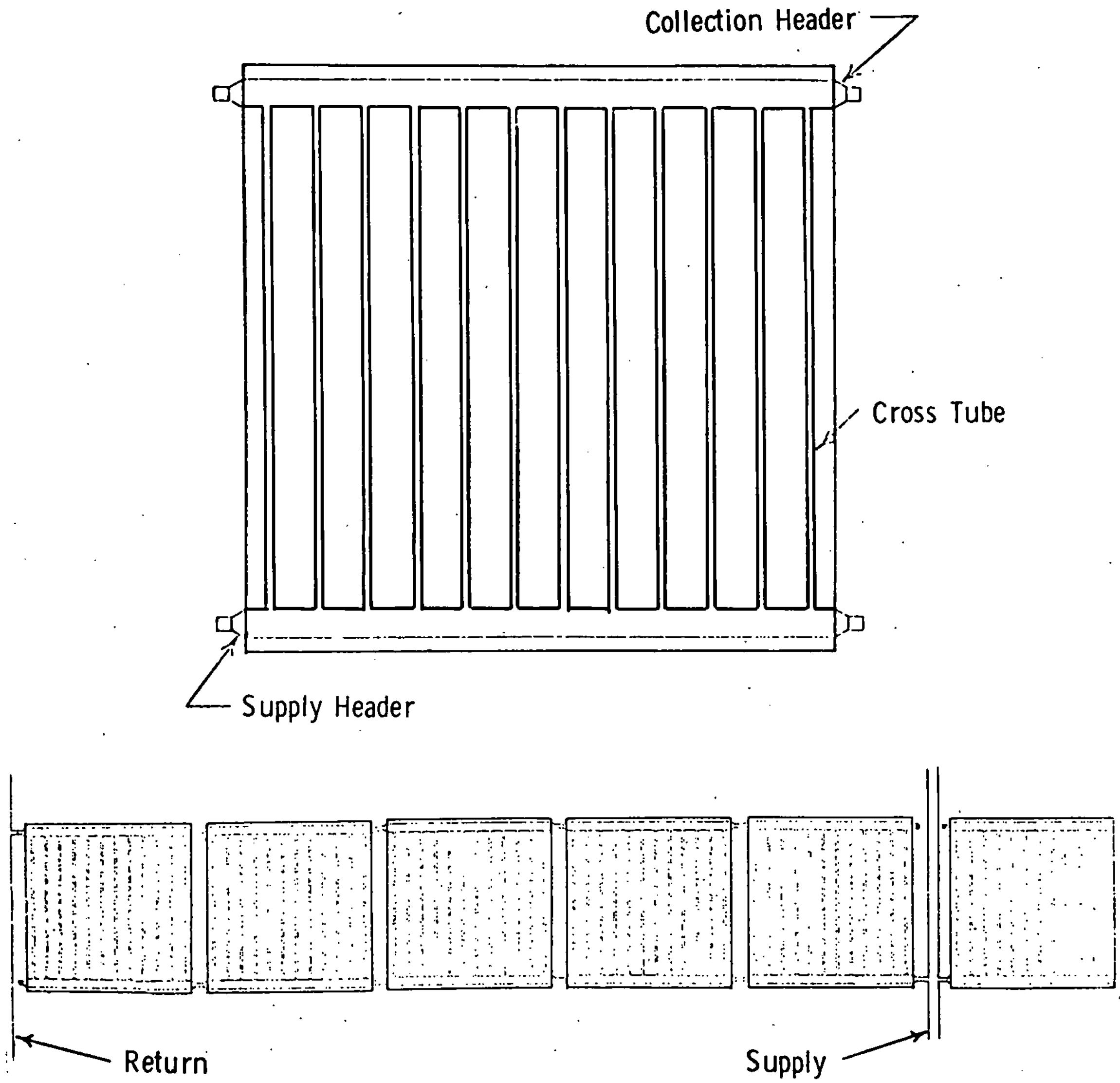


Figure 3-59. Schematic of Individual Collector and Potential Array Configuration

in the supply header flow. In the collection header the frictional drop is reinforced by the accleration 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 (Figure 3-59). This number of collectors was chosen as typical. A complete array would consist of several rows. The present analysis is concerned with the distribution 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 in Figure 3-59 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 (Figure 3-59). The supply and collection headers are on 42-inch centers and the cross-tubes are 40 inches long on 3.552 inch 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 spacing and dimensions have been arrived at from the results of the present analysis.

Figures 3-60 to 3-62 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 3-60 for the 0.050 inch by 0.500 inch tube, note 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 3-61 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 3-62, 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 inch of water per °F, the temperature variation will significantly improve the flow distribution for the 0.200 inch by 0.500 inch tubes, but very little for the thinner tubes.

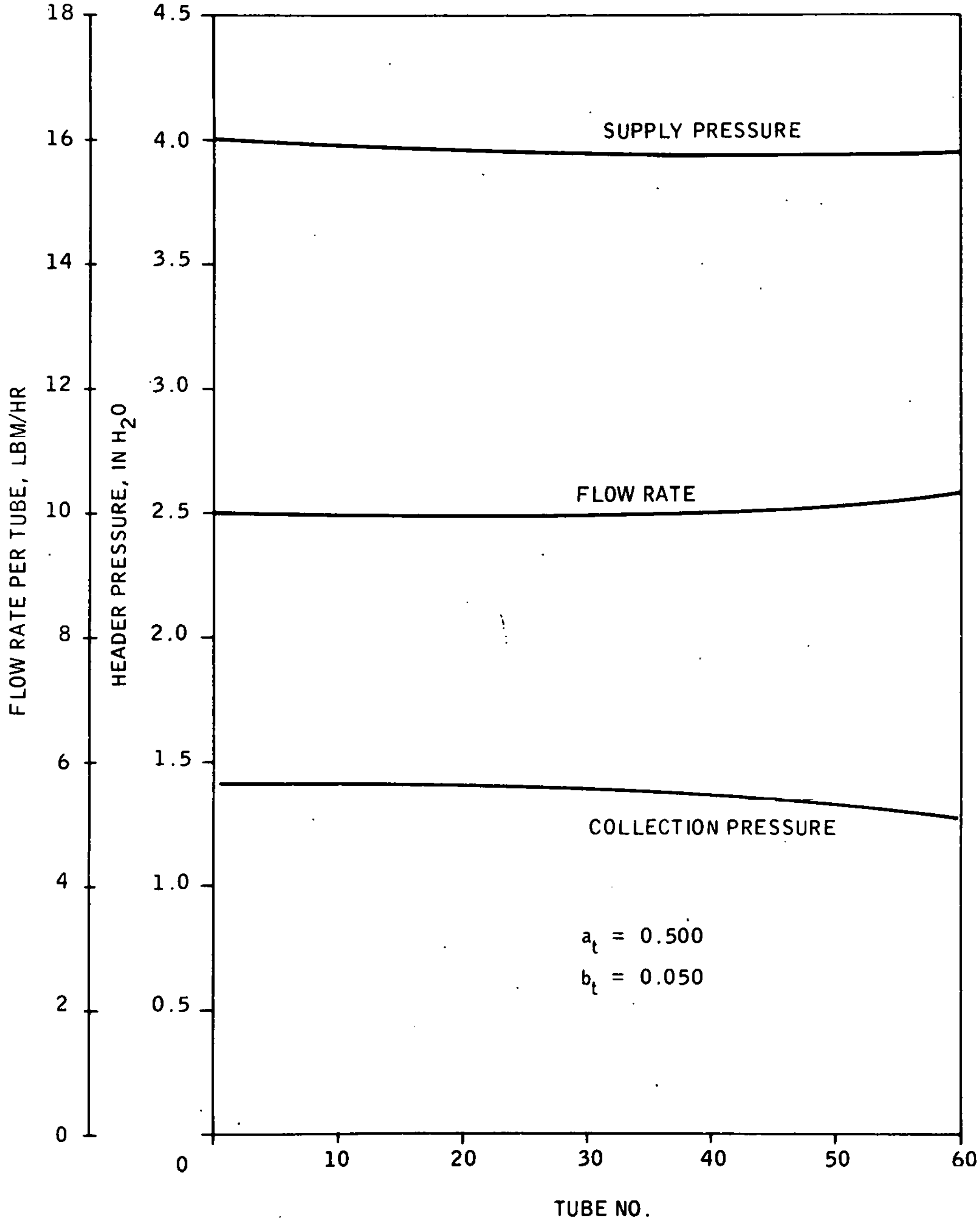


Figure 3-60. Flow Distribution for 0.05 Inch by 0.50 Inch Cross Tube

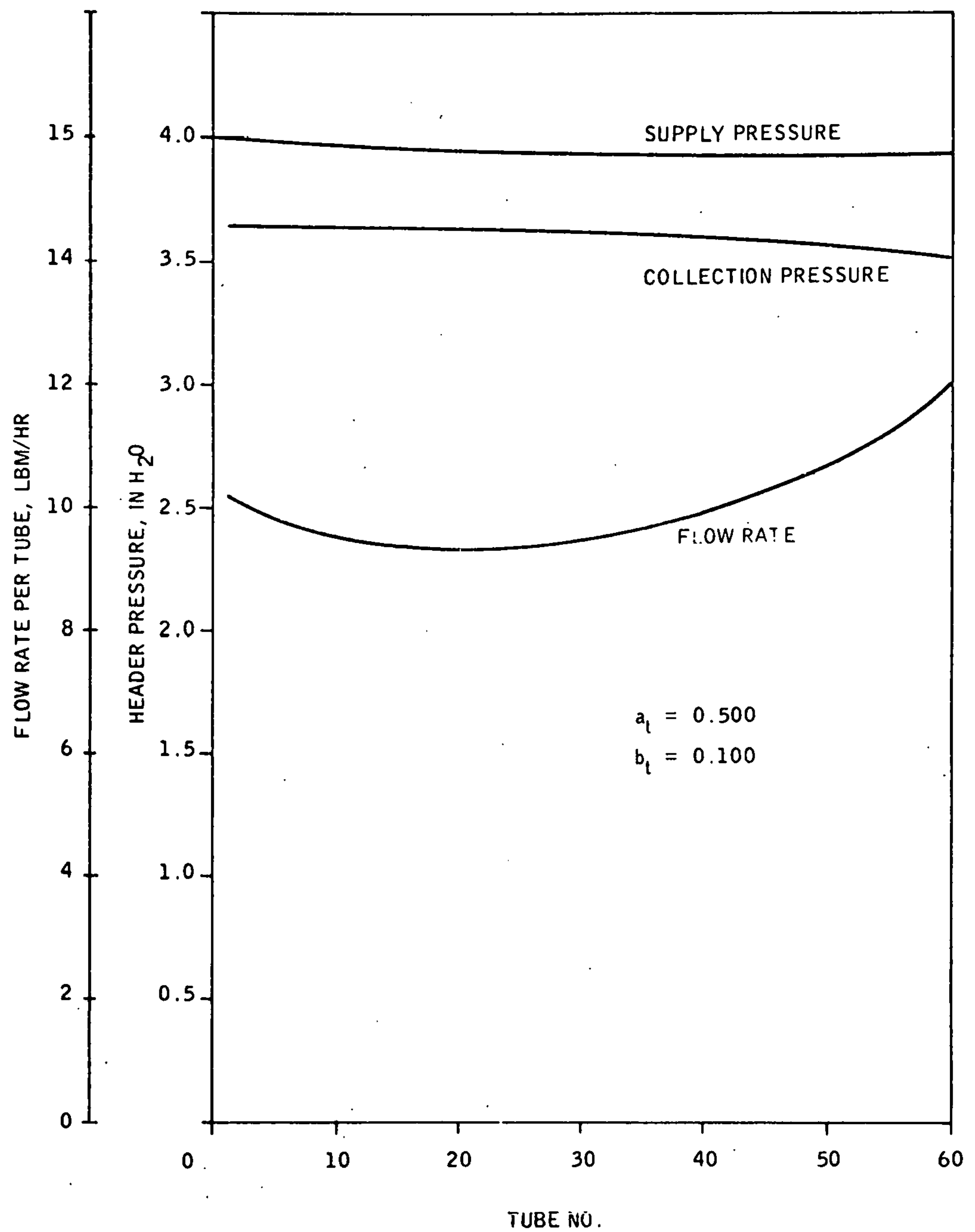


Figure 3-61. Flow Distribution for 0.1 Inch by 0.5 Inch Cross Tube

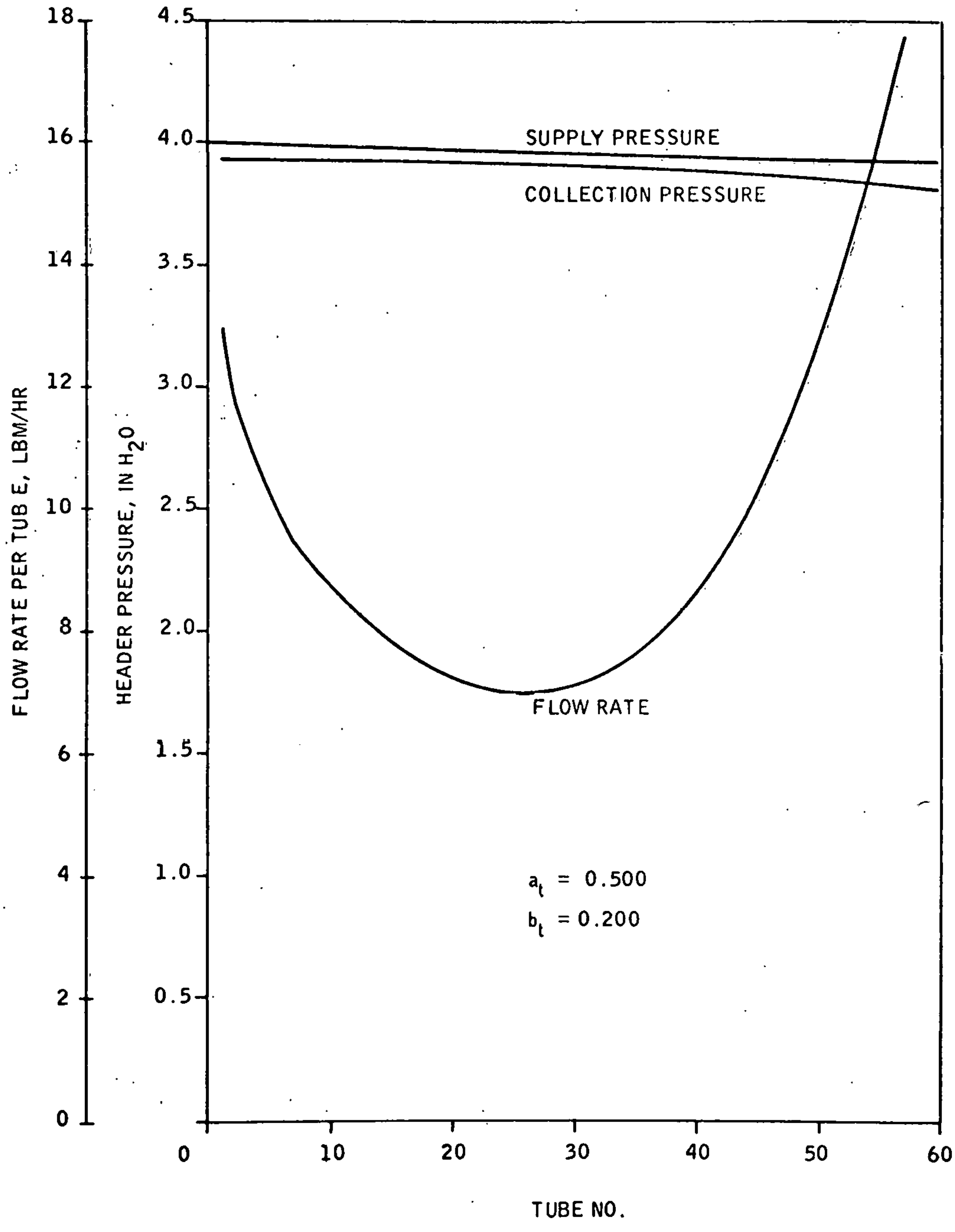


Figure 3-62. Flow Distribution for 0.2 Inch by 0.5 Inch Cross Tube

COLLECTOR TRADEOFF STUDIES

The information in this subsection discusses the tradeoff studies made in arriving at the design selected for the T.S.L. In many cases, the actual selection mode was not based entirely on superior performance but on availability, state of the art, transportability, etc. Therefore, this subsection may have conclusions that were not the direction taken in the TSL design. This subsection is included to help designers select other avenues of design approach more amenable to their specific applications.

Absorber Coatings -- The absorbing coating for a flat plate collector may be either selective or nonselective. If the collector application is solely for heating, then a nonselective coating might be preferable. 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 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 3-6. Among the coatings shown, Honeywell has experience with the three highest in 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 3-6. 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

Fabrication and Material Costs -- Cost of fabrication will be a major factor in absorber plate design. Attention must be given to designs which lend themselves to automatic fabrication and testing. Material costs vary, but at the present time minimizing the use of copper and aluminum consistent with good thermal design appears likely. Considerations in design should also include ease of installation of the absorber plate into the collector assembly.

Honeywell has been conducting studies related to fabrication costs of absorber plates in large production quantities based on steel absorber material. These studies were made by Honeywell production engineers and will be applied to this program to minimize production and material costs. It is anticipated that consultations with these people at an early phase of the design program should lead to an integrated design approach without any major production problems.

Operating Pressures and Flow Distribution -- Absorber plates operating at temperatures in excess of 220°F with water as the circulating fluid will require pressure in excess of atmosphere. The addition of ethylene-glycol will raise the boiling point to 230°F at atmospheric pressure with a 50-50 water-glycol mixture.

Various flow channel configurations have been analyzed for uniform flow distribution in arrays of serial and parallel connections. Consideration has been given to pumping energy and thermal efficiency. The collector system analysis illustrates the analysis conducted on the steel panels and manufactured by Honeywell. This analysis determined that channels of 0.050 x 0.250 inch were optimum from thermal and flow distribution standpoints; but, as mentioned earlier, appear not to be optimum for corrosion and particulate contamination when used in practice.

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 3-63.

Honeywell's investigation of black nickel coatings included process scale-up to 3 x 4 ft 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 by carefully placing 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 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 not known if they lead to a significant long-term improvement.

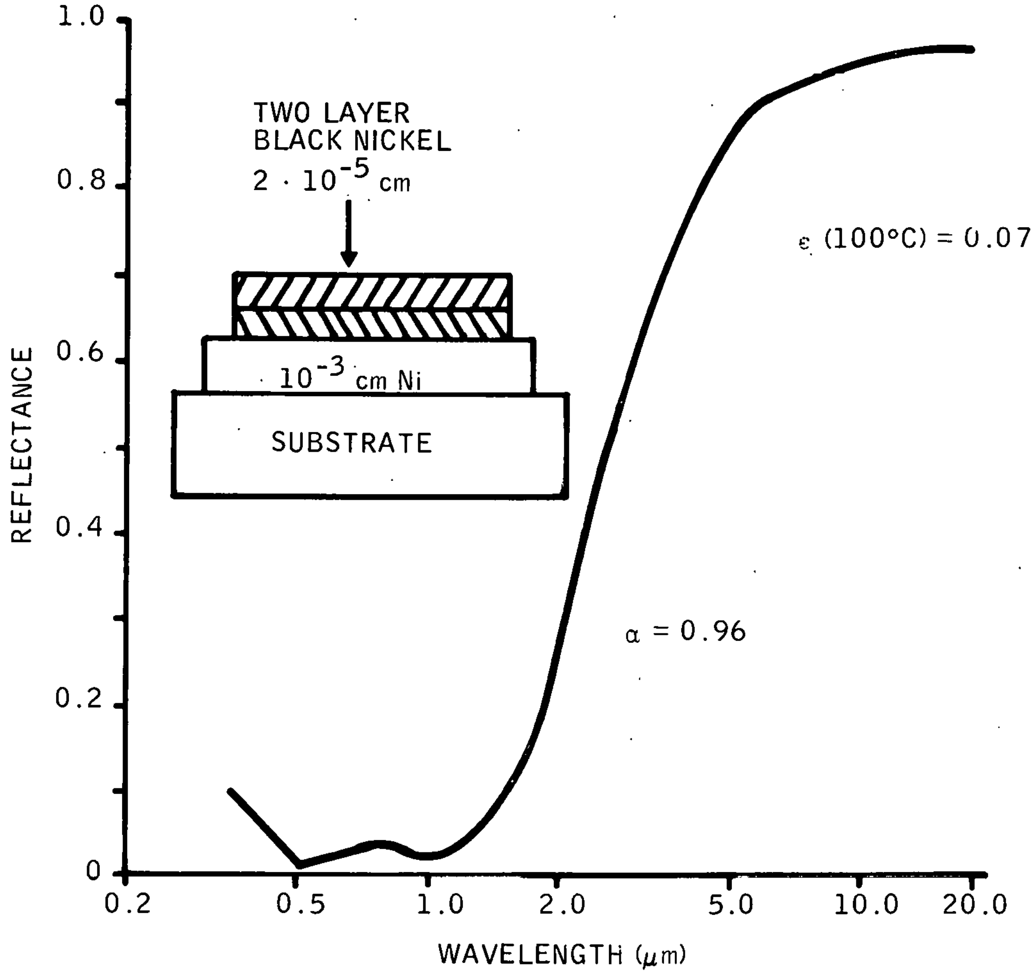


Figure 3-63. Spectral Reflectance

Black Chrome -- Black chrome is a commercial electroplated chrome oxide coating with diffuse reflectance properties. Manufacturer's data indicates that the coating remains black up 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).

Copper Oxide -- Our experience with CuO coatings is rather limited, since the performance achieved in early studies could not greatly improve upon values in the literature 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's experience has been primarily with Ni-coated steel substrates 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 the 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¢/mil ft^2 for Ni alone).

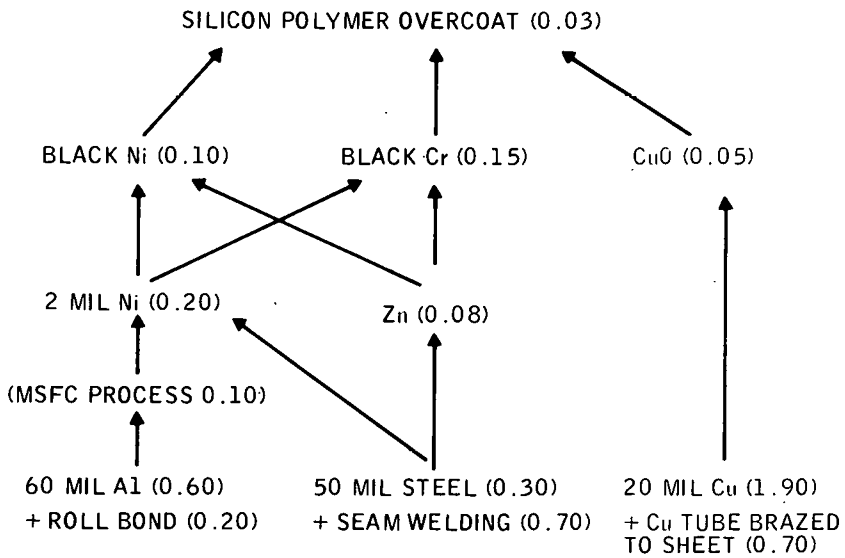
A most important, and poorly understood, requirement for the absorber coating is long-term durability. Candidate coatings should be durable to all reasonable environmental degradation mechanisms expected for over a 15-year span. Thermal runaway conditions of 400°F, humidity, and thermal cycling must be withstood. Long-term corrosion due to combined effects of humidity, 220°F temperatures, and dissolved CO_2 and SO_2 gases must be minimal. The question of absorbing coating durability must be answered. The selection of tests to evaluate coating durability must be carefully made in order to allow meaningful extrapolation of short-term accelerated test results to predict long-term lifetimes.

The cost and optical efficiency must be tied together in analysis. Although the use of selective coatings is justified for 200°F flat plate collectors, their cost is a significant consideration, and decisions concerning candidate coatings must be based on cost-efficiency. Examples of cost-efficiency questions which should be addressed include:

- What is the minimum Ni coating thickness for acceptable durability?
- Is the higher emittance of coatings deposited on Zn or Cu justified by the lower cost of such coatings?
- Can absorption values be further improved?

*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.

Some preliminary estimates of large-scale coating-substrates costs are given in the diagram below. Each item is followed, in parentheses, by the estimated cost in dollars/ft^2. The costs are based on the present industrial rates and/or our estimate of the process. The estimated cost of a particular candidate coating-substrate combination can be found by adding the component costs on the diagram. For example, the estimated cost of a black Cr (0.15) coated galvanized (0.08) steel collector panel (0.30 + 0.70) would be \$1.23/ft^2.



The use of Cu as a basic panel would seem to be unlikely due to high material cost; hence, Cu may also be considered as an intermediate coating for a cheaper metal substrate, i.e., like Ni and Zn.

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.

Testing performed under contract to NASA-Lewis Research Center 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 3-64.

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 round, then the two-cover configuration is preferable.

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 considered include: glass, tempered and anti-reflection 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

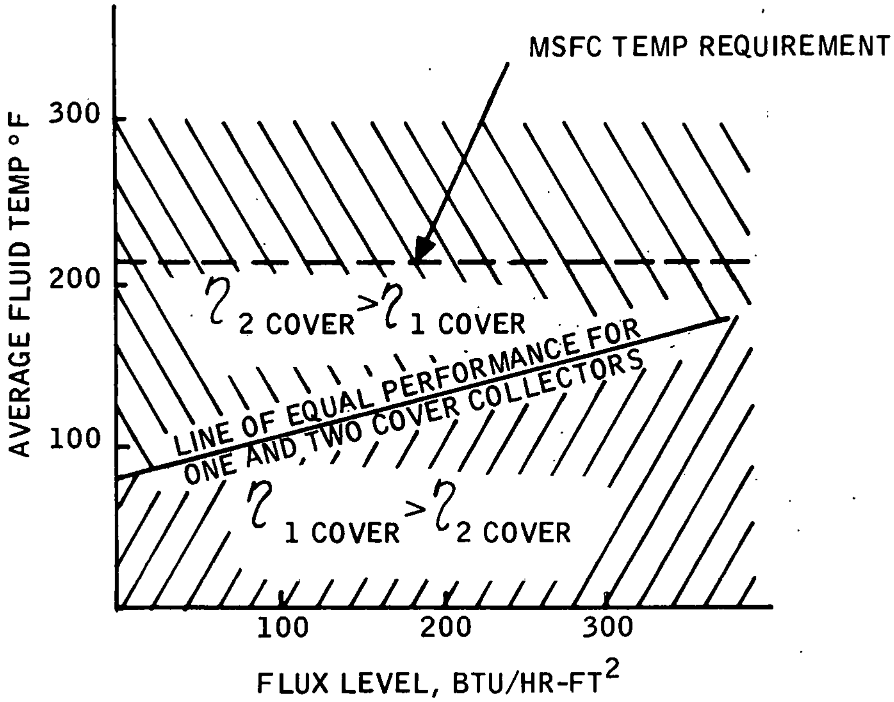


Figure 3-64. Performance Preference Curve

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.

Mylar should be considered as an inner cover material due to its very low cost, \$0.01 to \$0.02 per ft^2. Although known to degrade under UV, the first cover will reduce the level of UV reaching the second cover. It should be determined if the reduction is adequate to protect Mylar. Honeywell is currently testing solar collectors on the Transportable Laboratory with both Tedlar and Mylar inner covers.

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

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 included appropriate brackets and supports to fasten the covers to the collector housing.

*Based on June 1974 material and labor costs.

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 distinctly salient features, yet both also have drawbacks significant enough to warrant a detailed examination of their impact on each specific collector application and its installation requirements.

While under contract to NASA-Lewis Research Center, 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. Consequently, the current housing design for the NASA-Lewis collector has the equivalent of 3 inches of soft insulation around the sides of the housing, the equivalent of 6 inches of soft insulation on the bottom of the housing, and bakelite mechanical absorber supports connected to the bottom of the housing.

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, in the collector module approach, edge losses can also be reduced by making the housing larger than the enclosed absorber panel to accommodate several inches of insulation.

Apart from heat loss, the other major concern for evaluating the candidate approaches is installation and maintainability once installed. The collector array integrates readily into the roofline of buildings. By using the existing roof as a backing plate, construction is reduced to merely constructing a frame to support the cover system. A modular collector installation, however, would probably use a separate supporting framework. This makes it well-suited for ground and building installations where it is either impossible or undesirable to use an existing roof.

The installation and subsequent collector maintenance is a distinct advantage of the collector module approach. The collector array must be mounted from above, necessitating cranes or scaffolding. Maintenance must also be 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.

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 it appears 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), workability, 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 equally cost effective.

Insulation -- The back surface of the collector absorber plate is thermally insulated to minimize the amount of heat loss 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 3-7. The federal specification numbers are included for reference.

Table 3-7. 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

*Based on June 1974 labor and material costs.

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 temperatures 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 3-3 are 225°F and 165°F, respectively.

The 45-degree 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.

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 fiberlgass 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 higher 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.

Collector Installation and Integration -- Collector design efforts should consider the following problem areas in addressing collector installation and integration:

- Fluid connections
- Flow distribution channels (main header)
- Collector mounting in arrays
- Repair or removal of collectors

Fluid Connections -- Recent experience at Honeywell with connections between adjacent collectors where the header is an integral part of the absorber plate has led to the following conclusions regarding fluid connections.

- Rubber hose connections leak.
- Existing hoses composed of Buna N rubber take almost a 100 percent compression set when clamped at 300°F, thus eliminating any clamping force.
- Any exterior hoses exposed to ultraviolet radiation decompose with time.
- Interconnections between adjacent panels leak into the collector box, degrading insulation performance.

These problems led to a decision to remove all rubber hose connections on the main header supply and replace them with a silicone "O" ring and tube connection. This type of connection is stable from -80°F to +450°F and is relatively easy to install.

Fluid connections to a main header should be outside of the collector box if possible. This permits easier installation and removal of an individual collector, and any leakage would be easier to detect. Individual connections should be semipermanent screw-type fittings or silicone "O" ring and screw-type fittings of noncorrosive material.

Flow Distribution Channels (Main Supply and Return Header) -- To provide uniform flow distribution, the main headers must be sized for a given number of collectors operating in parallel. These headers can be arranged in various ways to minimize nonuniform flow distribution. Studies should be made to optimize the geometric configuration of collector arrays to minimize header quantities.

SECTION IV

SYSTEM OPERATION

This section describes the necessary procedures and precautions for initially starting the system, and the function and use of the various auxiliary equipment required for proper operation. Only the heating system is discussed; however, much of the equipment described is also used in the cooling system.

CLEANING AND FILLING THE SYSTEM

Complete flushing and cleaning of the entire system is mandatory for efficient and trouble-free operation. No matter how carefully the installation is made, there are always traces of oils from the manufacture of the pipe, valves, and fittings; threading oil and chips; solder flux or pipe joint compound; welding flash and slag; and plain dust, dirt, and grime. These will result in the formation of scale and sludge which can coat and clog the heat exchanger passages, jam the valves, and cause a loss of efficiency of up to 50 percent in the first months of operation. The corrosion inhibitors of antifreeze solutions will also react with these contaminants, promoting sludge formation and depleting the inhibitors.

The first step is to flush the system, with the pump running, with clear water to attempt to collect the larger particles in the storage tank where they can be drained out. Also, a filter-strainer can be temporarily installed downstream of the pump to keep particles from re-entering the system. This can be done while the system is being checked for proper functioning of all valves and making sure that there are no leaks.

The system is then drained to remove any foreign material collected and refilled with a cleaning solution. Suggested cleaners are as follows (use only one):

- Trisodium phosphate - 1 pound for each 50 gallons in system
- Sodium carbonate - 1 pound for each 30 gallons in system
- Sodium hydroxide - 1 pound for each 50 gallons in system

It is desirable to bring the system up to operating temperature at this stage, if possible, and circulating through each of the flow paths, one at a time for maximum velocity, for a total of 48 hours. The system should then be drained completely and refilled with fresh water. A small amount of cleaner will adhere to the piping, giving an alkaline solution which is desirable to help prevent corrosion. For a plain water system, a pH reading between 7 and 8 is preferred, and a small amount of cleaner can be added, if necessary. If antifreeze is to be used, the system should be reflushed to prevent a reaction between the cleaner and the inhibitors in the antifreeze.

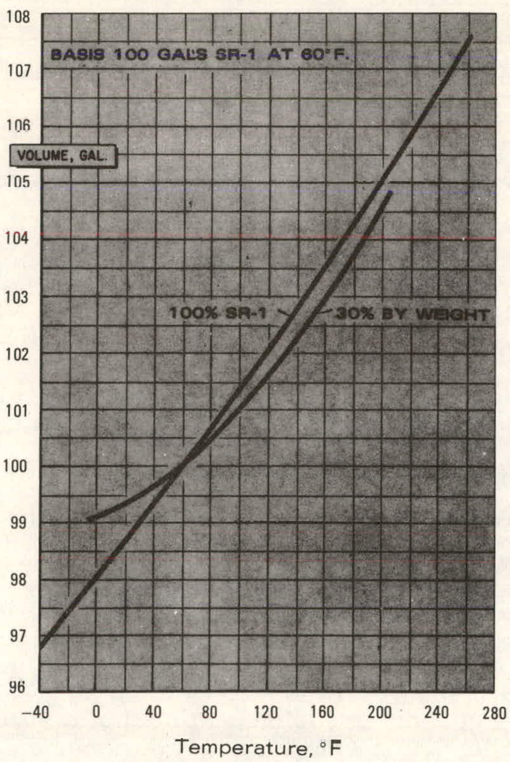
In addition, it is recommended that the final fill water have a low mineral content which otherwise causes scale formation and reacts with and depletes the inhibitor. Distilled or condensate water is recommended.

The most convenient method to either fill or drain this particular system is through the drain valves in the bottom of the storage tanks. While filling the system the vent valves in the expansion tanks and other high points must be open to allow the trapped air to escape.

LIQUID LEVEL

This is a closed system; therefore, it is critical that the system liquid volume and expansion tank pressure be maintained within the design limits for proper operation. For the initial filling, especially if the water is cold, a fill level to the bottom of the expansion tank or to the bottom of the sight gauge is sufficient if the air has been reasonably well vented from the rest of the system. When the circulating pump is turned on, additional air will be returned to the tank, but it is not necessary to add more liquid unless indicated by the storage tank low liquid level alarm.

Heating the fluid will drive out additional air that had been dissolved in the water and also will cause expansion of the fluid as indicated in the figure shown below.



The system is designed for approximately 1000 gallons total capacity containing approximately 30 percent SR-1. If 1000 gallons is the initial fill at ambient, it may be seen from the curve that it will drop to 990 gallons at 0°F which is the temperature at which it forms a slush. Raising the temperature to 150°F causes expansion to approximately 1027 gallons, and at the design operating temperature of 200°F, the liquid volume will be 1045 gallons. This expansion is into the expansion tanks, but even at this operating temperature the expansion tanks should be no more than one-third full.

Excess liquid may be drained directly from the expansion tanks by means of the garden-hose-type valves on the bottom of the tanks, or through the storage tank drain valves that were used to fill the system. Do not attempt to add liquid through the valves on the expansion tanks because they have an integral check valve that allows draining only.

An excess of fluid in the system should be avoided even if the operating pressure is correct, since the fluid contracts on cooling and the gas expands to fill the space and drops proportionately in absolute pressure. If too small an amount of gas is in the system, the pressure at the top of the solar collector panel will drop below ambient, which may draw air into the system, or could even cause physical damage from the vacuum.

NORMAL SYSTEM PRESSURES

When the system has been properly filled and pressurized, the design static (pump not running) system pressures are as shown in Table 4-1.

Table 4-1. Design Static System Pressure

Location	lb/in^2 Gage	Feet H_2O
Top of Collector Panel	1.7	4.0
Bottom of Collector Panel	7.7	17.8
Vehicle Floor	9.0	20.8
Center of Storage Tank	7.5	17.3
Air in Expansion Tank	5.0-6.0*	11.5-14.0

*The exact pressure will depend on the fluid temperature, expansion and therefore height in tank.

These pressures are established to prevent boiling and provide good air bleeding at the top of the solar energy collector panel, and yet prevent overpressurizing the panels when the pump is turned on. The static pressure at the vehicle floor level is included as a ready reference to use for calibrating the various pressure gauges for their actual physical location by dividing their height above the floor by 2.31 to determine the static head reduction to be applied.

The system pressure profile changes drastically when the pump is turned on, due to the head added by the pump and the friction losses in the pipes, valves, fittings, and heat exchangers. The only point in the system that remains constant is at the connection to the expansion tank, which in this case is the storage tank pressure. From there, the pressure drops in flowing to the pump, is then raised by the pump head for the flow rate of the system, and from there drops in each segment of the loop until it reaches the storage tank at the starting pressure. To calculate the total pressure at a specific point in the system, the altitude head gain or loss must also be taken into account along with the friction loss.

Table 4-2 shows the design total pressure at selected points in the system assuming a flow rate of 11 gpm in the loop analyzed.

If the collector flow control valve is less than full open or if part of the flow is going through the boiler, the flow rate and the total pressures will change from the above depending on the specific conditions.

TEMPERATURE CHECKS

For informational purposes, temperature readings may be taken from almost a hundred different locations. However, for normal system operation, the storage tank outlet temperatures, in and out of the solar collector, out of the boiler, and in and out of the air heating coils are expected to be the most important fluid temperatures to monitor. The air temperature rise across the heating coils will probably be the most useful of the air

Table 4-2. Design Total Operating System Pressure

Location	lb/in^2 Gage	Feet H_2O
Collector-Coll. Coil Loop		
Storage Tank at Floor Level	9.0	20.8
Pump Inlet	7.7	17.8
Head Added by Pump	+10.8	+25.0
After Pump	18.5	42.8
Friction Loss to Collector	- 3.6	- 8.4
Altitude Loss to Collector Bottom	- 1.3	- 2.9
Bottom of Collector	13.6	31.5
Altitude Loss to Collector Top	- 6.0	-13.8
Friction Loss through Collector	- 0.9	- 2.1
Top of Collector	6.7	15.6
Inlet to Collector Coil Heater	11.6	26.8
Boiler-Auxiliary Coil Loop		
Inlet to Boiler	16.7	38.6
Inlet to Auxiliary Coil Heater	15.1	34.8

temperature sensors. Of all of these, however, the fluid temperature out of the solar energy collector is of prime importance to ensure that the fluid temperature is at a useful level and yet not so hot that it may be in danger of boiling.

AIR BLEEDING

Theoretically, air trapped in the system will be carried along by the fluid velocity and be brought to the storage and expansion tanks. This is generally true, but there are always high points and corners that catch and hold air bubbles which can cause a noisy and inefficient system. For this reason, several manual air vent valves (the automatic type are not recommended since they are reported to have a tendency to leak) are installed at key locations of the system.

One is at each extreme end of each outlet manifold of the solar collector panels, and one is in the center of the topmost manifold where all the heated fluid is combined to return to the system. Another important place where air will collect is in the auxiliary boiler which has a built-in air trap. This is the normal location for connecting the line to the expansion tanks, but this is undesirable for this particular system. However, it is also the designated location for the boiler pressure relief valve which is installed and should be used as required to vent trapped air. Two other air vents are in the return lines to the top and bottom of the storage tanks just after valve V6 since there is a down run at reduced velocity which may not carry the air along to the tanks. The last of the air vents, one which should not be overlooked, is in a bypass line around Valve V1 and is partially hidden by the boiler. This vent is built into the valve so it looks different and therefore may not be noticed.

These air vent valves must be opened during the filling process to prevent pressure build-up and ensure a completely filled system. They should also be opened frequently during the cleaning and flushing operations to prevent flow blockage which may otherwise occur. Then again on initial warmup of the system the air which had been dissolved into the fluid will be driven out by the rise in temperature so large amounts of air will collect in the solar panels and/or above the boiler. Once the system has been thoroughly heated, liquid level and system pressure established, and sealed, the air bleeds need only to be opened on a periodic basis, governed by experience, to keep the fluid flowing properly.

DRAINING AND ADDING LIQUID

The indication of too much liquid in the expansion tanks is that the system pressure gets too high when it is at operating temperature. Liquid may be removed through the hose drain valve from either of the expansion tanks, or if more convenient, through the storage tank drain valve. Draining from the storage tank should also be done periodically to remove any sludge or sediment that has accumulated.

Too low a fluid level may allow air to be sucked into the pump which will cause cavitation; this is extremely damaging to the pump impeller. When the fluid is cold, contraction may lower the level to the top outlet which could cause this to happen. At all temperatures and conditions, the fluid level must be above the top tank outlets and below the one-third full level of the expansion tanks. Add fluid through the storage tank drain line. A check valve in the expansion tank drain valves will not allow fluid to be added directly to them.

EXPANSION TANK AIR PRESSURE ADJUSTMENT

The design static pressure at the top of the solar collector panel is 1.7 psig which translates to 7.5 psig at the center of the storage tank where the low-pressure alarm is located, and 9.0 psig at floor level where it may be read by the gauge located alongside the boiler. The tank pressure remains the same with the pump on or off, which is why the alarm was located there. However, the gauge near the floor ahead of the pump will read a lower pressure with the pump running, and therefore should be read with the system off to check system pressure.

Since the water level will change the pressure due to both static head and varying the air volume, the level should first be checked and corrected if necessary. Then, if the total static pressure is too high, air can easily be bled from the expansion tank by loosening the screw in the center of the valve handle of the expansion tank drain valve. This valve has a hollow stem that leads through a tube up to the center of the expansion tank and therefore into the air space above the liquid.

If air must be added, it may also be done through the same opening using a small air compressor, or through the storage drain valve. This could easily be done by using the installed compressor to pressurize the fresh water tank when it is empty and then connect the fresh water fill valve to the storage tank drain valve. Then bleed air in slowly while watching the pressure gauge to avoid over-pressurizing the system.

FLUID FLOW BALANCE VALVES

There are five balance valves, one in each major flow path loop, to allow establishing equal flow rates in each loop. There are also balance valves in the four lines leading to the top and bottom of both storage tanks. Balancing the system should be done in a normal operating mode at a stable operating temperature.

Setting the valves is most easily done by using the pressure differential meter connected to the pressure gauge fittings on each side of the valve. These connections should be made finger tight, which opens an integral check valve allowing the pressure drop across the valve to be measured on the readout meter. Then using the circuit setter calculator the gpm flow through the valve may be read.

It is suggested that the valves leading to the bottom of the storage tanks be balanced first with all the manual shut-off valves around the tanks in the full open position.

With the system flow divided between these two lines, the valves should be adjusted so that the two pressure drops are equal, and the two equal flow rates should add up to the flow rate measured on the other system flow meters. For maximum system flow, one of these valves should stay in the fully opened position and the other partially closed for balance. Note the final readings.

Next change valve V6 to divert the flow to the top of the storage tanks leaving the rest of the system in the same mode. Then set these balance valves to the same differential pressure readings as the previous valves to the bottom of the tanks.

Again, not changing the rest of the system, turn valve V4 to the bypass mode and set this balance valve (located between V3 and V4) so the system flow is again equal to what it was when flowing through the storage tanks.

The heating flow mode through the solar collector, the collector heating coil, and return has the largest calculated pressure drop through the system. This means that the balance valve in this loop (located at the outlet from the lower heat exchanger) should be left fully open. However, a reading should be taken across it to find the flow with the flow control valve (FC1) open all the way. This flow rate will be used to set the rest of the balance valves.

Closing the flow control valve (FC1) and opening valve V5 bypasses the collector, allowing the balance valve located near V5 to be adjusted to give the same system flow rate as it had going through the collector. Then turning valve V7 to its other position bypasses the collector heating coil so that the balance valve (located on the line from V7 back to the storage tanks) may be set to maintain the established flow rate.

SAFETY PRESSURE RELIEF VALVES

Two safety pressure relief valves are installed in this system. The one connected to the solar collector panel is set to start opening at 15 psi and the one on the auxiliary boiler opens at 30 psi. Due to other safety measures built into the boiler controls, it is very unlikely that this valve will open from boiler pressure, but it also has a very useful function for manually bleeding air trapped in the boiler.

If the solar panel relief valve opens, it is most likely due to insufficient circulation causing boiling of the fluid and having valves closed, preventing circulation to the storage tank. To correct this, increase the panel flow rate and make certain fluid is flowing through the storage tanks, or at least increase the opening of the safety bypass valves. If lack of circulation is due to system shutdown or power failure, the manual air vents at the top of the panels can be temporarily opened to allow the vapor to escape, but this may also allow air into the system.

Too high a system pressure can also cause this relief valve to open, indicating that there is too much liquid or air in the system. Correcting these will then allow proper system operation.

ALARM INDICATORS

Several alarm indicator lights are located at the top of the instrumentation console. These lights turn on when an abnormal fluid condition exists in the system. Some conditions may cause more than one indicator to light simultaneously.

The remaining balance valve is in the auxiliary heating mode loop and is located at the outlet from the upper air heating coil. Opening V10 and closing V5 (FC1 should already be closed) will divert the fluid through the boiler to the auxiliary coil and return, which is the required path for setting this balance valve for the design flow rate.

Now with all the balance valves properly set, the system flow rate through any one of the possible flow paths will be equal to the others. However, if two flow paths are used simultaneously, such as for second stage heating when both heating coils are being used, the flow is divided, which reduces the system friction loss in the piping and therefore increases the overall system flow.

SAFETY BYPASS VALVES

There are two bypass lines with shut-off valves in the system which should be left at least partially open at all times unless the system pressures are being carefully monitored. The first is the manual valve near V7 to relieve pressure in the solar collector panel to the storage tank in case other return paths are closed off. This cannot happen with the present heating-only system; however, when the air conditioning units are installed it could happen, so the line was installed.

The other bypass line is near V1, tying the pump inlet to the storage and expansion tanks. This is necessary to prevent a closed system with no expansion space when the system is in the storage bypass mode which could cause a buildup of large pressure differences between the tank and system pressures.

A temperature sensor located at the top of the solar collector panel lights its indicator when the collector liquid temperature rises above the set point. The set point is adjustable between 160°F and 260°F and has a fixed differential of 1°F. The design setting is 210°F to indicate that it is above the normal operating temperature; if allowed to continue rising, the liquid will start to boil.

The two pressure alarms are designed to indicate when critical pressures exceed the normal operating conditions. The one in the solar collector line makes on a pressure rise to the set point of 9 psi plus the differential setting of 5 psi, or 14 psi, which is above the normal operating pressure but below the safety relief valve setting. Due to the necessary high differential setting, it may require that the circulating pump be temporarily shut down to reset the alarm.

The storage tank alarm lights when the system pressure drops below the set point of 6 psi which is the minimum pressure to ensure fluid at the top of the solar collector panel. This sensor should have the differential set at the minimum of 1.0 psi so that it will reset when the design pressure is reached.

The liquid level sensors trigger the alarm when there is insufficient liquid for efficient operation. A low level in the collector outlet manifold will prevent proper circulation of at least some of the heated fluid. This condition may be caused by trapped air, too low a system pressure, a boiling situation, or a combination of these which indicates the need to analyze the overall system for the correction required.

The liquid level sensor near the top of the storage tank indicates when there is danger of drawing air into the pump. The sensor is located several inches below the desired level for the coldest possible liquid temperature. If this alarm is lit, there is probably a leak in the system which must be corrected and liquid added to resume proper operation.

There is provision for a gas leak alarm that could be used with a suitable sensor located near the auxiliary boiler.

APPENDIX A

CALIBRATION AND HEATING TEST REPORT OF TRANSPORTABLE SOLAR HEATING/COOLING SYSTEM LABORATORY

SUMMARY

The main objective of Honeywell's transportable solar laboratory is to provide a solar heating and cooling system which can be evaluated under various types of climatic conditions. Evaluating the system depends on the calibration and certification of the instrumentation. At the National Bureau of Standards, Washington, D.C., several key thermocouples, six flowmeters, and two pyranometers were calibrated. Significant errors were observed in everything except the thermocouples; correction of these errors resulted in substantial improvements in measured collector efficiency.

Initial heating testing in the Washington, D.C. area revealed efficiencies in the order of 70 percent. Even on cloudy or overcast days, the solar collection was adequate enough to heat the two trailers. System operation proved the system design concept was feasible.

SYSTEM DESCRIPTION

Honeywell's transportable solar laboratory is built to provide a solar heating and cooling system capable of being evaluated from comparative performance in all types of climatic conditions. Since solar energy is being used to conserve energy, energy conservation was kept in mind throughout the design. Solar panels were designed to maximize the quantity of solar energy absorbed; panels and trailer outside surfaces were insulated with high-quality insulation; doors and windows were kept to a minimum; and an air economizer cooling system was used for cooling in mild weather.

The total system consists of two separate units to allow a reasonably sized heating and cooling load and yet be able to transport it to different locations. One unit is the display van which shows information relative to solar energy and provides conference room space. The other is the equipment van which contains all of the mechanical equipment for handling the performance data. In operation the two units are coupled together, but each has a different heat load and can be analyzed separately.

The solar energy collection and distribution system basically consists of a flat plate collector containing a water-glycol mixture, pump, storage tanks, heat exchangers to warm the air, two hot water air conditioning systems, a domestic water heater, and an auxiliary boiler to provide system heat at the times solar energy is not available. The data system measures the solar and outdoor conditions and monitors the status of the heating/cooling system temperatures, flow rates, and mode changes. The processing subsystem records this data and provides communication capability with a central site computer.

CALIBRATION

To evaluate the system performance adequately, it must be assumed that the instrumentation is providing accurate values. To assure this, several thermocouples, all the liquid flowmeters, and the two pyranometers were calibrated by the National Bureau of Standards (NBS). Dr. James Hill was the main contact at NBS and handled the calibration and coordination of the instrument calibration.

Pyranometers (Solar Radiation)

The National Bureau of Standards normally does not do pyranometer calibration. However, since the solar laboratory was at NBS and Dr. Hill was overseeing the calibration project, the calibration was performed at NBS. A visit was made to the Weather Bureau where calibration of pyranometers is normally conducted. The man in charge of pyranometer calibration completely explained their methods and loaned NBS an Epply precision pyranometer for a calibration standard.

The calibration was preformed by placing the Epply and one of Honeywell's Kipp-Zonen pyranometers side by side on the roof of one of the NBS buildings. Readings were taken on clear days over an extended period of time. The two pyranometers were checked separately to prevent shutting down the solar system.

The resulting pyranometer output was determined and plotted against the standard Epply pyranometer. The sensitivity of the collector pyranometer was determined to be 8.11 mv per Langley as opposed to the 7.77 mv per Langley specified by Kipp-Zonen (see Figure A-1). The sensitivity of the horizontal pyranometer was determined to be 7.93 mv per Langley as opposed to 7.5 mv per Langley specified by Kipp-Zonen (see Figure A-2). The higher sensitivities of the pyranometers will result in a significant increase in collector efficiency.

When the solar laboratory was returned to Minneapolis, the pyranometers were attached to a Leitz Dividing Head (Figure A-3) in order to accurately measure the incident angles. A digital voltmeter was used to record the variations in output. Figure A-4 shows a polar coordinate plot of the collector pyranometer output as a function of incident angle, while Figure A-5 shows the percent deviation from the cosine of the incident angles. Note that the deviation is not symmetrical about the direct incident radiation. Figure A-6 is the polar coordinate plot of the horizontal pyranometer output as a function of incident angles. The percent deviation from the cosine of the incident angle is shown in Figure A-7 for the horizontal pyranometer. Again the deviation curve is not symmetrical; however, this unit has a smoother curve than the collector pyranometer.

Thermocouples CU/C (Temperature)

Five thermocouples, removed from the fluid system, and one spare thermocouple were calibrated at the NBS laboratory in an ice bath and a stirred water bath. These thermocouples were evaluated using a readout system which included the immersion thermocouple (Omega), a thermocouple connector, copper-constantan lead wire, and the Omega digital thermocouple readout. Results of this calibration from 32°F to 196.2°F showed the thermocouple readout system to be accurate within one degree (see Figure A-8 which is a NBS report of the thermocouple calibration). The thermocouples calibrated were from the collector inlet and outlet (T-1 and T-2), collector coil inlet and outlet (3 and 4), auxiliary coil inlet (10), and one spare thermocouple.

Later NBS calibrated the thermocouples more accurately using a precision digital voltmeter to compare the thermocouples to a standard platinum resistance thermometer. The collector input and output thermocouples (T-1 and T-2) plus three spare thermocouples were calibrated this way.

These test results showed the collector thermocouples to be within 0.5°F. Two of the thermocouples were within 0.045°F proving that pairs can be matched for more accuracy. Figure A-9 is a report of this calibration.

When the laboratory was returned to Minneapolis, several critical thermocouples were removed. These thermocouples along with several new thermocouples were calibrated against a precision platinum resistance thermometer at Honeywell. From these test results pairs were selected for inlet and outlet temperature measurements at critical locations in the collector fluid system. The results are shown in Table A-1.

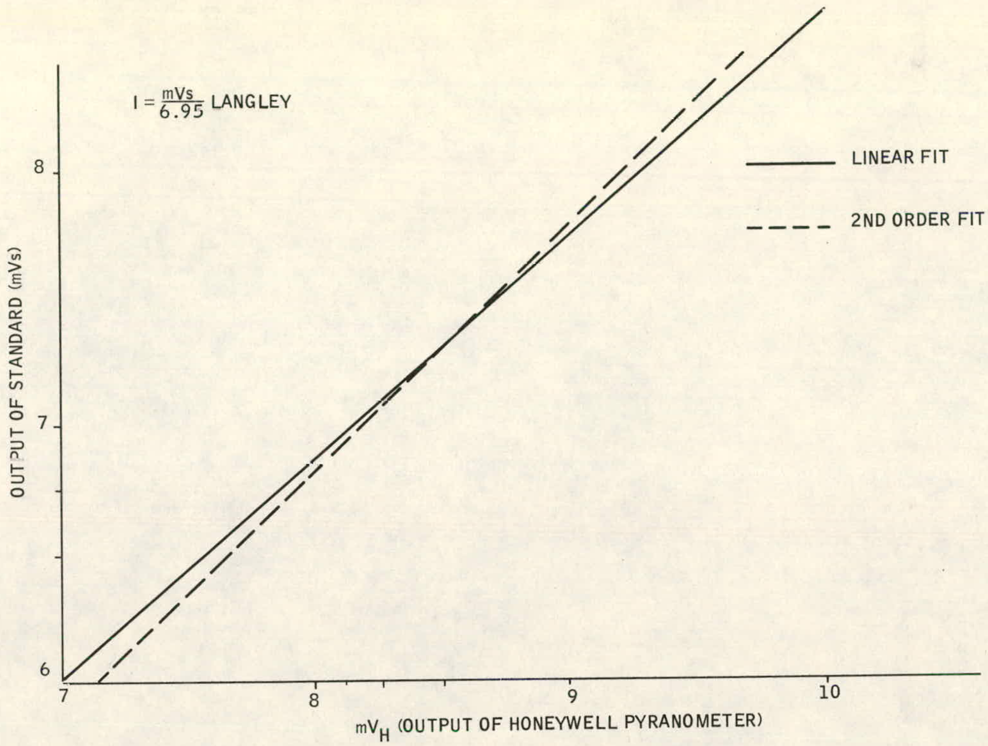


Figure A-1. Calibration - Collector Pyranometer, March 1974

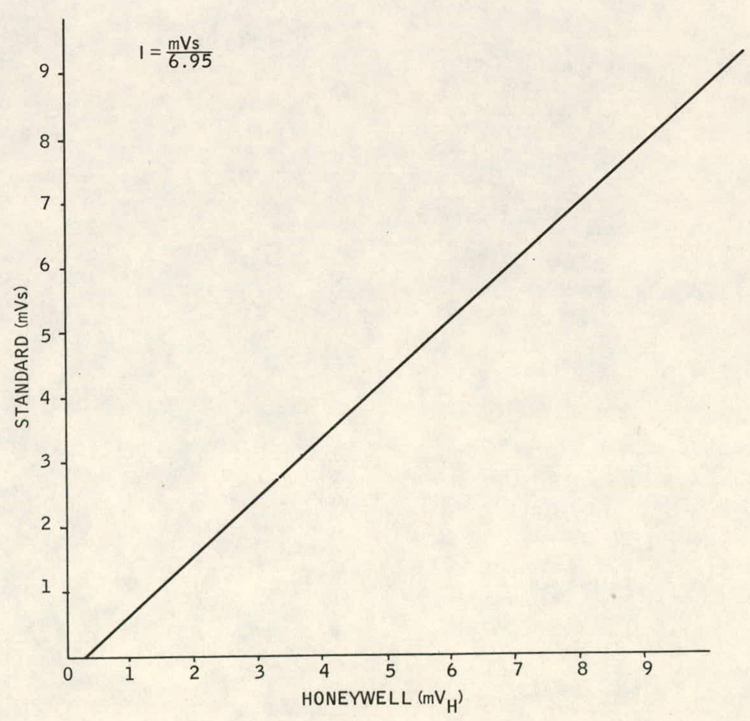


Figure A-2. Calibration - Horizontal Pyranometer, March 1974

Figure A-3. Leitz Dividing Head

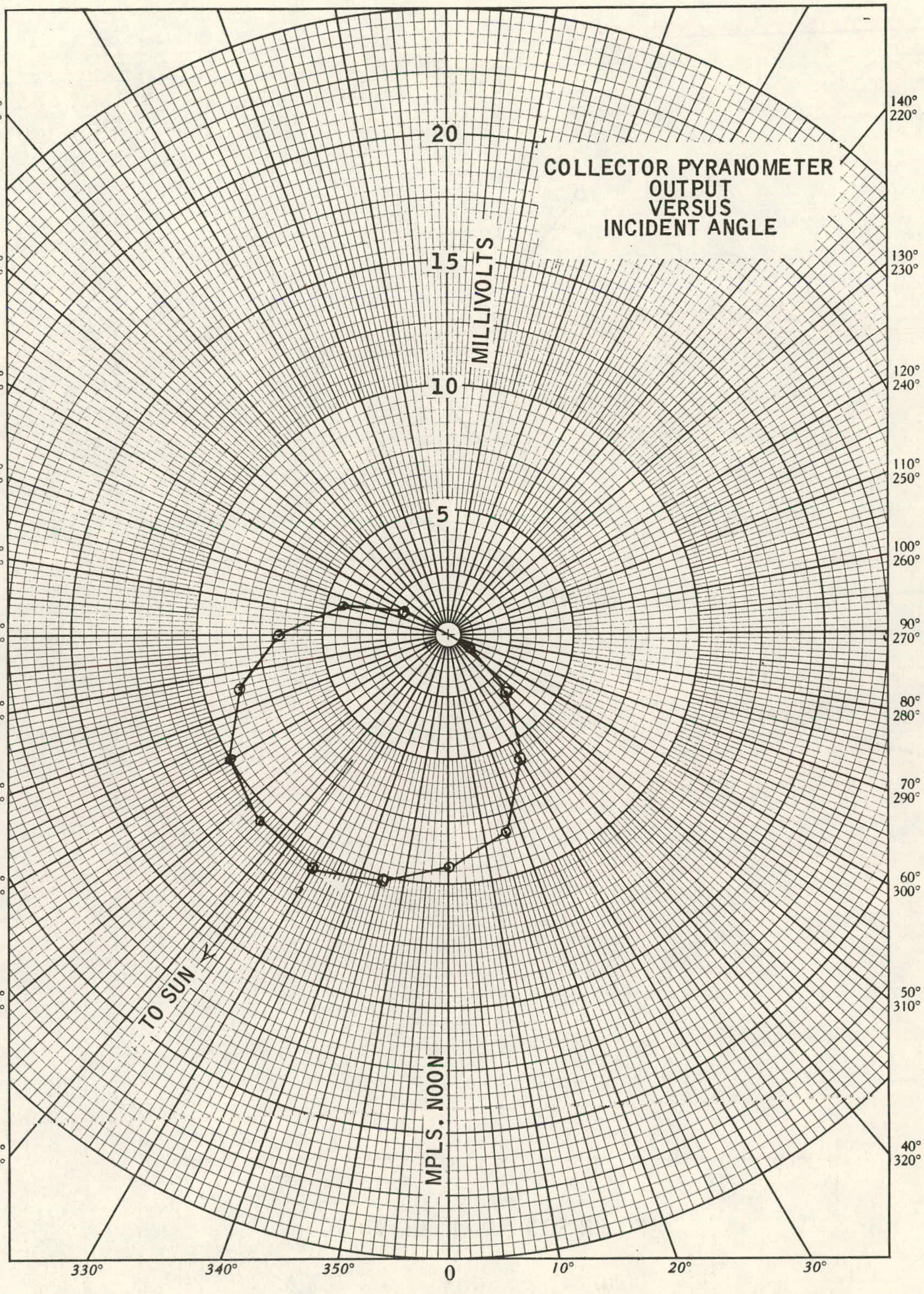


Figure A-4. Collector Pyranometer Output versus Incident Angle

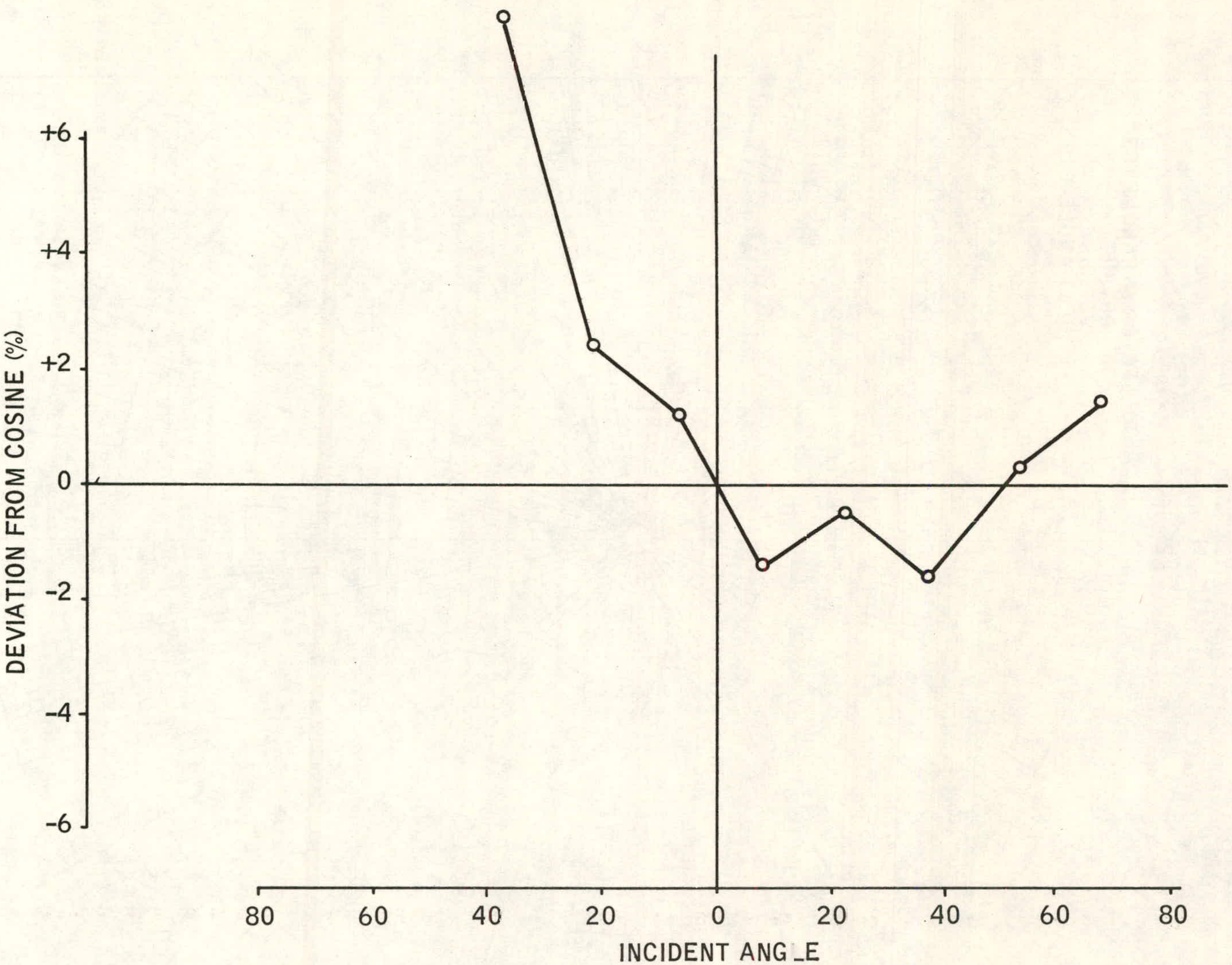


Figure A-5. Collector Pyranometer Cosine Deviation

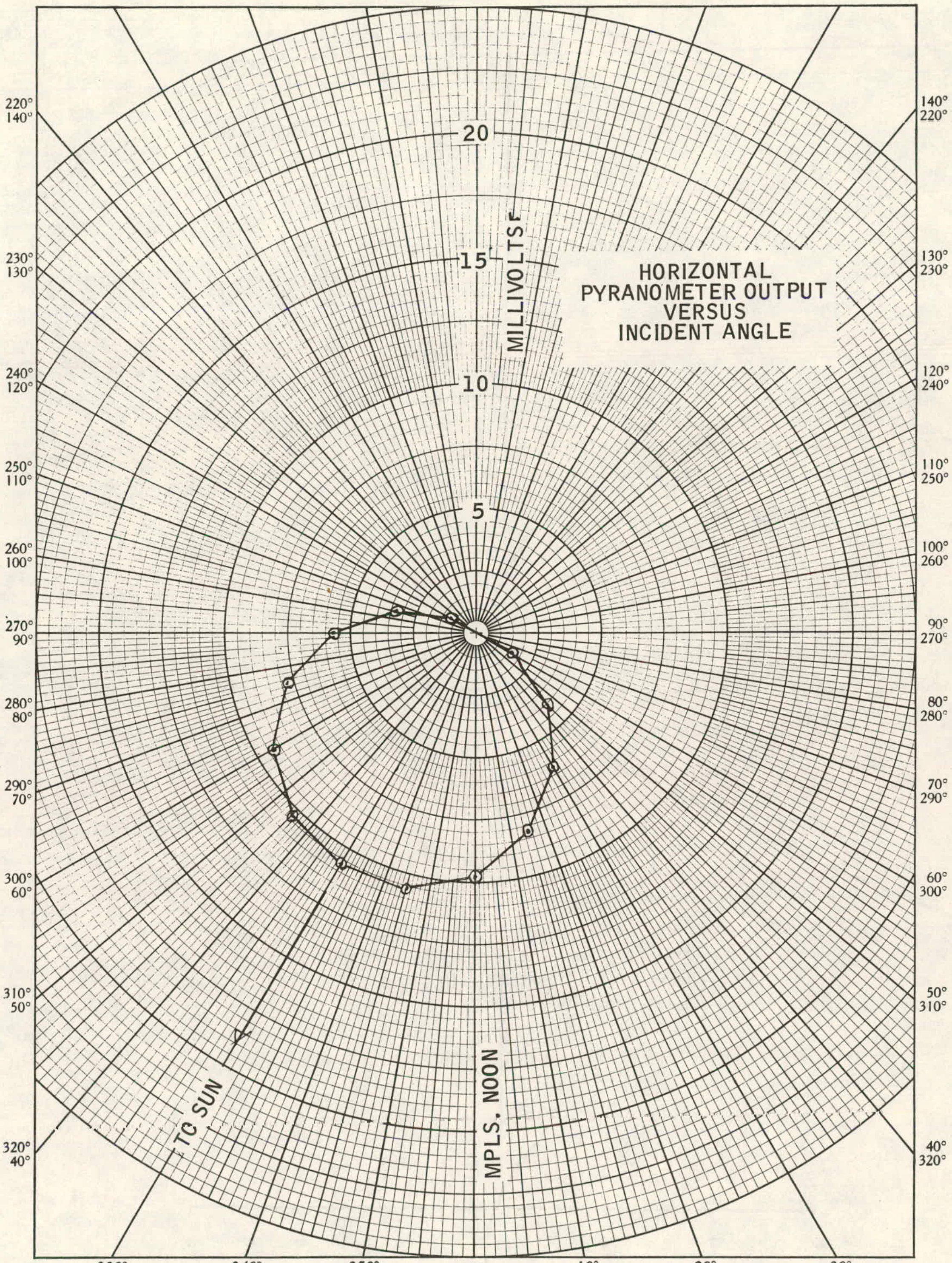


Figure A-6. Horizontal Pyranometer Output versus Incident Angle

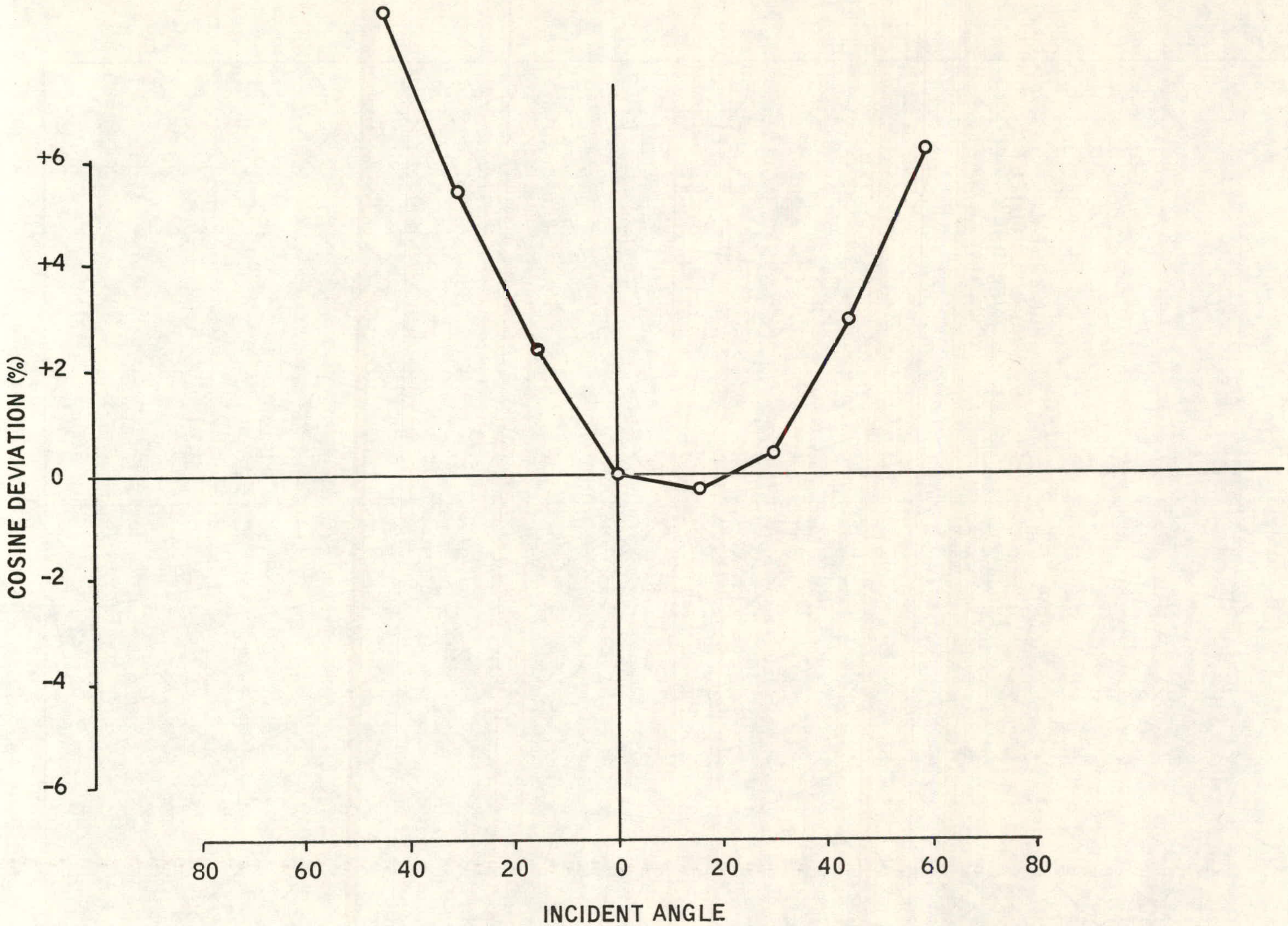


Figure A-7. Horizontal Pyranometer Cosine Deviation

Date: March 22, 1974

To: Mr. James E. Hill
Section 462.01

From: George W. Burns GWB
Section 221.11

Subject: Calibration of sheathed Type T thermocouples

We received seven 1/8 inch o.d. stainless steel sheathed Type T thermocouples from you for calibrations. They were tagged "spare", 3, 4, 10, 11, T-1 and T-2. It was necessary to straighten the sheaths of thermocouples 3, 4, 11, T-1 and T-2 prior to testing. The thermocouples were then calibrated by intercomparison with a standard platinum resistance thermometer in a stirred water bath at about 77, 122, 167 and 196 °F. The thermocouples were also calibrated at 32 °F in an ice bath. During testing, they were immersed approximately 9 inches into the baths. Electrical connections to the thermocouples were made by using the Type T thermocouple extension wires, terminal strip and mechanical connectors that were provided. The Omega Model DS-500 thermocouple indicator was used to read the output of the thermocouples. These components were maintained at room temperature (73 ± 2 °F) during the calibrations.

Thermocouple 11 was found to be defective during preliminary measurements. A check of the electrical continuity of this thermocouple indicated that there was a break in the copper leg. For each of the other thermocouples, corresponding values of the bath temperature and the observed reading of the thermocouple indicator are given in the following table.

Bath Temperature	Reading of Thermocouple Indicator, Degrees F					
Degrees F	"Spare"	3	4	10	T-1	T-2
32.0	31	31	31	31	31	31
77.2	76	76	76	76	76	76
122.0	121 or* 122	121 or 122	121 or 122	121 or 122	121 or 122	121 or 122
167.0	166	166	166 or 167	166	166	166
196.2	196	197	197	197	196	197

*Reading oscillated between these two values

The uncertainty in the value of the bath temperature given in the above table is estimated not to exceed 0.1 °F.

Figure A-8. Letter re Calibration of Sheathed Type T Thermocouples

Date: April 19, 1974

Reply to Attn of: GWB:221.11

Subject: Calibration of sheathed Type T thermocouples

To: Mr. James E. Hill
Division 462, Section 01

Test results for the five 1/8 inch o.d. stainless steel sheathed Type T thermocouples that you submitted on April 9 are presented below. Three of the thermocouples were tagged "spare", T1 and T2. We tagged the two unmarked thermocouples A and B. The thermocouples were calibrated by intercomparison with a standard platinum resistance thermometer in a stirred water bath at about 74.5, 122, 167 and 196 °F. The thermocouples were also calibrated at 32 °F in an ice bath. They were immersed approximately 9 inches into the baths during testing. Electrical connections to the thermocouples were made with the Type T thermocouple extension wires and connectors that you furnished. The extension wires were connected to copper lead wires and the connections were maintained at 32 °F in an ice bath during testing. The voltage measurements were made with a precision digital voltmeter having a resolution of 0.1 μV and a limit of error not exceeding ±(0.002 percent of voltage measured +0.5 μV).

Corresponding values of the bath temperature and the emf of the thermocouples are given in the following table. For convenience, values of emf from the ASTM reference table for Type T thermocouples (ASTM Standard E230-72) at the indicated bath temperatures are also given.

Bath Temperature Degrees F	ASTM Table emf, mV	Thermocouple emf, mV				
		T1	T2	"Spare"	A	B
32.0	0.000	0.000	-0.001	0.003	0.002	-0.001
74.5	0.936	0.937	0.937	0.937	0.936	0.937
122.0	2.035	2.036	2.041	2.037	2.037	2.040
167.0	3.131	3.130	3.139	3.134	3.135	3.140
196.0	3.864	3.859	3.872	3.866	3.866	3.871

The uncertainty in the values of bath temperature given in the above table is estimated not to exceed 0.1 °F. You will note that the maximum spread in the values of emf for these five thermocouples is 0.013 mV (equivalent to about 0.5 °F). In addition, the results for thermocouple "spare" agree with those obtained in the previous calibration (included with my memorandum of March 22, 1974) to within the equivalent of 0.1 °F.

George W. Burns

GEORGE W. BURNS
Electrical Engineer
Temperature Section
HEAT DIVISION

Figure A-9. Letter re Calibration of Sheathed Type T Thermocouples

Table A-1. Thermocouple Calibration

Reading	Temperature				
	32°F MV	100°F MV	150°F MV	200°F MV	32°F MV
1	0.005	1.529	2.725	3.980	0.007
2	0.010	1.529	2.720	3.973	0.010
3	0.007	1.528	2.722	3.978	0.0065
4	0.006	1.529	2.725	3.981	0.0065
5	0.01	1.527	2.720	3.973	0.0094
6	0.007	1.529	2.726	3.981	0.0062
7	0.010	1.527	2.720	3.974	0.009
8	0.008	1.529	2.723	3.978	0.007
9	0.007	1.529	2.724	3.980	0.006
10	0.009	1.528	2.719	3.974	0.009
11	0.008	1.529	2.723	3.981	0.007
12	0.006	1.529	2.724	3.978	0.006
13	0.009	1.528	2.717	3.970	0.01
14	0.006	1.529	2.722	3.978	0.0068
15	0.006	1.530	2.726	3.987	0.006
16	0.006	1.530	2.726	3.986	0.006
17	0.007	1.528	2.721	3.988	0.0075
18	0.01	1.527	2.718	3.964	0.01
19	0.007	1.525	2.709	3.967	0.0068

Several other thermocouples were checked at their locations. These thermocouples were removed and immersed in an ice bath and then a boiling water bath. Output readings were made using the thermocouple readout on the Data Logger. Because of inaccessibility, some thermocouples were calibrated by comparison only (see Table A-2).

The reference junctions in the Omega digital temperature readout and the Kaye data logger were calibrated at Honeywell and found to be within ±1°F.

Flowmeters (System Fluid)

There are three visual flowmeters and three electric output flowmeters in the fluid system. They are set up in sets of one visual and one electric. The flowmeters are located directly after the main system pump (total flow), at the inlet to the solar collector (collector flow), and at the inlet to the boiler (auxiliary flow).

The visual flowmeters were calibrated at NBS using a precision flowmeter in the system. A pipe section was removed after the main pump and a precision turbine type flowmeter inserted in the system plumbing. The system flow paths were set so each flowmeter could be calibrated. The flow was varied from approximately 2-1/2 to 8 gallons per minute. Several points were taken to plot calibration curves as shown in Figures A-10 through A-12.

Table A-2. Trailer Thermocouple Checkout

Location	Ice Bath	Boiling H2O
Storage Inlet Top	31.6	209.8
Storage Inlet Bottom	31.6	209.8
Storage Outlet Top	31.5	
Storage Outlet Center	31.4	
Storage Outlet Bottom	31.7	209.8
Storage Tank Top-Front	31.8	
Storage Tank Bottom	31.8	209.7
Storage Tank Top-Rear - TSI	31.6	
Auxiliary Boiler Inlet	31.5	209.8
Auxiliary Boiler Outlet	31.6	
TR Sensor (V4 Inlet)	31.5	
Pump Outlet	31.9	209.8
Total Flowmeter	31.7	
Domestic Cold Water	31.8	210.4
Domestic Hot Water	31.8	210.1
Standard	32.0	209.66
Comparison Methods for Thermocouple Which Were Not Removable		
Storage Tank Top-Rear (TSI)	80.7	Allow for some stratification
Storage Tank Top-Rear	80.6	
Storage Tank Center-Rear	80.1	
Storage Tank Bottom-Rear	79.7	

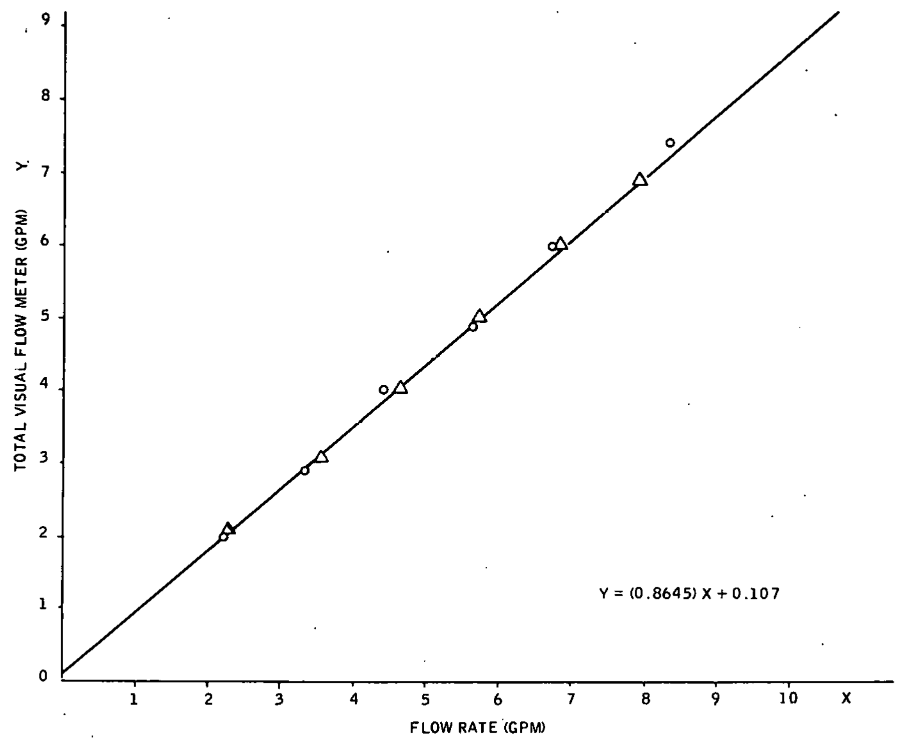


Figure A-10. Calibration - Total Flowmeter, March 1974

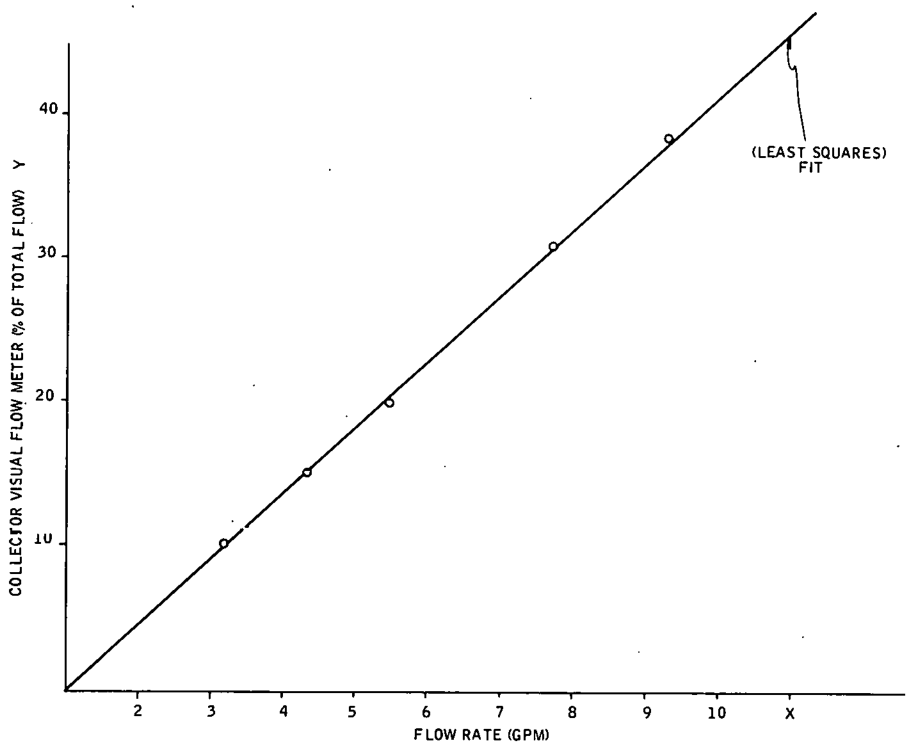


Figure A-11. Calibration - Collector Flowmeter, March 1974

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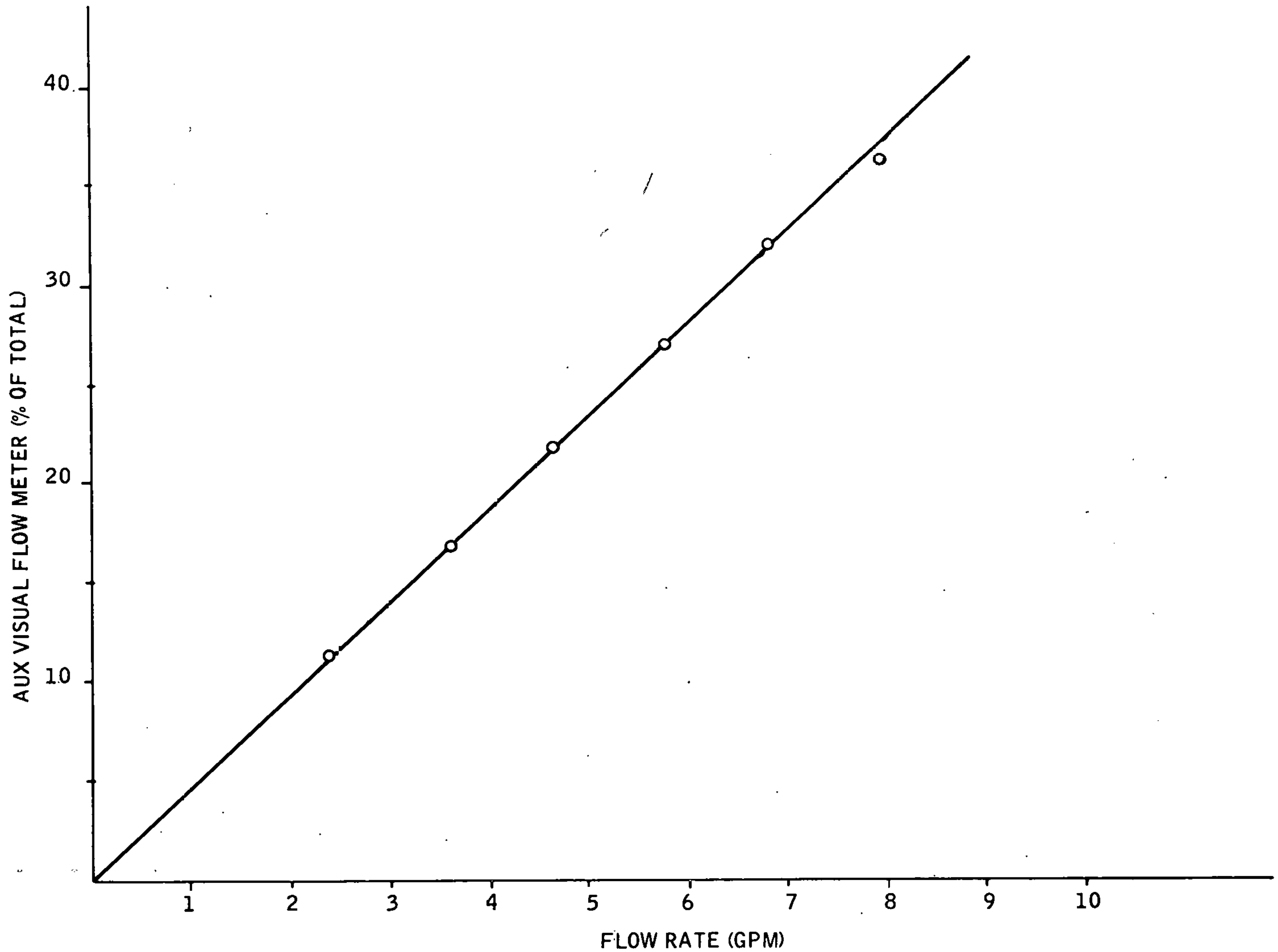


Figure A-12. Calibration - Auxiliary Flowmeter, March 1974

According to this calibration, the worst case was the total flowmeter where errors of 15 percent were measured. According to NBS, the flow was erratic and it was difficult to take accurate readings without a visual averaging technique.

The automatic flowmeters were not calibrated at NBS because the pulse-to-analog converter was saturating at about two gallons per minute.

When the laboratory was returned to Minneapolis, the flowmeters were again calibrated. This time a pipe was run from the collector inlet line to a 55-gallon drum. The drum was precalibrated in gallons. The system fluid was then pumped out of the system into the drum and timed. From this the gallon-per-minute flow was calculated. The flow was varied from approximately two to eight gallons per minute in two-gallon-per-minute steps. Readings were taken simultaneously on the visual and automatic flowmeters. The flow conditions were very steady and the errors in the visual flowmeters were found to be less than 2 percent. Plots of the visual flowmeters are shown in Figures A-13 and A-14. The auxiliary flowmeters were not calibrated at this time because it was difficult to extract the fluid after the flowmeters. They will have to be calibrated by comparison with the total flowmeter at a later date.

The output voltage from the pulse-to-analog converter was adjusted to produce approximately one millivolt per gallon per minute. Calibration curves are plotted in Figures A-15 and A-16. An approximate flow calibration formula is also shown as extracted from the curves.

Flowmeters (Cooling Tower)

There are no true flowmeters in the cooling tower water system but adjustable flow control valves (Bell and Gosset Circuit Setters) are provided in each air conditioner condenser circuit. Both the Arkla and Rankine flow paths were calibrated. Again a precalibrated 55-gallon drum was used and the flow was timed with a stop watch. Figure A-17 gives plots of the flows in gallons per minute.

Flowmeters (Air)

To accurately measure the air flow in the distribution ducts, an air velocity probe was purchased from Thermo-Systems Inc. It is essentially a hot wire anemometer type which produces a voltage proportional to air velocity in feet per second. The TSI calibration from 0 to 20 feet per second is shown in Figure A-18.

SYSTEM PERFORMANCE AND OPERATION

National Bureau of Standards - Gaithersburg, Md.

The mobile home arrived at NBS on 7 March and the equipment van arrived on 8 March. The entire system was set up and operational on 8 March. Since it was a weekend, no propane was available so the system was operated strictly off solar energy. Operation was excellent (no propane was purchased) so operation continued on solar energy. All equipment operated satisfactorily with only a few minor difficulties. Following are some operational comments.

Controls --

a) Flow control valve has insufficient range (140°F maximum) thus allowing a storage heat loss until the storage temperature returns to approximately 140°F.

b) Stratification in storage tanks is minor -- outlets at top, middle, and bottom are questionable.

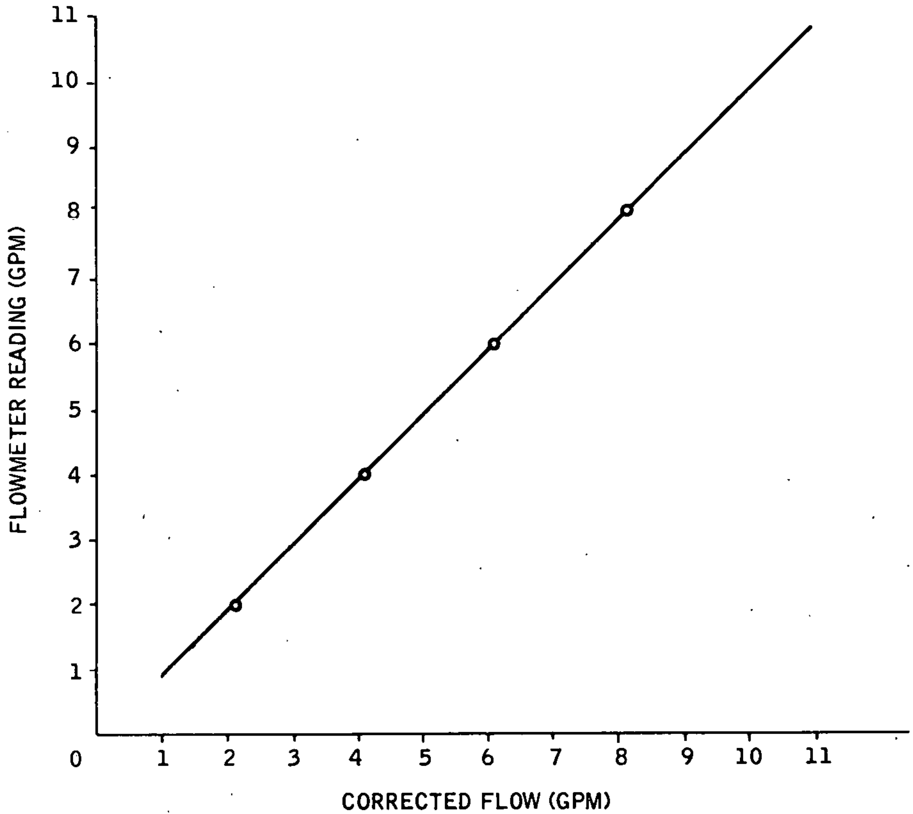


Figure A-13. Total Flowmeter Correction Calibration

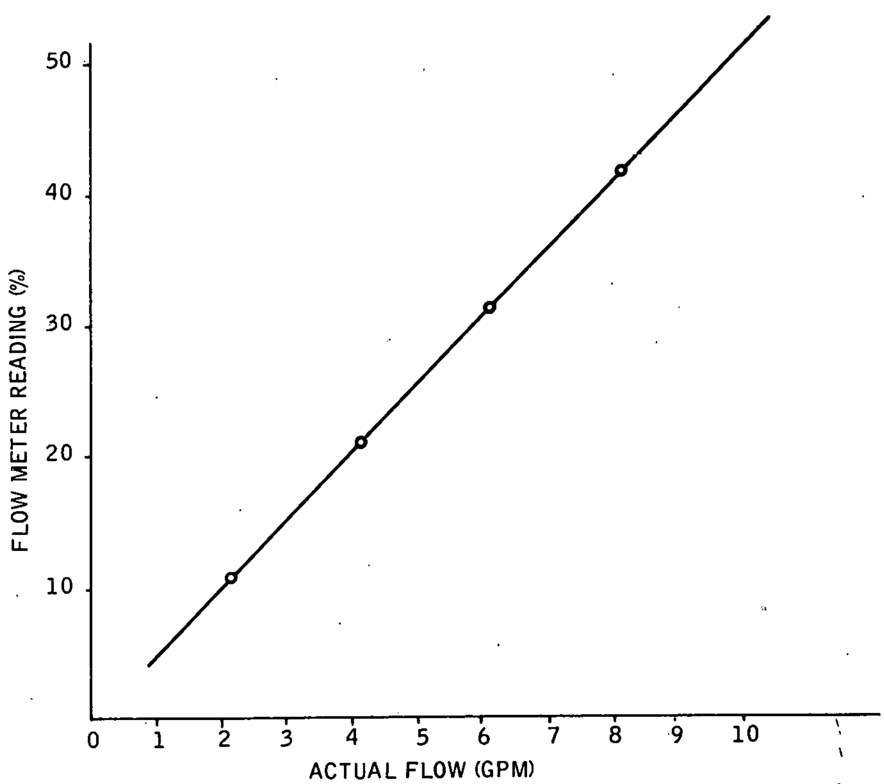


Figure A-14. Collector Flowmeter Calibration

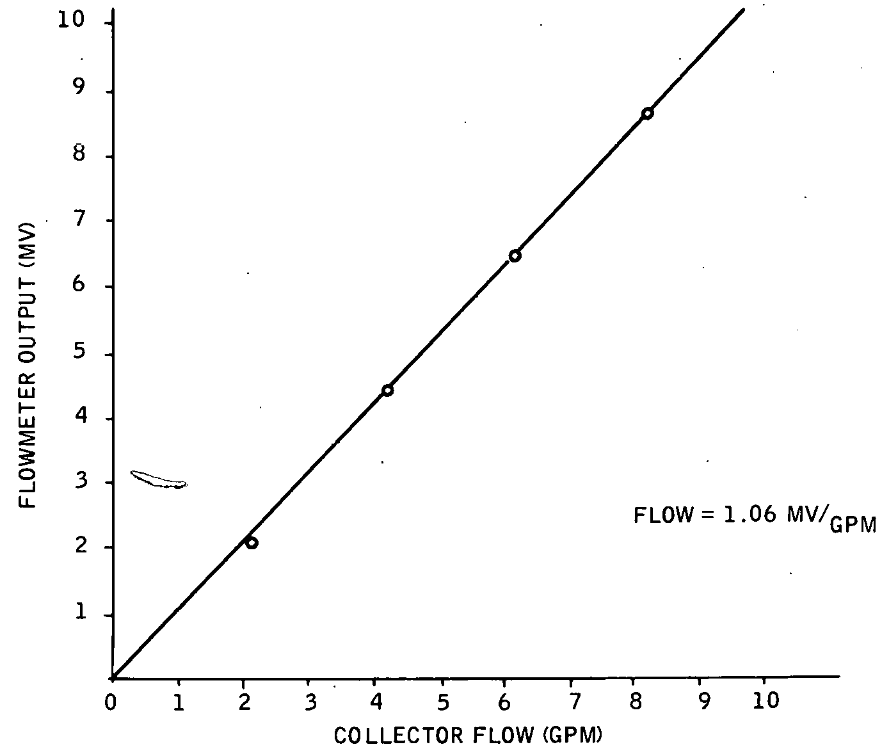


Figure A-15. Electric Collector Flowmeter Calibration

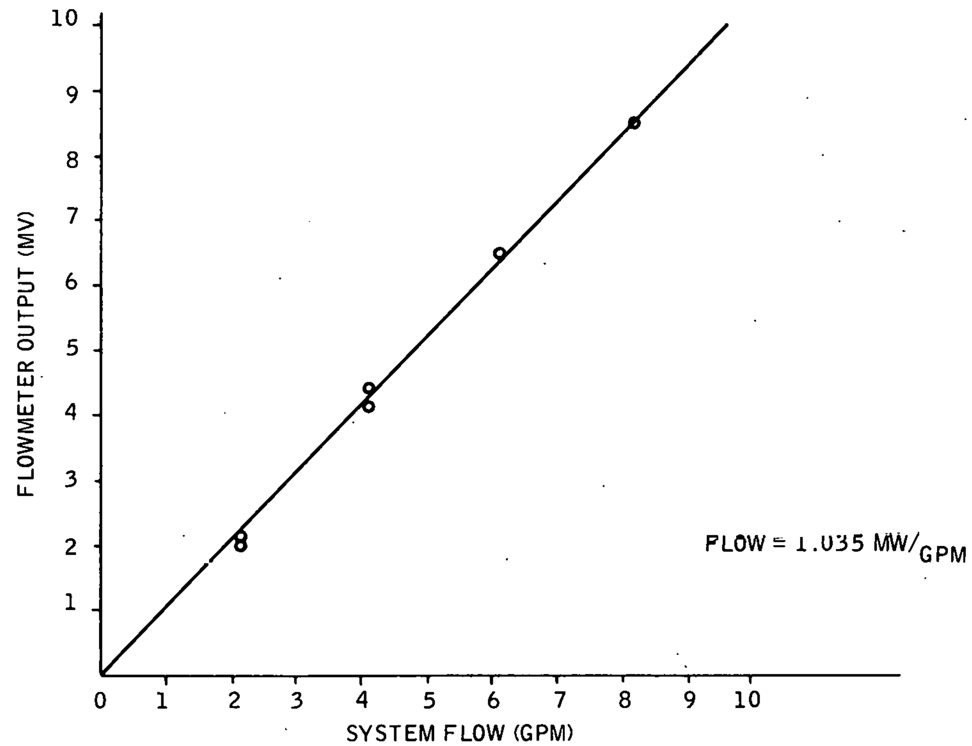


Figure A-16. Electric Total Flowmeter Calibration

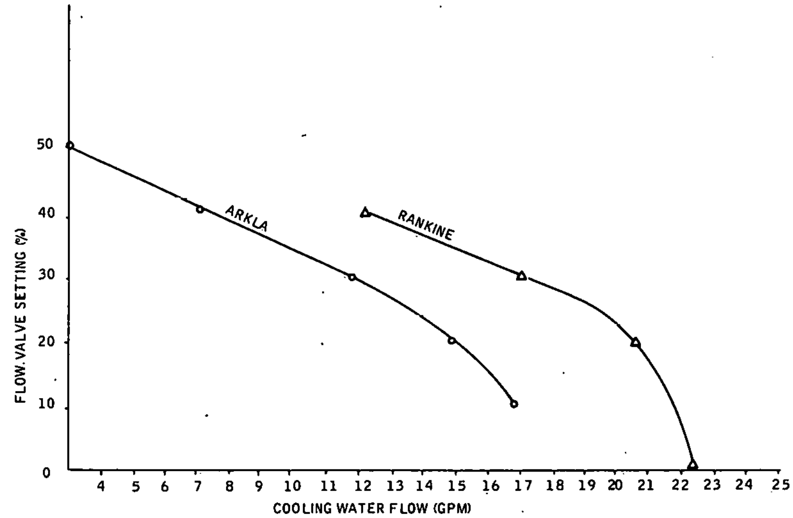


Figure A-17. Cooling Tower Water Flow

FLOW CALIBRATION DATA SHEET

W.O.# 94403 CUSTOMER Honeywell

TSI MODEL 1610-12 SERIAL 386 T.C. PROBE ________ SERIAL ______

FLOW RANGE: 0 TO 20 FPS IN AIR AT 14.2 PSIA AND 85 °F

FLOW CALIBRATION DATA

FLOW POINTS	OUTPUT	LINEAR OUTPUT
.0	.540	.
.1	.563	.
.2	.594	
.3	.616	
.4	.638	
.5	.660	
.6	.677	
.8	.710	
1.	.738	
1.2	.763	
1.5	.799	
2.	.845	
2.5	.874	
3.	.923	
4.	.989	
5.	1.061	
6.	1.112	
8.	1.186	
10.	1.258	
12.	1.329	
15.	1.430	
20.	1.565	

Refer to Instruction Manual to use the Linear Output data for setting up linearizer adjustments. The Linearizer Function Board for this data is 1117-______ or 1055- ________. End points have been set up at TSI on ______________.

Required Control Resistance:

________ Ω TSI 1304

________ Ω TSI 1305

________ Approx. Overheat Ratio

________ TSI 1050 or 1056 Setting

✓ Temp. Comp. System

The cable used for this calibration was AS SUPPLIED length __________ with a resistance of ______ Ω.

A TSI Model _________ probe support was used with an internal resistance of ________ Ω.

Special Notes:

Signed M. Johnson Date 7/19/74

Figure A-18. Flow Calibration Data Sheet

Collectors --

a) Temperature rise in collector ranges from approximately 40°F at 8 gpm to 100°F at 2 gpm.

b) Maximum collector energy has been 525,000 Btu under less than optimum conditions. One thousand gallons of fluid in storage were raised from 80°F to 150°F in one day.

c) Tedlar tends to sag under high-temperature conditions, but no shrinkage is noticeable.

d) The upper panels boiled on occasion of low flow testing. Maximum plate temperature reached was 300°F. Boiling of panels causes vapor lock and thus eliminates one set of panels to result in a lowering of overall efficiency.

Water (Fluid) Distribution System --

a) The flow rate is insufficient. Definitely need higher flow rate or higher system pressure to prevent panel boiling.

b) Need balance valves in collector quadrants to control flow distribution. This may help prevent boiling problem.

Air Distribution --

a) Excellent performance -- maintains stable and comfortable room conditions very well.

Instrumentation --

a) Need more accurate temperature measurement in critical locations. Consider a thermopile or matching of thermocouples in sets.

b) Automatic flowmeters not working.

c) Kipp-Zonen pyranometers are marginal according to NBS. They suggest change to Epply precision units.

d) Weather station working well.

e) System needs watt-hour totalizers. Current readings taken show drain of approximately 32 amps.

Much of the time at NBS was spent calibrating and checking various conditions for operation. Figure A-19 shows the curves of a typical day. Note that a partly cloudy day produces erratic results in both solar energy insolation and collector efficiency. The laboratory was at Gaithersburg, Maryland, from 7 March to 10 April.

Inter-Technology Corporation School - Warrenton, Virginia

The laboratory was moved to Warrenton, Virginia, and was operational on 12 April. The purpose of the move was to compare operating efficiencies with the ITC school system and to calibrate their pyranometers. The portable laboratory is to be used as a reference between the various school projects sponsored by NSF (National Science Foundation). Operation can only be compared on clear days to obtain fairly consistent readings. Figure A-20 shows a typical plot of solar insolation and efficiencies on a clear day.

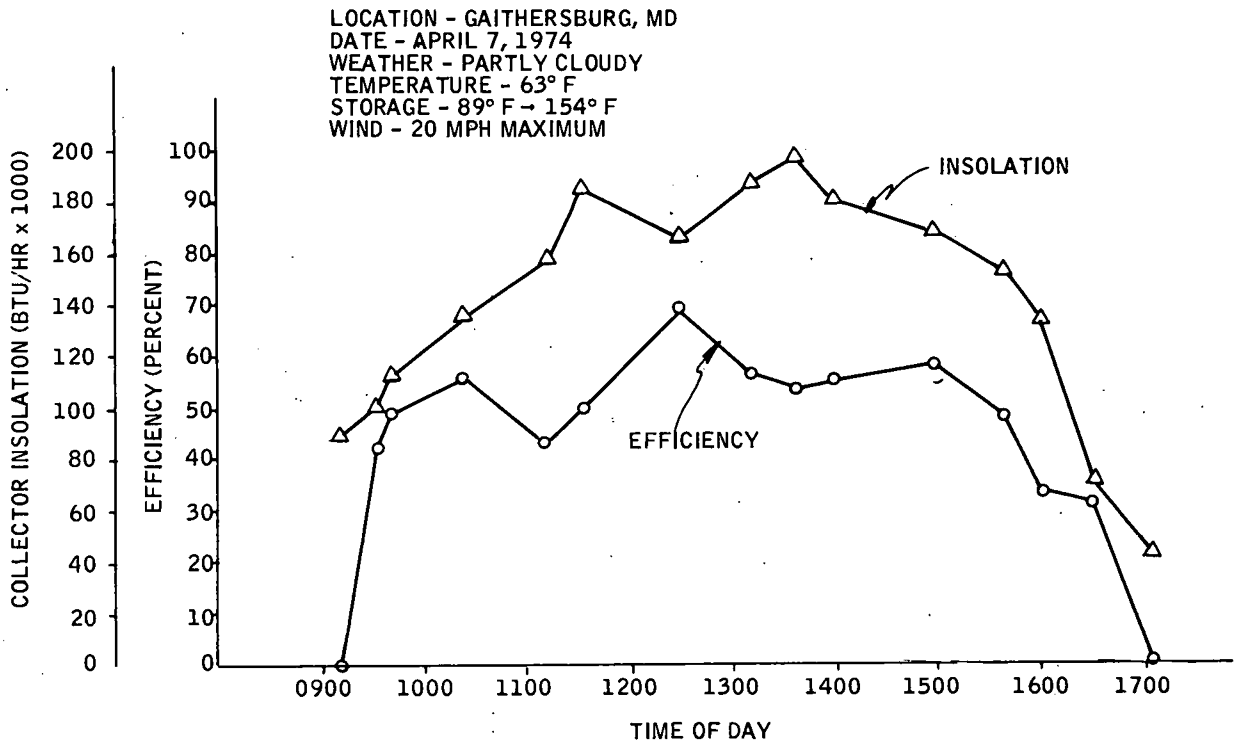


Figure A-19. Typical Data Collection, Gaithersburg, Md.

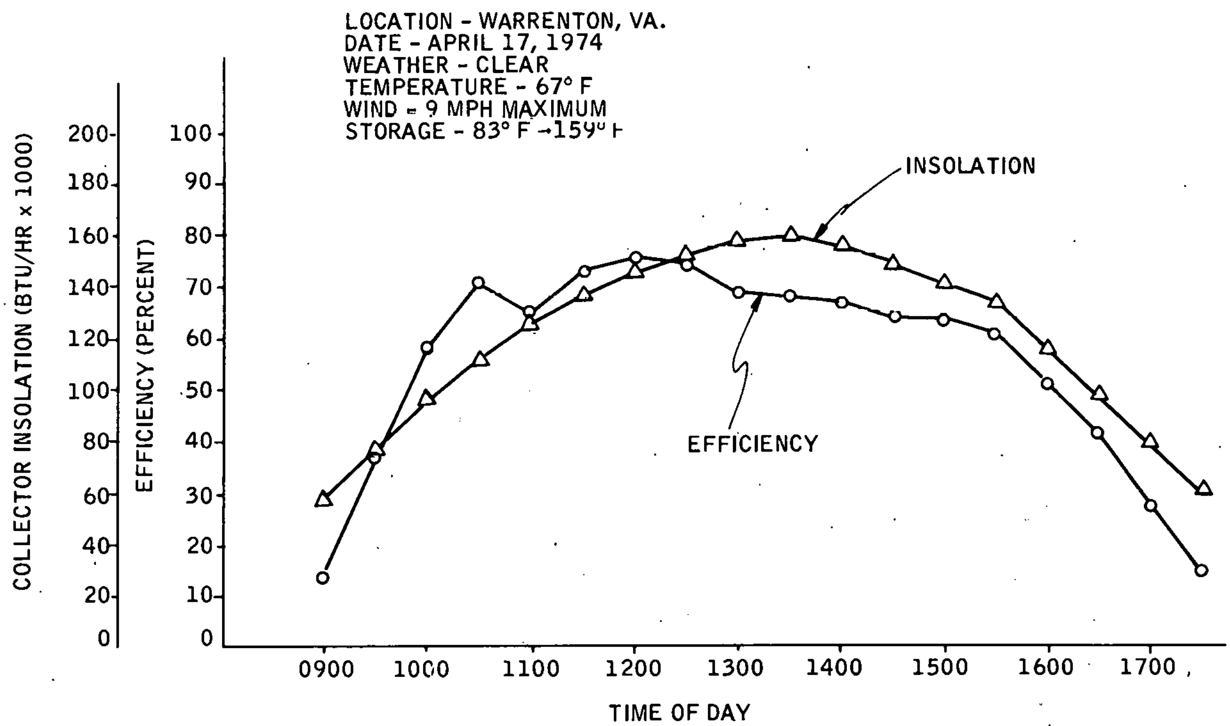


Figure A-20. Typical Data Collection, Warrenton, Va.

The laboratory was at Warrenton, Virginia, from 12 April until 23 April. With the varying conditions of weather present, operation was monitored daily and efficiencies calibrated and tabulated. During this period the average efficiency was 51 percent. Plots of these efficiencies are shown in Figures A-21 and A-22. Another method of presenting the collector performance is to plot efficiency versus collector temperature -- ambient temperature/input radiation. Such a plot is shown in Figure A-23.

A.A.I. School - Timonium, Maryland

The laboratory was set up and operating in Timonium on 26 April. Again it was set up to compare with a school heating system and to calibrate the A.A.I. pyranometer. Operation of the system proved much the same as with previous date. A time shift is noticeable here in the plots because the equipment van was located approximately 9 degrees east instead of 6 degrees west to be compatible with previous tests. Figure A-24 is a typical performance plot at Timonium. Figure A-25 is a plot of efficiency versus collector temperature-ambient temperature/input radiation at Timonium.

The laboratory was at Timonium, Maryland, from 26 April to 2 May. It was then returned to Minneapolis.

Honeywell Inc. - Minneapolis

The laboratory was set up and operated at the Honeywell Ridgway Plant in Minneapolis. A typical performance curve at Minneapolis is shown in Figure A-26.

Following some basic operation at Minneapolis, the laboratory was reworked. The main work involved installing two air conditioning systems and removing the solar panels for replating and cleaning. During the panel removal, sludge was discovered in the lower manifolds of the solar collector. Samples of this sludge were removed and sent to Dow Chemical for analysis. A copy of this report is attached as Figure A-27.

COMMENTS AND CONCLUSIONS

Rapid Heat Mode

While the lab was in the east, no propane was purchased. Consequently, it was quite uncomfortable during the early sunlight hours. To warm the air rapidly, the console was placed in manual mode and set up to circulate through the solar collector panels, through the collector coil in the air duct and directly back to the collector. This produced a fast warm-up mode. This mode should be included in the automatic control system as a base conservation method.

Solar Collector Flow Control

The present flow control valve system (FCI) proportions the flow to maintain a desired outlet temperature from the solar collector. This has two distinct disadvantages in that it does not control beyond a set temperature where the flow is not adequate to cool the panel to the set temperature and it does not shut down at night until the total system temperature (storage) drops to the set temperature. The system could be simplified and the cost reduced by having a differential controller sensing inlet and outlet controller temperature and controlling a simple two-position valve.

With this type control, as soon as the solar collector panel temperature exceeded the storage temperature (collector inlet) the valve would open thus collecting some energy regardless of amount. The flow would remain through the collector until such time as the inlet temperature exceeds the outlet temperature, then the valve would close. This eliminates the problem of the operator having to compare the manual set temperature with the collector inlet temperature.

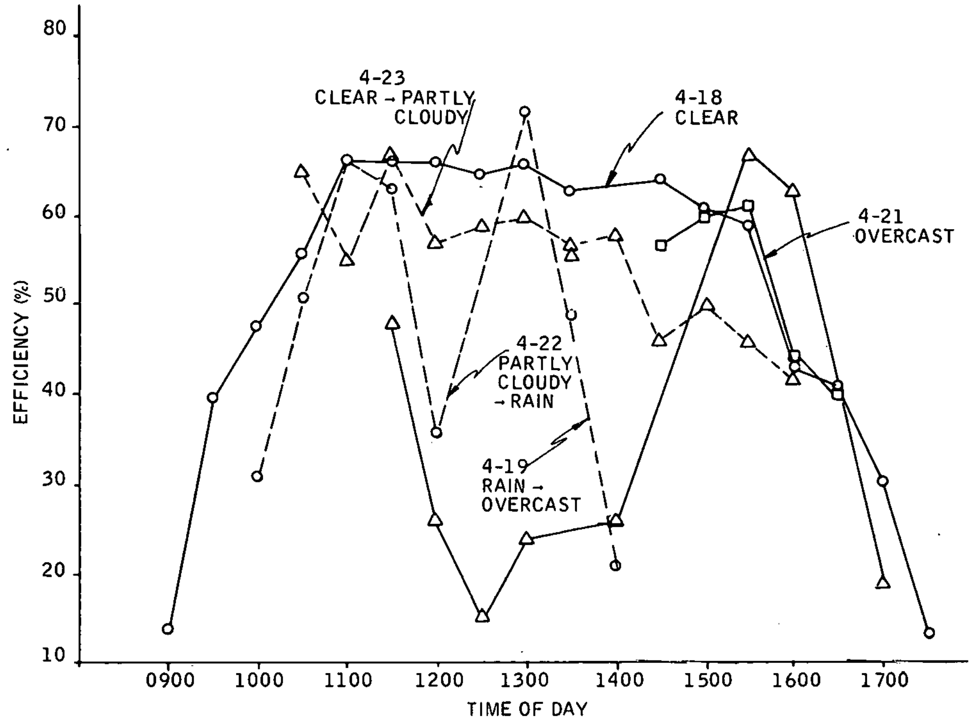


Figure A-21. Efficiency Data, Warrenton, Virginia

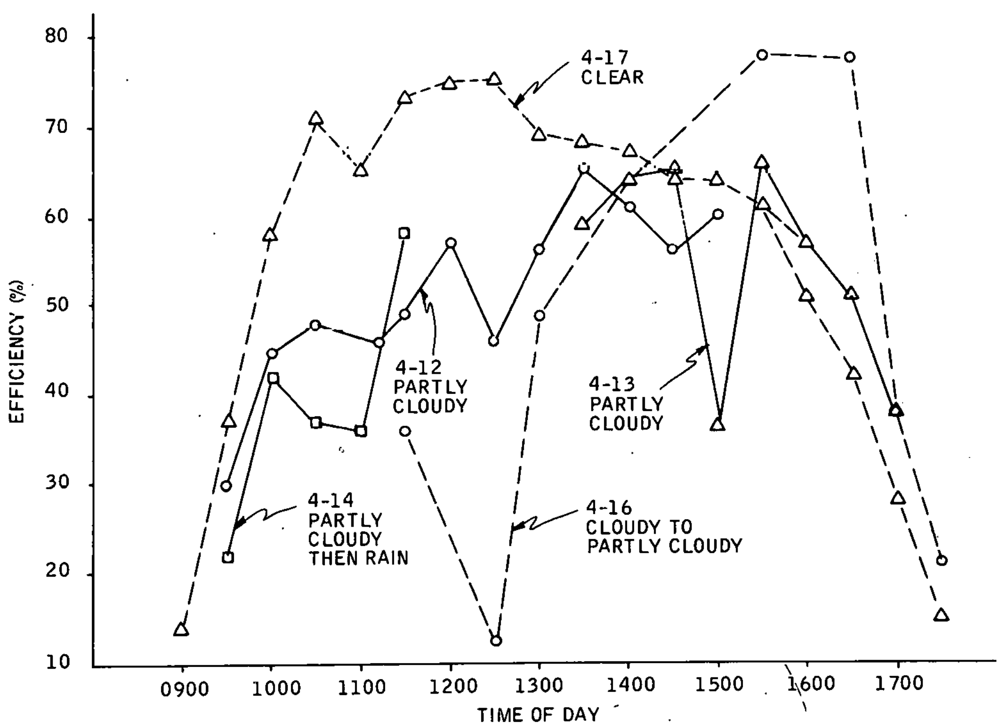


Figure A-22. Efficiency Data, Warrenton, Virginia

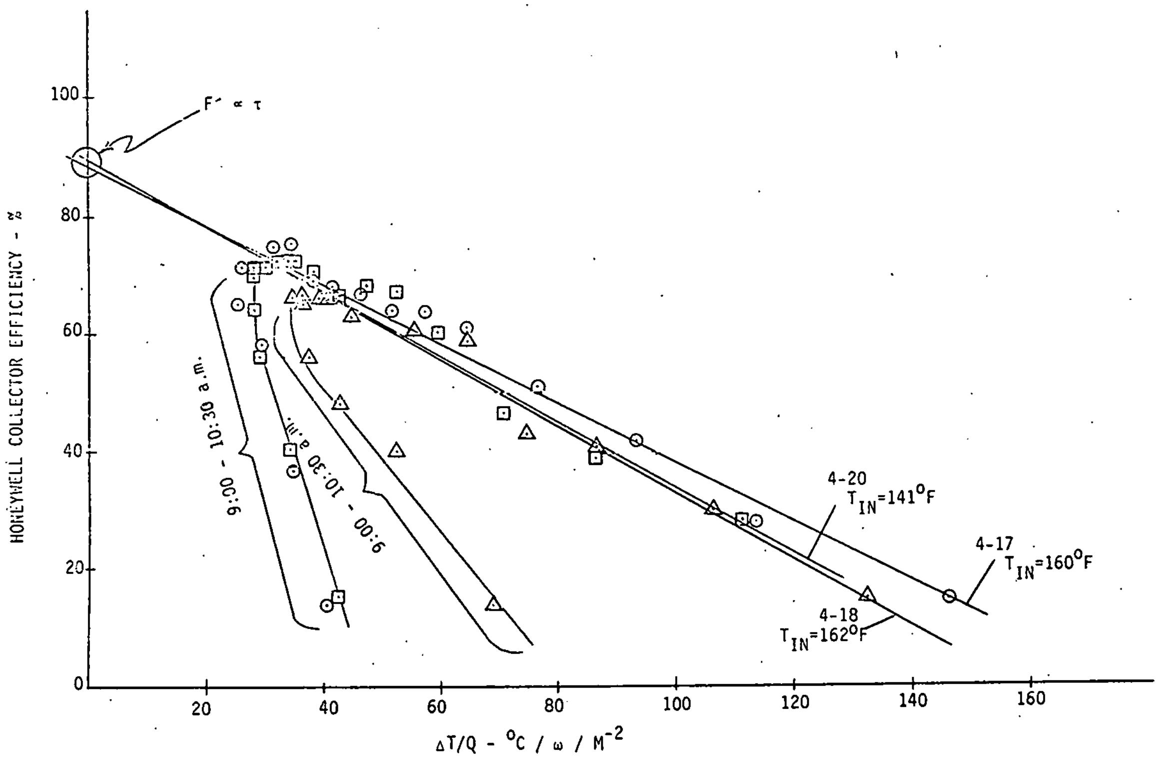


Figure A-23. Honeywell Collector Efficiency f[(Collector Temp.-Ambient)/ Input Radiation]

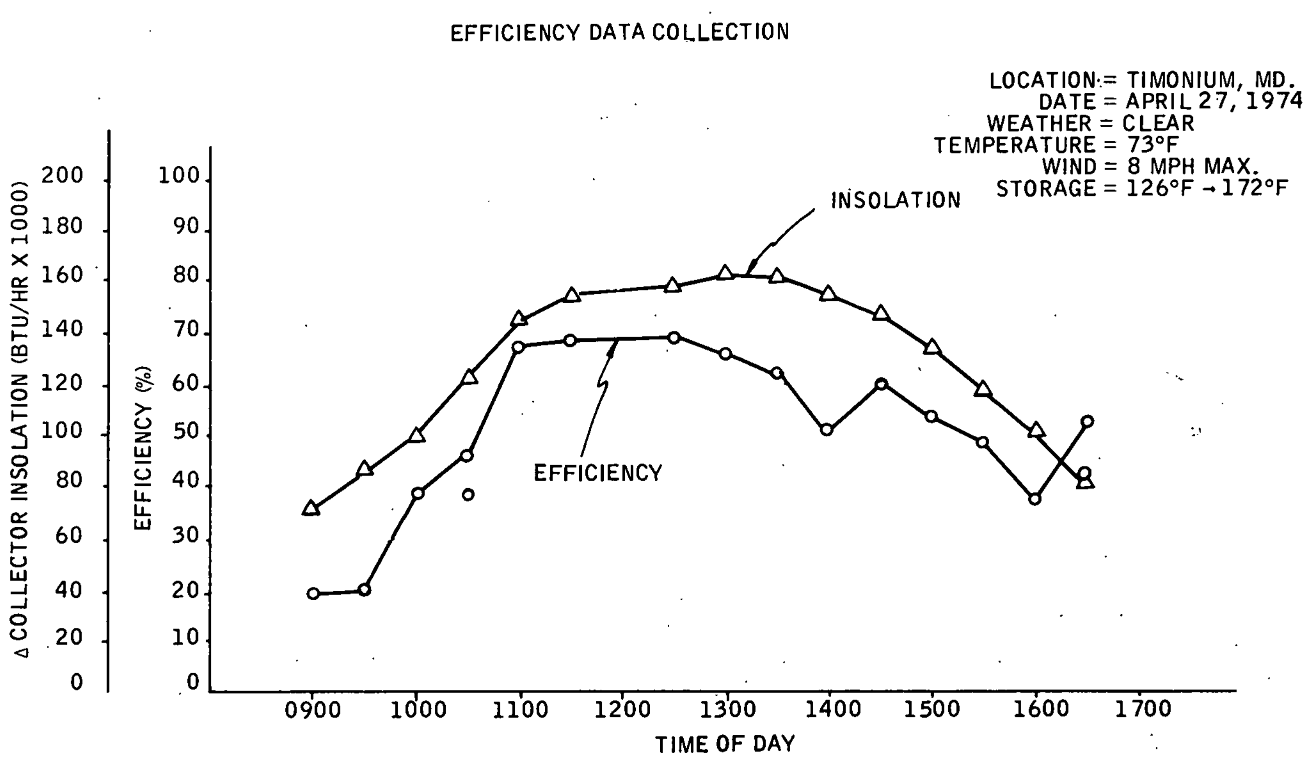


Figure A-24. Typical Data Collection

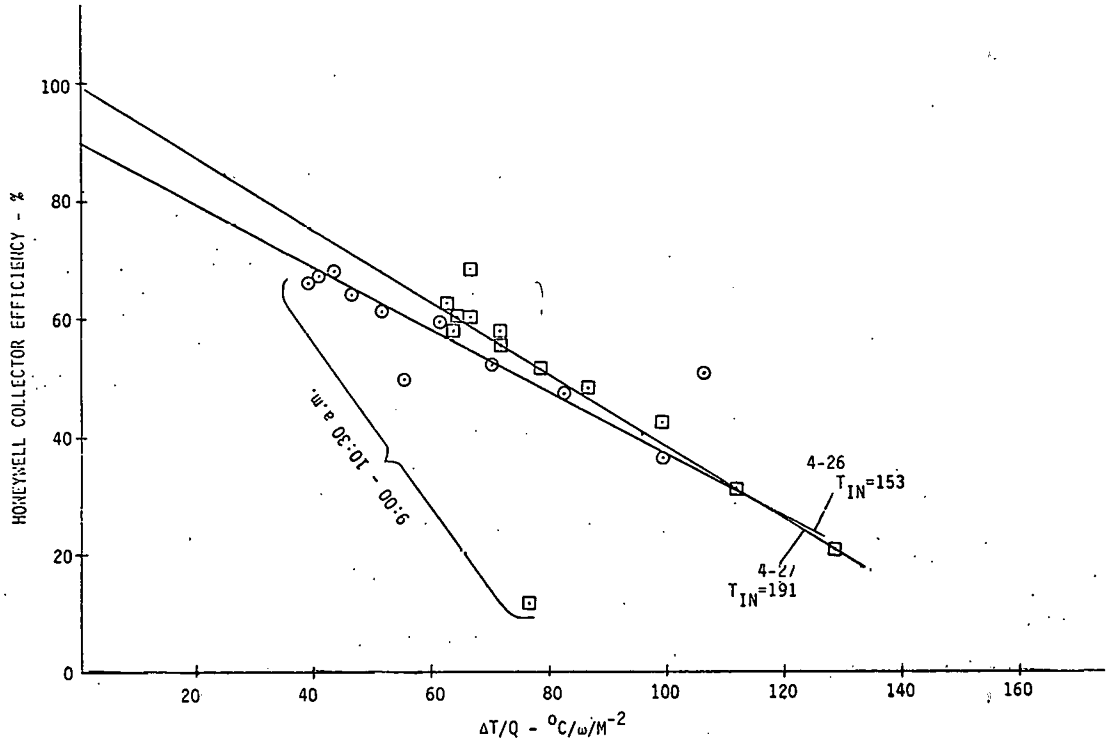


Figure A-25. Honeywell Collector Efficiency f[(Collector Temp. - Ambient)/ Input Radiation]

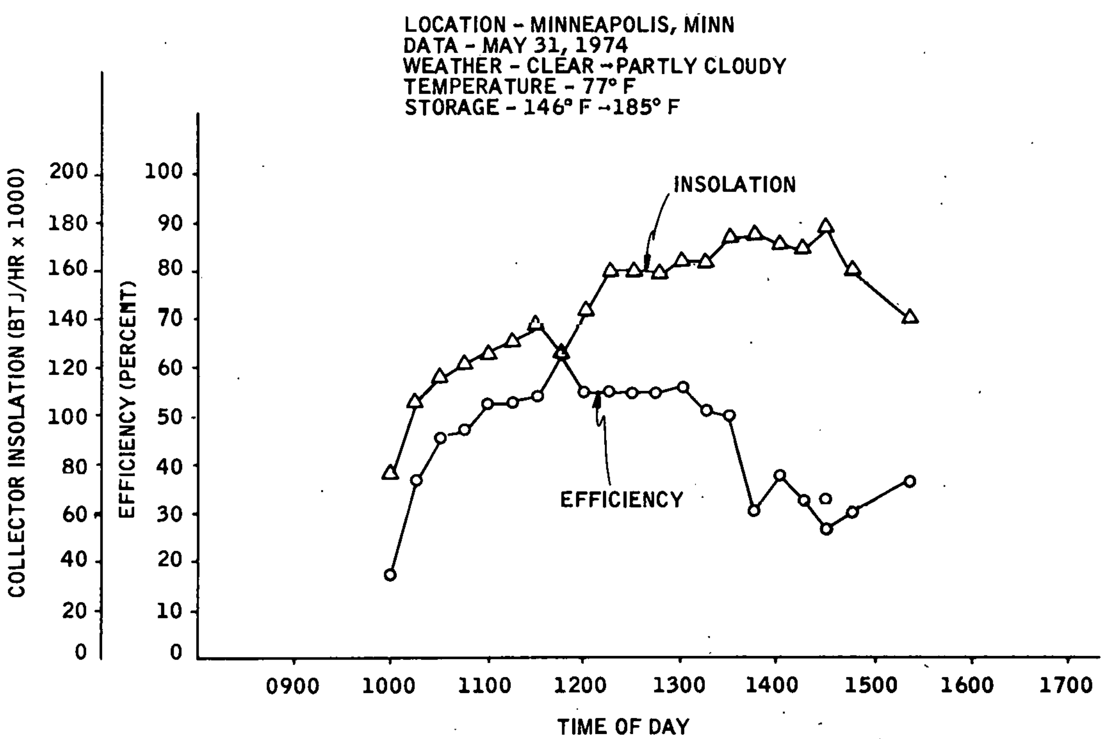


Figure A-26. Typical Data Collection

SRC SYSTEMS

TELEPHONE CALL SUMMARY

DATE : July 22, 1974

Originator(s): R. R. LeChevalier

Recipient(s): Walt Scifert - Dow Chemical
Midland, Michigan - 517/636-3993

Subject: ANALYSIS OF SLUDGE AND GLYCOL REMOVED FROM SOLAR LAB COLLECTOR PANEL

DISTRIBUTION

A. Dib
J. Kopecky
J. Ramsey
R. Schmidt
M. Selcuk
A. Severson
E. Zoerb

SUMMARY:

ANALYSIS

Samples of sludge removed from the lower manifolds of the solar collector and also a clean solution of glycol were sent to Dow Chemical for analysis.

The sludge was composed of glycol, oil residue, 9% iron and low concentrations of aluminum, calcium, magnesium and zinc. The presence of zinc suggested we might have galvanized pipe in the system. More likely the zinc, magnesium and calcium are due to hard water in the system. Since we soften the water prior to adding it, this does not make sense. The presence of iron was attributed to scale on the insides of the panels due to improper cleaning prior to filling.

The glycol looked good and requires no additional additives. The phosphate inhibitors were reduced from .5% to .4% due to precipitation of the organics in the water. The reserved alkalinity was 11% - 7% to 13% is the specified range. The glycol concentration is 35 1/2% by weight.

RECOMMENDATIONS

Prior to filling, sodium phosphate or suitable substitutions should be used to remove oil and grease. Acidic solution is needed to remove any scale in the system. He suggested we contact Oakite for recommendations. He also recommended the use of an in-line filter for a period of one month to collect any additional scale which may come loose from the inside of the system. If precipitation continues beyond this period, it suggests the presence of a problem in the system. A 100 to 200 micron filter should suffice.

ADDITIONAL COMMENTS

Mr. Seifert discussed the use of glycol in solar systems and emphasized the importance of a closed system for glycol solutions.

Mr. Seifert also mentioned that he has been contacted by many other solar energy companies and that one of them is having a pitting problem. The system with the problem is an open system and is using aluminum panels.

Figure A-27. Telecon re Sludge and Glycol Residue

Storage Stratification

The original plan was to use the appropriate temperature water from the storage tanks to best maintain efficiency. It was found that much of the time, because of system flow, that the storage temperature varied only a few degrees from top to bottom. In the automatic mode, the flow was predominantly into the top and out of the bottom during solar collection and into the bottom and out of the top when there was no solar collection (when heating was required). The center tap was not used for any appreciable time.

APPENDIX B

FINAL REPORT ON RANKINE CYCLE POWERED AIR CONDITIONING SYSTEM

BARBER-NICHOLS ENGINEERING

FINAL REPORT ON RANKINE CYCLE POWERED AIR CONDITIONING SYSTEM

October 3, 1974

Prepared for: John Kopecky
Honeywell, Inc.
Minneapolis, MN

Prepared by: Robert E. Barber
James E. Dillard, Jr.

6325 West 55th Avenue, Arvada, Colorado 80002 Telephone 303 [illegible]

1.0 SUMMARY

This final report includes a review of the testing at Barber-Nichols and the proposed improvements based on this testing. The first test was a performance test of the air conditioning system only. This test was to verify that 36,000 BTU's of cooling could be made with 1.9 HP into the compressor. The results showed the unit operated at approximately 5% above rated cooling and about 5% above rated power, meeting the design goal.

The Rankine cycle system tests were begun but it was found that the boiler capacity was low. At this time it was felt that a new boiler was necessary. A boiler made from a refrigeration water chiller was found to be approximately the correct size. During this period a preheater was also installed to be assured of having enough vapor flow to reach the design point. While the installation was proceeding, an electric motor/generator calibration was run showing a motor efficiency of 63% and a generator efficiency of 77%. The calibration allows correlation with the performance data for both the motoring and generating modes.

System testing was continued with the new preheater, boiler, and vapor separator. Testing proved that the Rankine cycle system would meet the design cooling goals but could not operate with the air conditioning compressor engaged and the electric motor turned off during reduced collector water tempterature tests. An electric pump drive was installed to solve this problem. Plumbing changes were made to reduce the overall pressure drop in the system and with this final system configuration it was found that the output horsepower of the gearbox at the design point was approximately 2.0 HP with 215°F collector water and 85°F cooling tower water, meeting the design goals. The pump motor and controls require approximately 750 watts input. The turbine efficiency in the final configuration was 72%, resulting in a system COP of 0.5.

2.0 COMPONENT TESTING

The following sections include the results of the air conditioning, motor calibration, and Rankine cycle system tests.

2.1 AIR CONDITIONING TESTING

The air conditioning test was run using an electric motor drive through a rotating torquemeter and belt drive to the air conditioning compressor. The actual compressor input horsepower (including the belt drive) could be observed by reading the input speed and torque at the design compressor inlet pressure and temperature, and the design cooling water flow and temperature. The air conditioning test results are shown in Table I, which shows that the air conditioning system met the design goals. Since the unit performance was proven, it was felt that now the Rankine cycle drive could be attached to the system and the system would meet the design point of 36,000 BTU output.

TABLE I

A/C TESTS

Data Point	Compressor			Condenser - Water Side					Cooling tons
	T_{in}, ^{o}F	P_{in}, psia	T_{out}, ^{o}F	T_{in}, ^{o}F	T_{out}, ^{o}F	ΔT, ^{o}F	w, gpm	Q, BTU/hr	
Design	70	56.2	128	85	93.2	8.2	10	40,800	3.0
3-4-1	78	56.2	125	85	94	9.0	10.09	45,232	3.34
3-4-2	70	56.5	124	86	94.5	8.5	10.08	42,720	3.13
3-4-3	73	56.2	125	85.2	93.5	8.3	10.08	41,715	3.05
Ave. of important values	74	56.4		85.4	94	8.6	10.08	43,222	3.173

Cond. Freon T_{out}, ^{o}F	Motor-Generator					Pillow Block Loss In-Lb	Input HP	Oil Press. psi
	N, rpm	τ, in-lb	Out, HP	Input Watts	Effic.			
90	3450	39.8	2.18	2168	.75	5.0	1.904	NA
92	3428	42.0	2.28	2353	.723	5.0	2.013	85
92	3428	42.0	2.28	2378	.715	5.0	2.013	85
92	3425	42.0	2.28	2370	.718	5.0	2.012	82
92					.719		2.013	

Conclusions:

a) Unit operates at 5% above rated cooling at 5% above rated power, therefore meets design goals.
b) Condenser performance slightly low, resulting in 97^{o}F exit temp.
c) Motor efficiency is 72%.

2.2 ELECTRIC MOTOR/GENERATOR TESTING

The electric motor/generator was tested in much the same way as the air conditioning compressor. The electric motor was driven by or loaded by a variable speed hydraulic drive through a rotating torquemeter to measure the input or output torque with a magnetic pickup to sense speed. This gives the actual horsepower out or into the motor/generator. The electrical power was measured with a standard residential wattmeter. The data shown on Figure 1 gives the complete operating range from two horsepower input to two horsepower output depending on whether the unit was operating as a motor or a generator. The efficiencies varied from 63% to 75% as a motor or a generator respectively at 2 HP. The motor required approximately 450 watts under no load conditions including the power required for the system electrical controls.

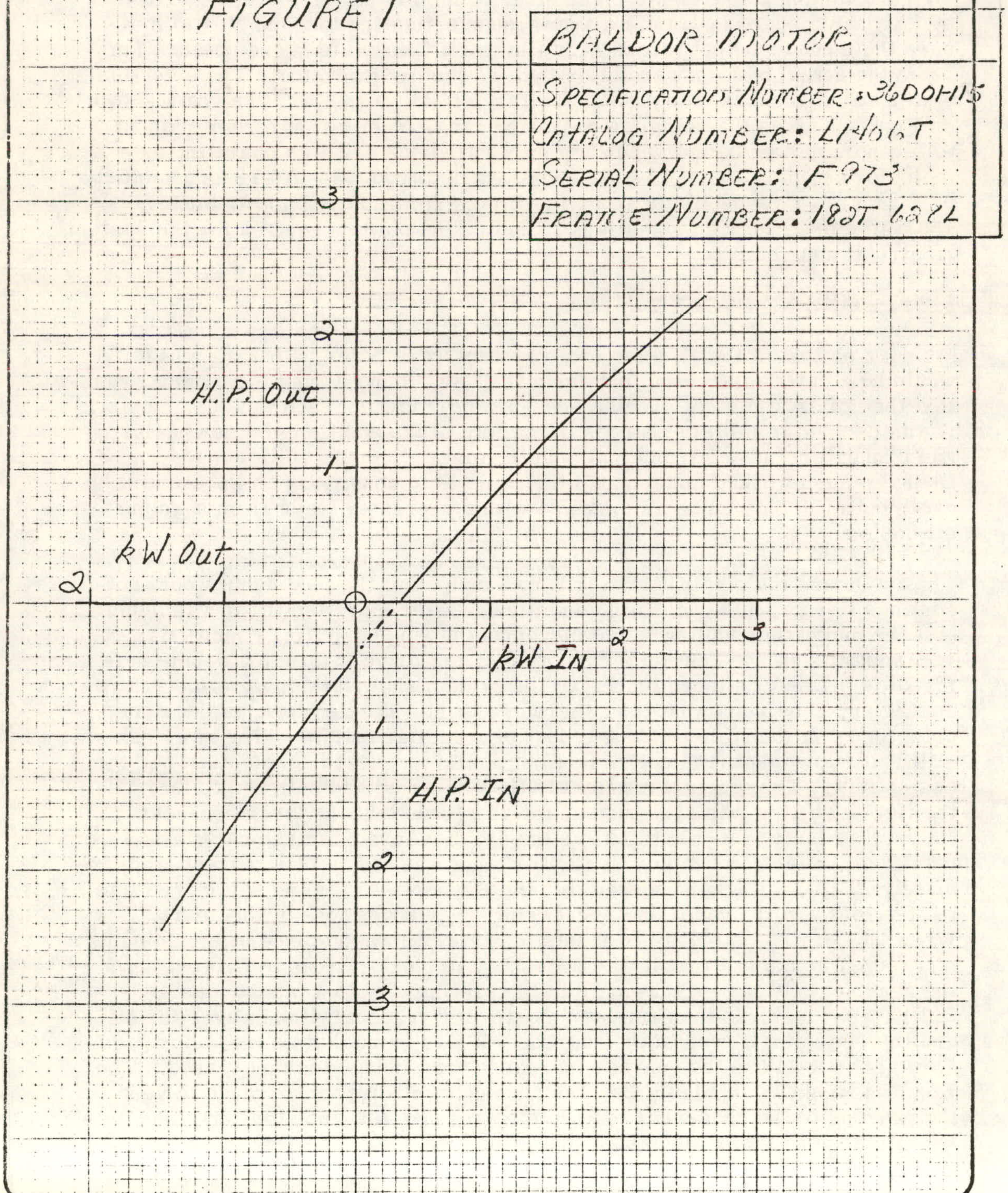
2.3 RANKINE CYCLE TESTING

The test of the Rankine cycle system were begun with what was felt would be a satisfactory working design. It was found during the original runs that the boiler would not provide enough vapor flow to reach the design point. It was decided to initiate changes in the way of the addition of a new boiler and preheater. It was found that a Refrigerant-12 water chiller made by Dunham-Bush worked under much the same fluid conditions as the Rankine cycle boiler. This unit was installed with a preheater made from a Young oil cooler.

During the original testing it was found that the feed pump would not continue to pump under certain conditions. After more testing it was found that the magnetic coupling would disconnect during cavitating conditions. A redesign was carried out on the pump to direct drive it though a double buffered lip seal. Continued testing of the system showed the nozzle block flow coefficient to be somewhat low, therefore, a new nozzle block was designed using information from a NASA report of a similar turbine. The new nozzle still had a low discharge coefficient but due to increased nozzle area had the design flow. The original nozzle was then enlarged and polished to increase the flow. Testing was continued and it was found that the discharge coefficient was improved. At this time the gearbox loss using the calibration data from the electric motor/generator tests was felt to be too great (approximately .7 HP). It was supposed that the Freon working fluid had leaked into the gearbox and the vapor was increasing the windage loss of the gears. When the gearbox was drained it was found to contain no discernable amounts of Freon. It was felt that it was necessary to conduct tests to find out why the gearbox loss had increased from the preliminary test (approximately 0.3 HP). A test was then run by motoring the gearbox with air over the oil and then injecting Freon in the gearbox. It was found that as the vapor pressure of the R-113 in the gearbox increased the windage loss increased. The results are shown in Figure 2. The necessary changes were made to vent the gearbox to the condenser to keep the pressure low and, thereby, the loss in the gearbox to the design goal of 0.5 HP.

BARBER-NICHOLS ENGINEERING CO.	PROJECT: HW	SUBJECT: MOTOR/GENERATOR CHARACTERISTICS
	BY: DAJO — DATE: 5-31-74	

FIGURE 1



BARBER-NICHOLS ENGINEERING CO.	PROJECT		SUBJECT
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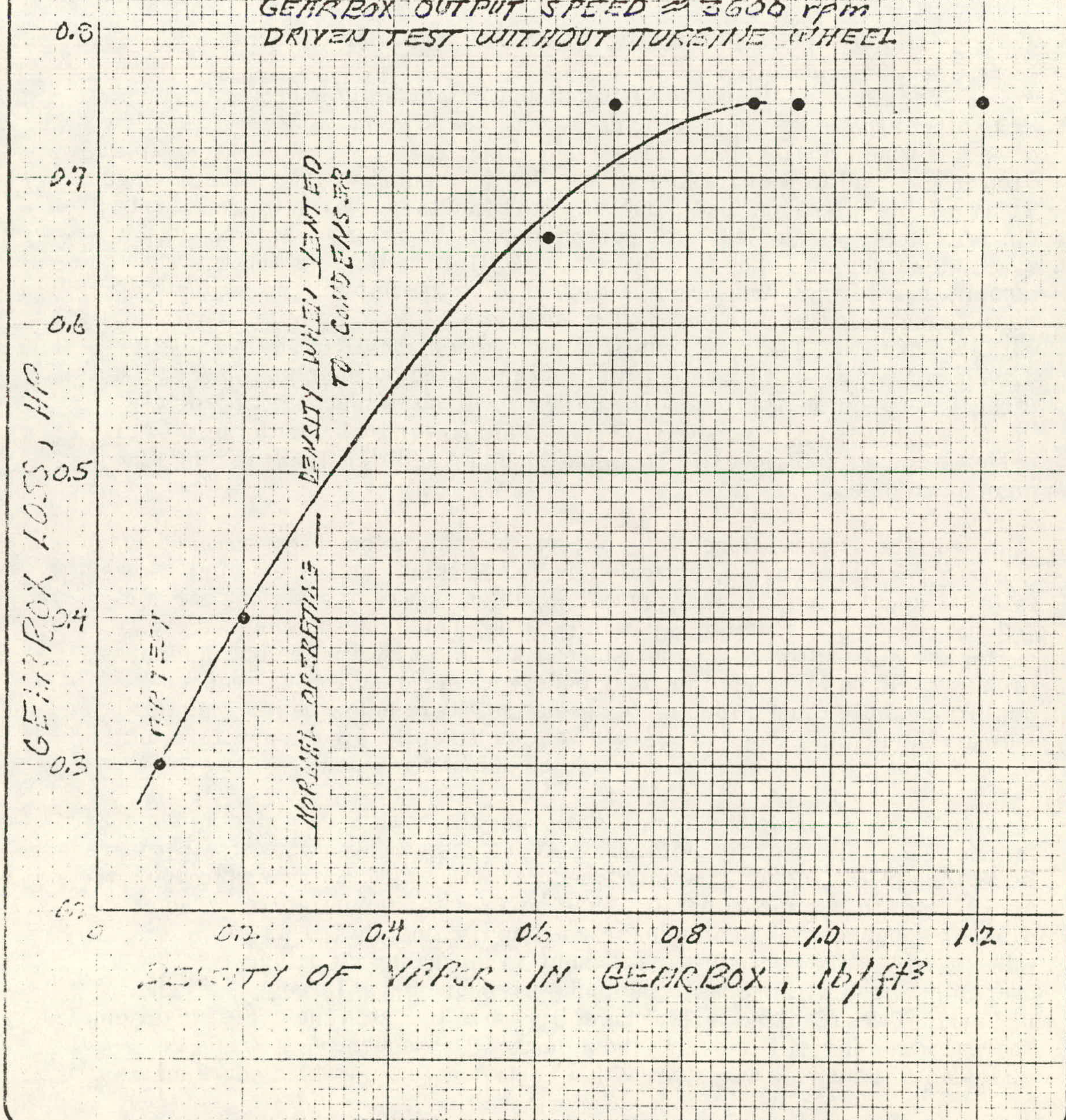
FIGURE 2
EFFECT OF DENSITY ON GEARBOX LOSSES
GEARBOX OUTPUT SPEED ≈ 3600 rpm
DRIVEN TEST WITHOUT TURBINE WHEEL



Testing had shown the original feed pump would not re-prime itself after cavitation and, therefore, could not be used with this system. A new pump was designed and fabricated using some hardware which was superior for this type of application (pumping a boiling liquid) but which would require more horsepower than originally intended. At this time certain other areas in the plumbing were found to have excessive pressure drop so changes were made to improve them. The testing with the new pump and revised plumbing showed the system to operate correctly with the pump cavitating and the turbine operating in a stable condition at the design point. At this time the testing with the Rankine cycle was felt to have proven the system satisfactory so the air conditioning Rankine cycle system tests were begun.

3.0 AIR CONDITIONING-RANKINE CYCLE SYSTEM TESTING

The air conditioning tests began and immediately it became evident that without the motor/generator operating, the system would begin to slow down and would finally stop. This performance was the result of the rapid drop in the head flow characteristics of the feed pump with decreasing speed. To avoid this effect the feed pump was modified to be driven by a separate electric motor which makes the pump independent of the turbine speed. An available motor was used (a 3600 rpm, 1/2 HP induction motor), and the combined motor and drive system proved to be inefficient (approximately 50 to 60% efficient) but the overall system was found to operate satisfactorily with this change. The test results with the mechanically and electrically driven pump are shown in Figure 3. As indicated by the horsepower difference between the two tests, the power required to mechanically drive the feed pump is approximately .5 HP. This power includes the loss of the belt system, an intermediate jack shaft, and an additional belt driving the high speed pump.

In the original Honeywell concept, it was proposed that the motor/generator would always be activated during Rankine cycle operation. However, it would provide makeup power when necessary to drive the air conditioning system at three tons output (2 HP). This concept was changed during the fabrication of the unit since it was felt that a more optimum approach would be to utilize the solar energy alone, resulting in reduced cooling capacity when collector temperatures were below design. In this manner the turbine speed decreased at reduced collector temperature until the compressor load matched the turbine output horsepower.

Figure 4 shows the compressor rpm and cooling provided as a function of solar water temperature. At the design solar water temperature three tons of cooling was accomplished and this output decreases to approximately one ton at approximately 175°F solar water temperature. The compressor speed and the cooling decreased approximately 3:1 as shown in this figure. If the centrifugal feed pump is mechanically driven a 3:1 change in speed will decrease the pump pressure 9:1. However, as shown in Figure 5 the required turbine inlet pressure to match the cycle conditions decreased a ratio of 2;1 during these tests. Consequently, it would be impossible to provide a system with a centrifugal pump which could match

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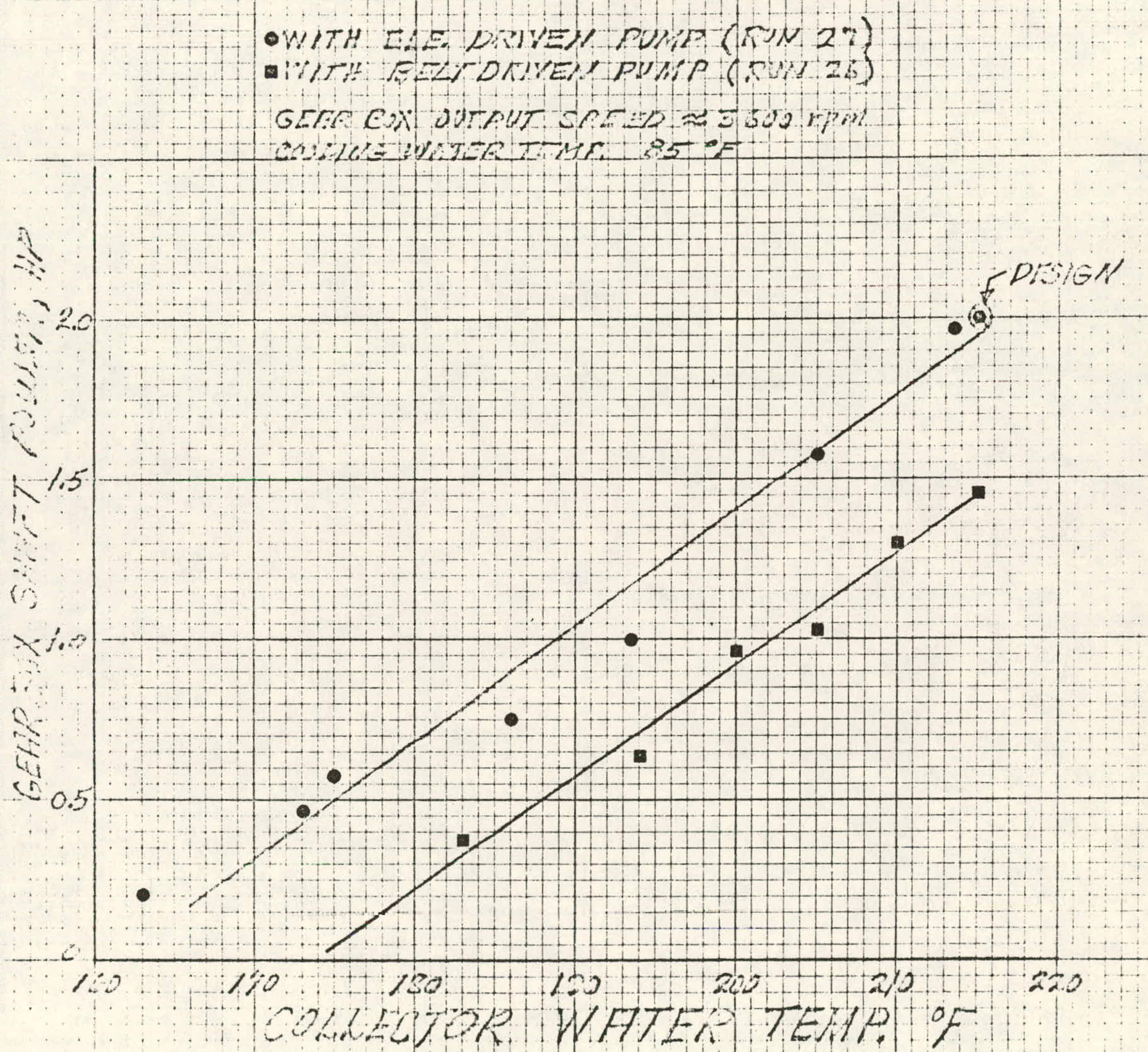
FIGURE 3

RANKINE CYCLE OUTPUT AT REDUCED COLLECTOR WATER TEMP.



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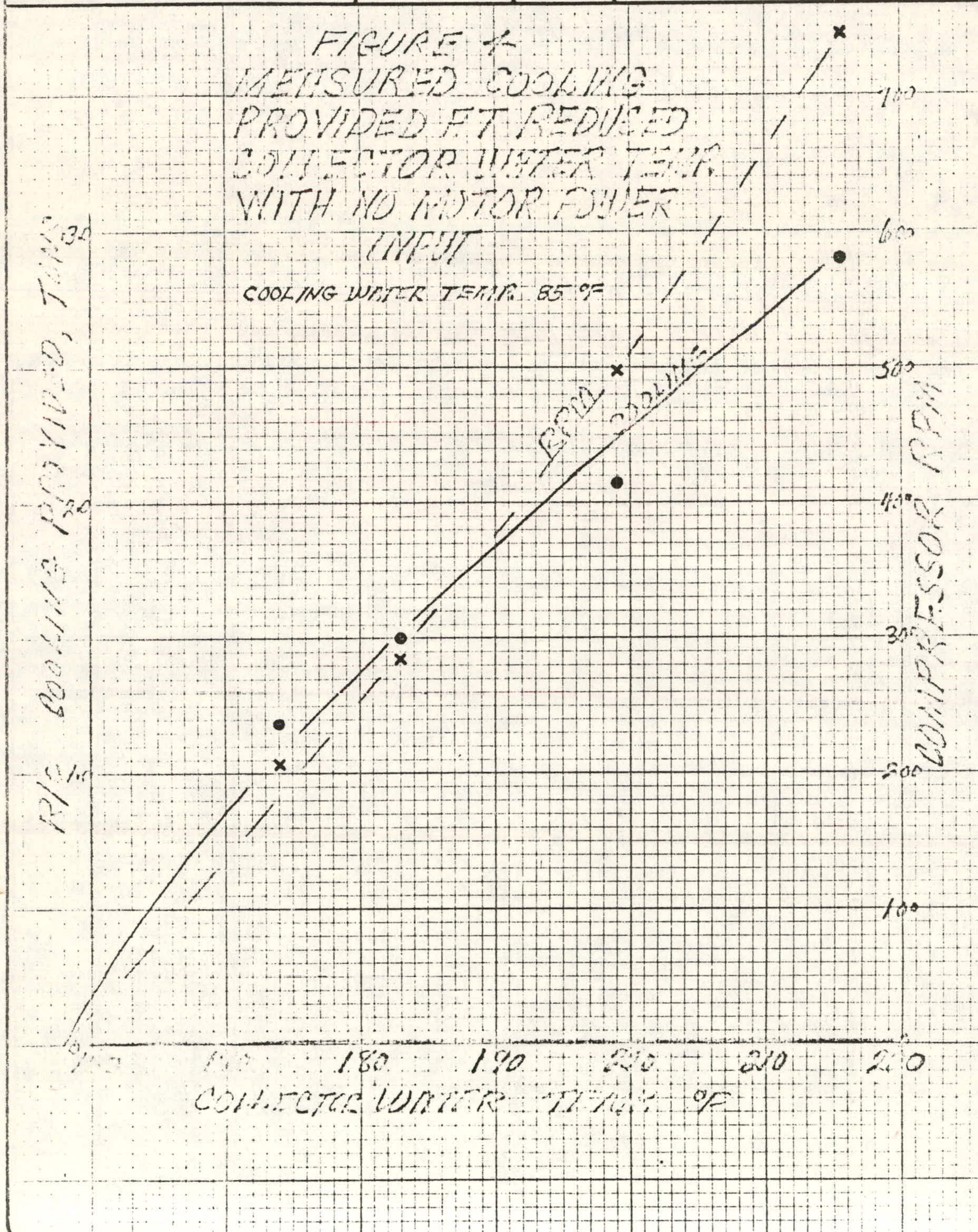

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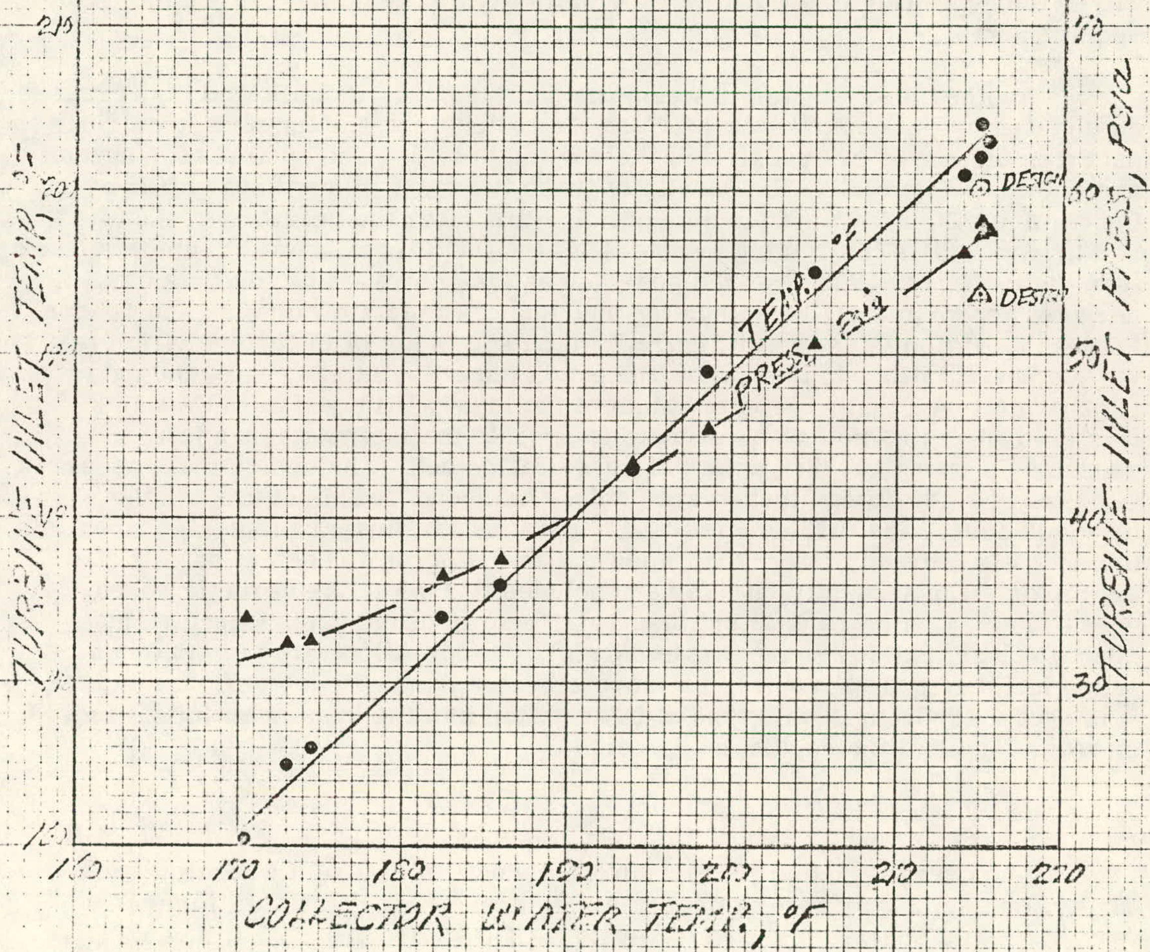
BOILER PERFORMANCE AT REDUCED COLLECTOR TEMP.

FIGURE 5

COOLING WATER TEMP 85°F



the system characteristics without a variable speed pump drive mechanism if the turbine speed and, hence, compressor speed were allowed to vary as the cooling water temperature decreases. Therefore, it was concluded that for the present system the feed pump must be powered separately from the Rankine loop in order to allow this variable speed operation.

3.1 TURBINE EFFICIENCY

Based on the data of Runs 26 and 27 which was partly shown earlier, it was possible to evaluate the turbine efficiency as a function of variable collector water temperature for two different operating modes. Initially the unit was driven with the motor operating and the collector water temperature was decreased to evaluate the amount of electrical generator power output at a fixed speed. The results of this test showed that at 168^{o}F collector water temperature only control and field power is generated, resulting in a net zero electrical out of the gearbox, and at a collector water temperature of 215^{o}F the generator is producing 1110 watts greater than the control and field requirements.

Another test was run in which the turbine speed was allowed to decrease as dictated by the compressor load and the collector water temperature. This data was discussed previously and the cooling provided and compressor speeds are shown in Figure 4. The results of these two tests allowed evaluation of the turbine efficiency as a function of collector water temperature as shown in Figure 6. It is interesting to note that at a 160^{o}F cooling water temperature the turbine efficiency at the design speed is approximately 60% as compared to approximately 20% when the unit is allowed to slow down. The reason for this variation is shown in Figure 7 which shows a plot of turbine efficiency versus speed ratio. In this figure the predicted curve is shown and the data for the two test runs. The data points to the higher than design U/C_o which occurred when the motor was allowed to operate maintaining essentially a constant tip speed U. The points at speed ratios below the design point occurred when the variable speed tests were made. As shown on this curve and the previous curve, the turbine operating conditions are further off-design when the speed is allowed to vary than when it was maintained constant by using the motor. Consequently, it can be concluded that while the variable speed operation may be more efficient on a solar energy usage basis (since no additional electrical power input is required), the operation with the motor as supplementary power results in a more efficient Rankine cycle. Consequently, it is necessary to input this type of information into the system optimization in order to select the superior approach.

3.2 SYSTEM PERFORMANCE

The overall system performance is shown in Figure 8 where the system COP versus collector water temperature is presented. This data for the variable speed operation shows the system COP varied from .5 at design temperature to approximately .25 at 170^{o}F collector water temperature. While the air conditioning data was not taken under the constant speed condition (motor on), the COP would increase to approximately 1.0 since half

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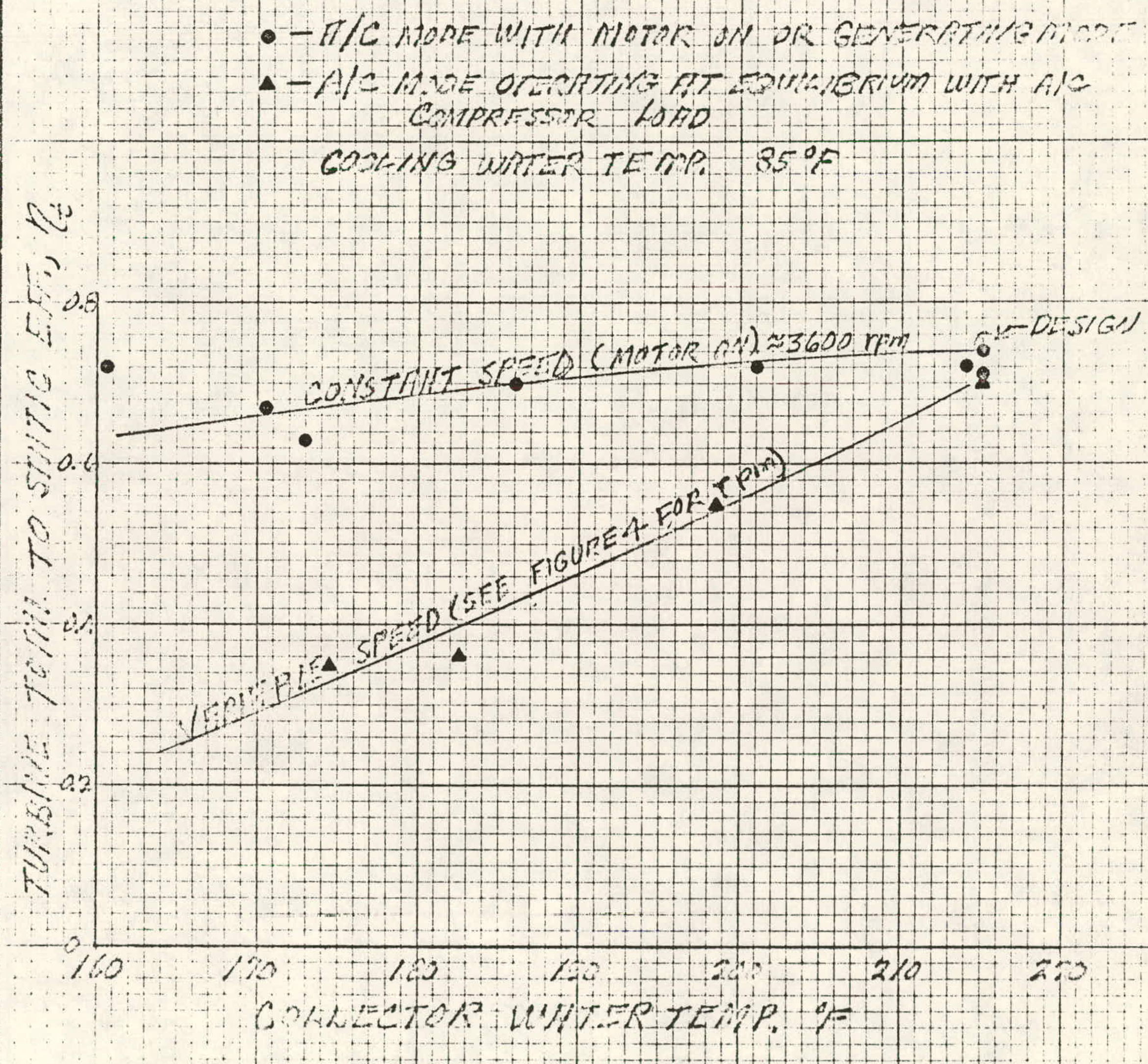
FIGURE 6
EFFECT OF OPERATING MODE ON TURBINE EFFICIENCY



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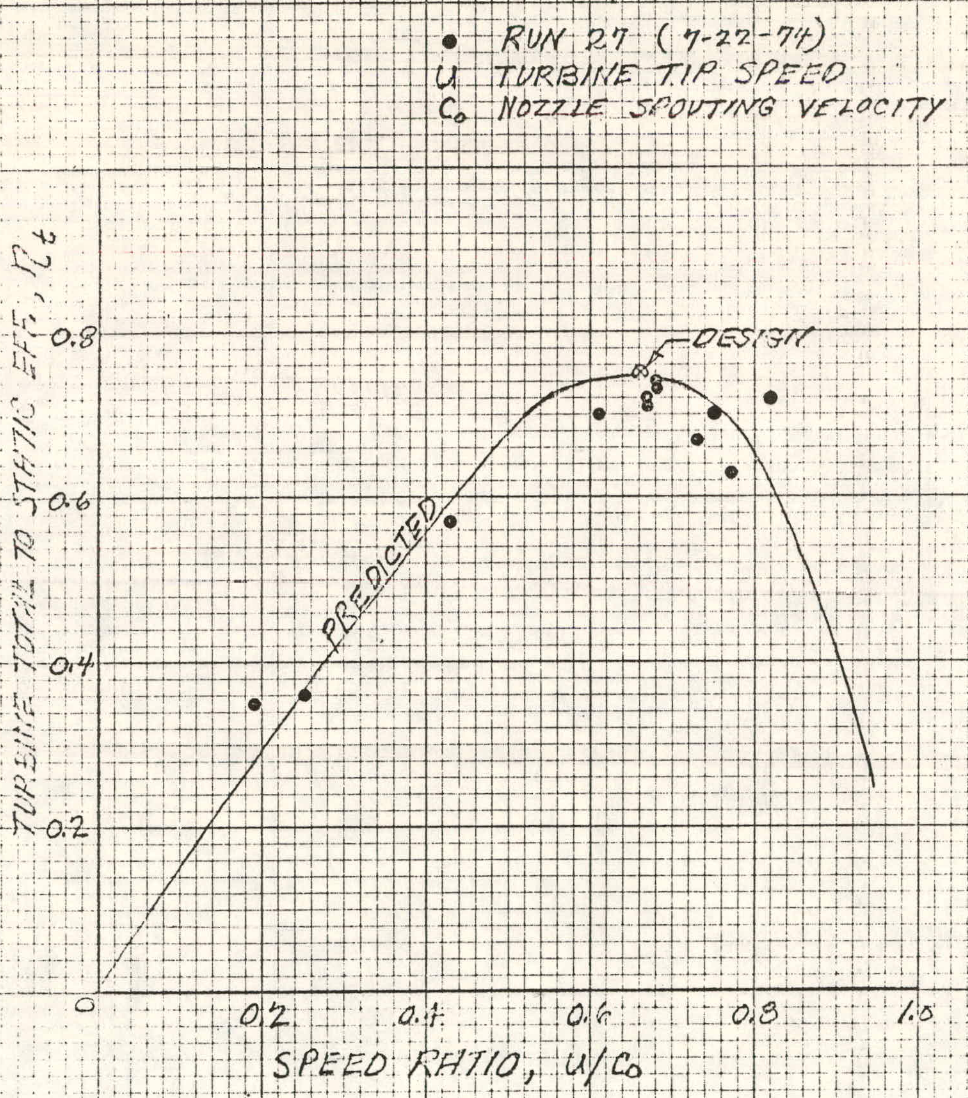
PROJECT

SUBJECT

BY

DATE 9-19-74

FIGURE 7
COMPARISON OF MEASURED AND PREDICTED TURBINE EFFICIENCY



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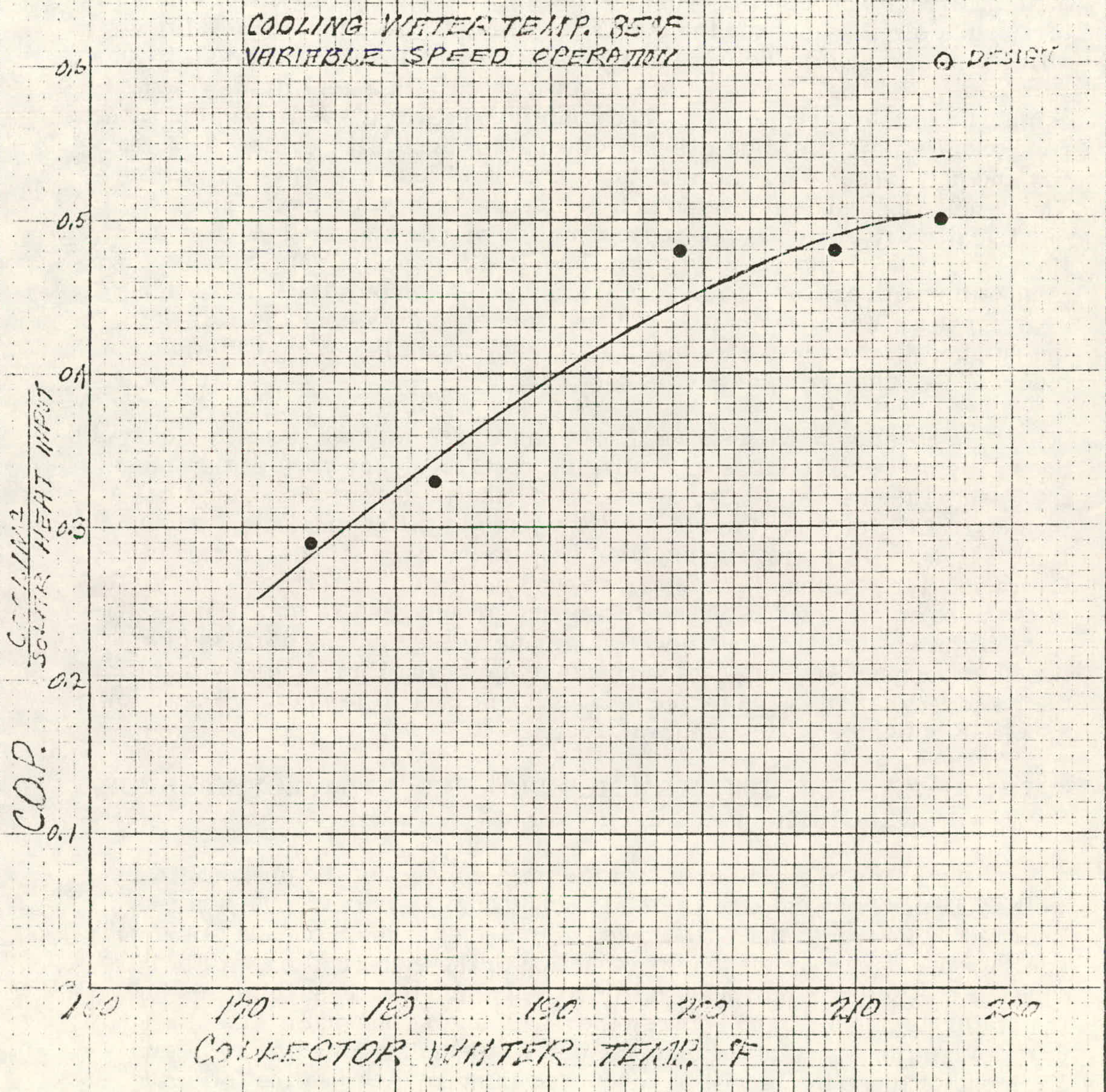
BY	DATE 9-18-74

C.O.P. AT REDUCED COLLECTOR TEMP.

FIGURE 8



of the cooling load is carried by the electrical system which has a COP of 0.6. Consequently, COP with electrical augmentation has little meaning as a measure of goodness for a solar system.

The overall cycle efficiency for the system at various collector water temperatures is shown in Figure 9. This figure shows the breakdown of losses throughout the system beginning with the ideal Carnot cycle efficiency for a system operating between the collector water temperature and 85°F cooling water. The condenser loss is that loss associated with the temperature difference between the cycle condensing and the cooling water temperature. In a similar manner the boiler loss results from the temperature difference between the boiler exit temperature and the collector water temperature. The real fluid loss is that associated with a real fluid cycle as compared to the Carnot cycle. The cycle losses are those associated with component efficiencies and pressure drops throughout the cycle. The gearbox and belt losses, generator loss, and control loss are self-explanatory. The calculations shown here do not include the feed pump power since it is electrically driven and is not representative of a proper feed pump system. However, it does show that an overall cycle efficiency of approximately 5% is possible with as low as 215°F collector water temperature. With improvement or larger sizes it is felt that this efficiency could be raised to the neighborhood of 6 to 8% overall efficiency which is not too bad when considering Rankine systems in existance today.

In conclusion it is shown that the unit presently operated by Honeywell does provide the three tons of cooling as required, does very nearly meet the design turbine efficiency, meets the predicted gearbox loss, and has a measured COP of 0.5. Simple improvements in the components, as outlined below, will improve the system COP to the original design goal of 0.6. Operational experience to date has shown that the unit is far superior to that of the absorption system. The operational advantages are primarily in the area of the electrical supplemental approach needed during periods of reduced solar input and the low temperature operating limit whereby some useful power can be made by the solar system even at temperatures as low as 170°F collector water temperatures.

4.0 RECOMMENDED SYSTEM MODIFICATION TO IMPROVE PERFORMANCE OF THE PRESENT SYSTEM

Six areas are recommended for improvement of the present Honeywell solar powered Rankine cycle air conditioning system. These areas include: 1) improvement of the feed pump drive; 2) modification of the gearbox to reduce gearbox losses; 3) reduction of the electrical control power; 4) a more efficient motor/generator; 5) reduction of pressure losses between the boiler and turbine and between the turbine and condenser; and 6) a study to optimize the match of the Rankine unit to the solar collector system. Each one of these items will be discussed in more detail in the following sections.

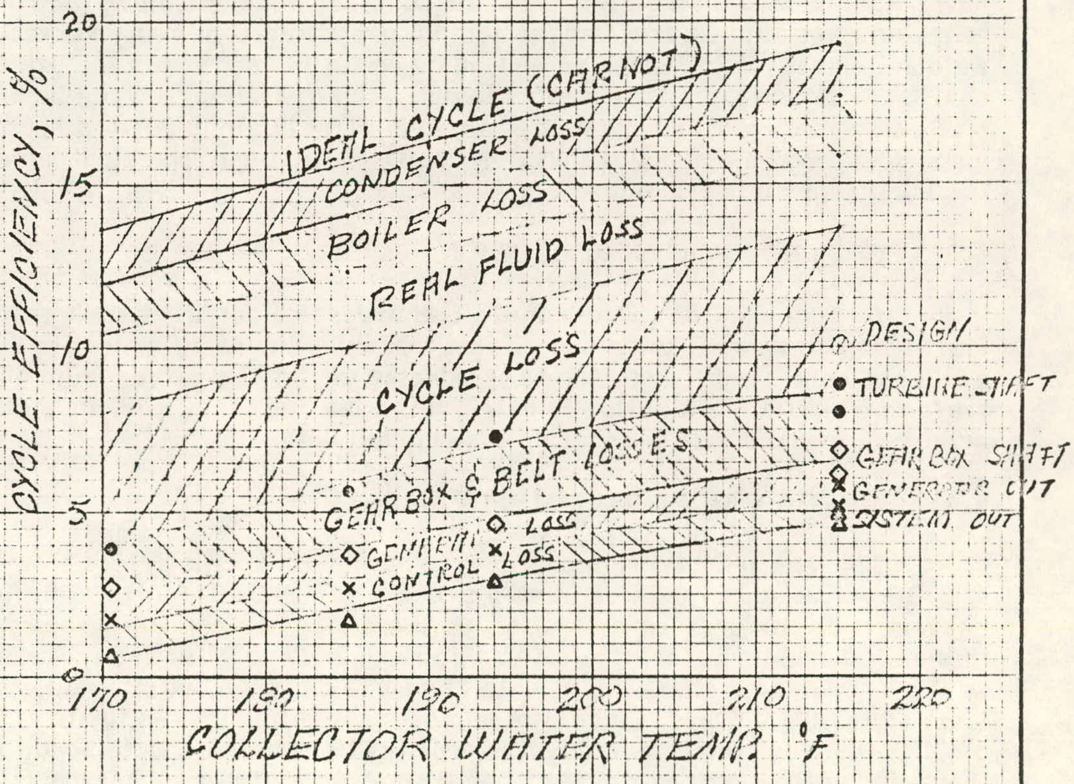
BARBER-NICHOLS ENGINEERING CO.

PROJECT	SUBJECT
BY	DATE 9-19-74

FIGURE 9
SYSTEM LOSS DISTRIBUTION

DOES NOT INCLUDE FEED PUMP (IDEAL PUMP WORK 0.05 HP)

COOLING WATER TEMP. 85°F
CONDENSING TEMP. 95°F



4.1 PUMP IMPROVEMENT

The present pump drive utilizes a low speed, 3600 rpm motor with a belt speed increaser to 13,000 rpm to drive the pump. On the high speed shaft a back-to- back shaft seal is provided to prevent fluid leakage. This particular pump drive arrangement requires 750 watts electrical input to the motor to drive the pump. It was found that a new high speed motor could be coupled directly to the pump drive thereby eliminating the belt losses and the losses from four bearings. Additionally the high speed motor has a measured efficiency of 75% versus the 50% of the present system. This would result in an overall electrical input to the motor of the reworked configuration of 350 watts, resulting in a 400 watt saving.

4.2 GEARBOX IMPROVEMENT

The present gearbox arrangement is set up with a seal on the high speed shaft and a back-to-back hermetic seal on the low speed shaft. Internally the gearbox is vented to the condenser to allow any Freon that leaks past the high speed seal to boil off and pass to the condenser. As shown in Figure 9, this present approach has a gearbox loss of approximately 0.5 HP. This loss could be reduced to 0.3 HP by venting the gearbox to ambient air conditions. It is proposed that the double low speed shaft seal be removed and located on the high speed shaft. This would result in the removal of two low speed shaft seals at the cost of an additional high speed shaft seal. Calculations show that these seal changes result in essentially the same total seal loss. Therefore, it is predicted that a 0.2 HP savings would be accomplished by venting of the gearbox to ambient air. While 0.2 HP is not a large amount, when one considers that this is 10% of the required output power of the Rankine cycle it becomes very significant.

4.3 CONTROL POWER REDUCTION

By reducing the voltage on the compressor electric clutch and the start solenoid valve it is estimated that a savings in control power of 90 watts can be accomplished, thereby reducing the control power from 100 watts to 50 watts. While this is only a minor savings in power, only a relatively small effort is required to accomplish this saving.

4.4 MOTOR/GENERATOR IMPROVEMENT

The present induction motor/generator is 63% efficient as a motor at 2 HP output and 77% efficient as a generator with 2 HP input. The field current is approximately 300 watts. Two changes can be made to improve this unit; one is to reduce the rated size from 3 HP to 2 HP and the second is to improve the motor efficiency to 75 to 85% and the generator efficiency from 80 to 90%. While these improvements are difficult to evaluate in terms of COP, the real result would be to reduce the electrical supplemental power input during the air conditioning mode from 2200 watts to 1600 watts (a 27% saving) and increase the rated electrical output during the generating mode from 1250 watts to 1450 watts (a 16% improvement). To obtain the higher efficiency a standard motor will have to be rewound.

4.5 REDUCTION OF SYSTEM PRESSURE LOSSES

Presently there is a 3 psi pressure loss between the vapor separator and the turbine inlet and additionally a one psi pressure loss between the turbine exit and the condenser pressure. These losses can be eliminated by increasing the tubing size and designing a new regenerator. The result of the reduction of these losses to the design goal of one psi and 0.2 psi respectively will result in a cycle improvement of 2% and 3% respectively for an overall 5% system improvement.

The items listed in Section 4.0 are presented in the order of recommended preference by Barber-Nichols Eng. Co. The item of Section 4.1 has the greatest cycle efficiency improvement for dollar input. Consequently, it is recommended that if funding is limited, the improvements be funded in order of the listing above. If the items listed above are all incorporated in an improvement it is projected that the system will exceed an overall COP of 0.6, which was the original design goal.

4.6 RANKINE SYSTEM MATCHING WITH SOLLAR COLLECTOR

An additional area that should be included in follow-on activities is the matching of the Rankine system with the Honeywell collector system. Presently the unit obtains maximum rated power at 215^{o}F collector water temperature. Additionally, the pump drive is electrical. These two items must be evaluated and modified to obtain greater usage and greater payback for the solar collector system. Other additional parameters should be considered such as a variable speed drive or a belt drive in place of the present gearbox unit.

After evaluation of the optimum matching of the Rankine and the solar collector system it is essential that a study for cost reduction be conducted in order to evaluate production costs and approaches for a production Rankine cycle system.

It is hoped that Honeywell and/or NSF funding is available to carry on the development of the Rankine system to a commercialization of this approach. It is still felt by Barber-Nichols that the Rankine cycle system will be proven superior to the absorption cooling system on both a cost and a performance basis.