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SOLAR RESIDENTIAL HEATING AND COOLING SYSTEM DEVELOPMENT TEST PROGRAM

INITIAL REPORT

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NOTICE

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16. ABSTRACT

This report contains a description of a solar heating and cooling system that is installed in a simulated home at Marshall Space Flight Center. Performance data are provided for the checkout and initial operational phase for key subsystems and for the total system. Valuable information has been obtained with regard to operation of a solar cooling system during the first summer of operation. Areas where improvements and modifications are required to optimize such a system are discussed.

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TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Continued)

LIST OF ILLUSTRATIONS (Continued)

LIST OF ILLUSTRATIONS (Concluded)

LIST OF TABLES

TECHNICAL MEMORANDUM X-64924

SOLAR RESIDENTIAL HEATING AND COOLING SYSTEM DEVELOPMENT TEST PROGRAM

SUMMARY

This report describes the Marshall Space Flight Center (MSFC) solar heating and cooling facility and presents a discussion of test results obtained from late May 1974 to September 1974.

The MSFC facility was assembled to demonstrate the engineering feasibility of utilizing solar energy for heating and cooling buildings. The primary purpose of the facility is to provide an engineering evaluation of the total system and the key subsystems and to investigate areas of possible improvement in design and efficiency. The system has not been optimized. This should be considered when reviewing the energy balances presented in this report.

As will be shown, considerable information is being obtained from operation of the system and significant improvements in operation and efficiency are being achieved.

The basic solar heating and cooling system utilizes a flat plate solar energy collector, a large water tank for thermal energy storage, heat exchangers for space heating, and an absorption cycle air conditioner for space cooling. A complete description of all systems is given in Section II.

Development activities for this test system have included assembly, checkout, operation, modification, and data analysis, all of which are discussed herein. System operation is continuing, and additional modifications will be made as necessary to support continuing system improvements and upgrading of performance evaluation techniques.

Selected data analyses for the first 15 weeks of testing are included herein. Particular attention is given to investigations associated with air conditioner performance optimization. Findings associated with energy storage and the energy storage system are also outlined and conclusions and recommendations resulting from test findings are provided.

I. INTRODUCTION

Among the possible energy sources which are alternatives to fossil fuels solar energy is possibly the most pollution free, nondepletable source of all. It may be expected that the sun will ultimately be harnessed to produce electricity, synthetic liquid and gaseous fuels, and high temperature thermal energy for industrial processes. However, the most immediately realizable application of solar energy will be to heat and cool buildings, since this application will require the fewest technological advances with the least expenditures of time and money.

Currently, a major obstacle to widespread use of solar energy for heating and cooling is the initial cost of the system. The design of the system discussed herein uses conventional construction techniques and materials; however, performance, cost, and energy savings were not optimized. In addition a number of subsystems (e.g. the solar collector) are custom made in small lots which results in material and labor costs that are not representative of production hardware. For these reasons cost data for this facility are not relevant to future marketable systems and are therefore not presented.

Efforts are currently underway at Marshall Space Flight Center to investigate the performance of a solar heating and cooling system. This report describes those activities to date and presents the results of tests conducted during the summer of 1974. Activities associated with fabrication and preliminary testing on the solar house located at MSFC are described. In addition selected test data are presented and a number of conclusions are provided.

The major design and fabrication effort extended from September 1973 through May 1974. Development testing began on May 28, 1974. These tests consisted of full system operation in the cooling mode only. Testing on the originally designed system lasted through most of June. Normal testing was interrupted by a series of special air conditioner tests in late June which continued through mid-July. These tests were followed by system modifications based on test findings. Finally, from late July through August, the modified system was tested in the air conditioning mode. Testing is continuing through the winter to obtain data on system operation in the heating mode.

The objectives of these early tests were to:

1. Assemble a solar heating and cooling facility using existing materials, equipment, and assembly methods.

2. Operate the facility for checkout and development testing purposes.

3. Analyze test data to establish system operational characteristics and to evaluate performance while defining any required modifications.

4. Modify the system where feasible to improve system performance within the framework of hardware available and schedule constraints.

5. Perform additional testing to allow evaluation of modifications.

II. FACILITY DESCRIPTION

A. General

The system and subsystem descriptions and control logic given in this section represent the original configurations that were effective at the time of test initiation on May 28, 1974. All significant subsequent alterations are discussed in applicable paragraphs of Section III.

The solar facility (Figs. II-1 and II-2) is located on the MSFC complex adjacent to Building 4619, at the corner of Rideout and Fowler roads. Three house trailers have been joined to form a single habitable structure. The test system is composed of the solar collector, an absorption air conditioning system with its associated cooling tower, a hot water heat exchanger for winter operation, a water tank for energy storage, and the habitable quarters (Figs. II-2 and II-3). These major subsystems are interconnected by a system of water pipes. The control network is fully automated to allow continuous operation.

A simplified operational description of the system follows, and detailed descriptions of the various subsystems and evaluations of their performance are provided in subsequent subsections.

B. Operational Description

As the sun heats the solar collector, water flowing through the tubes in the collector is heated. This hot water is returned to the energy storage tank. When the home thermostat requires either heating or cooling, hot water is directed from the storage tank to a three-way valve that diverts flow to either the heater or air conditioner. In the cooling mode hot water is used to power a 3-ton absorption air conditioner that produces cooling. Air from the house is blown over the cold air conditioner coils from which it is directed via ducting into the individual rooms of the house. In the event water from the storage tank is not sufficiently hot to activate the air conditioner, the water temperature is elevated to an acceptable level by auxiliary heaters. In the heating mode the hot water simply passes through a heating coil. In the event the water is too cool to provide sufficient heating, duct electrical resistance (strip) heaters activate to warm the air directly. The same blower used to circulate air across the cooling coil provides flow for the heating coil/heater combination. In both modes the thermostat is the central control device used to activate the automatic

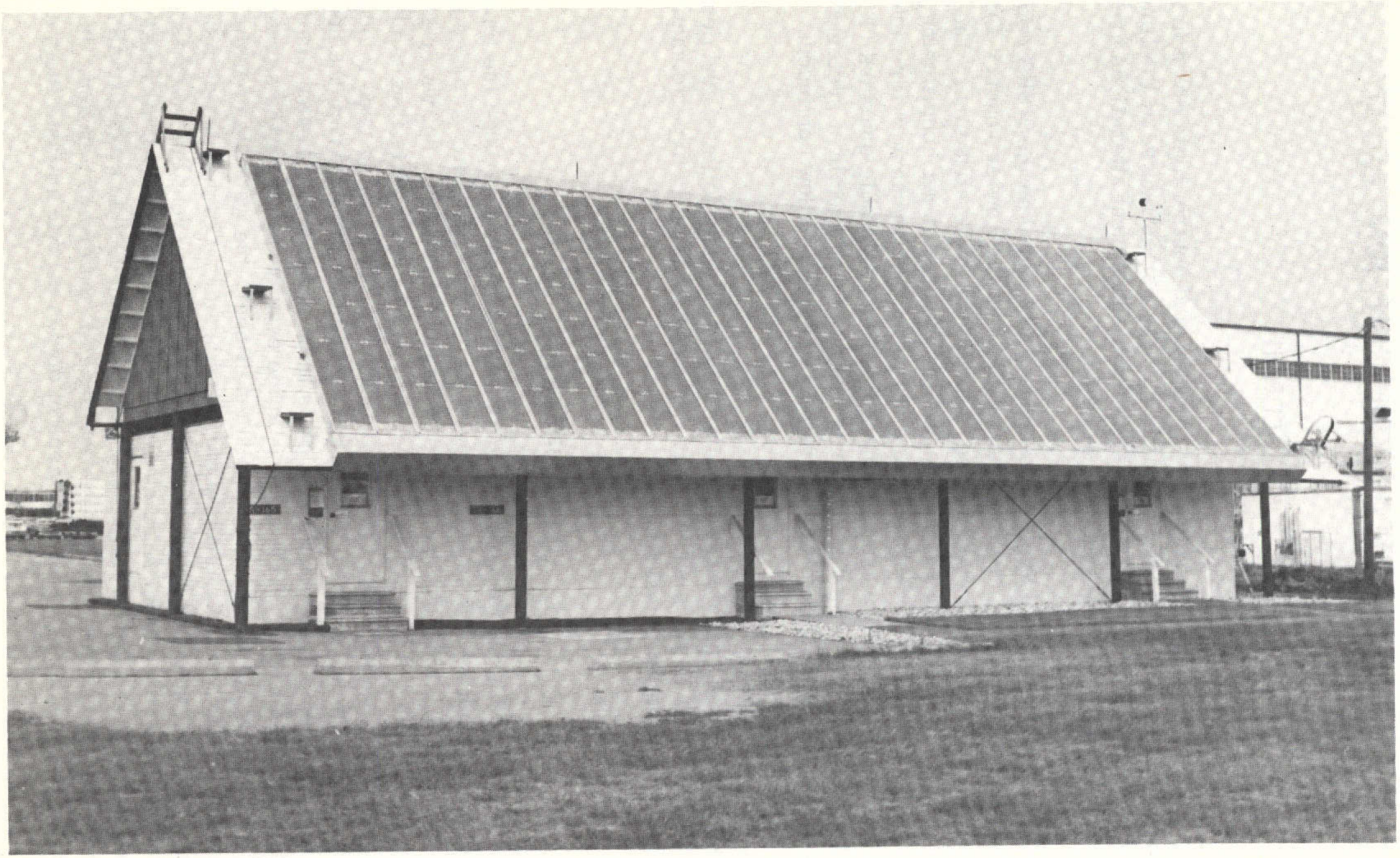

Figure II-1. Photograph of solar house.

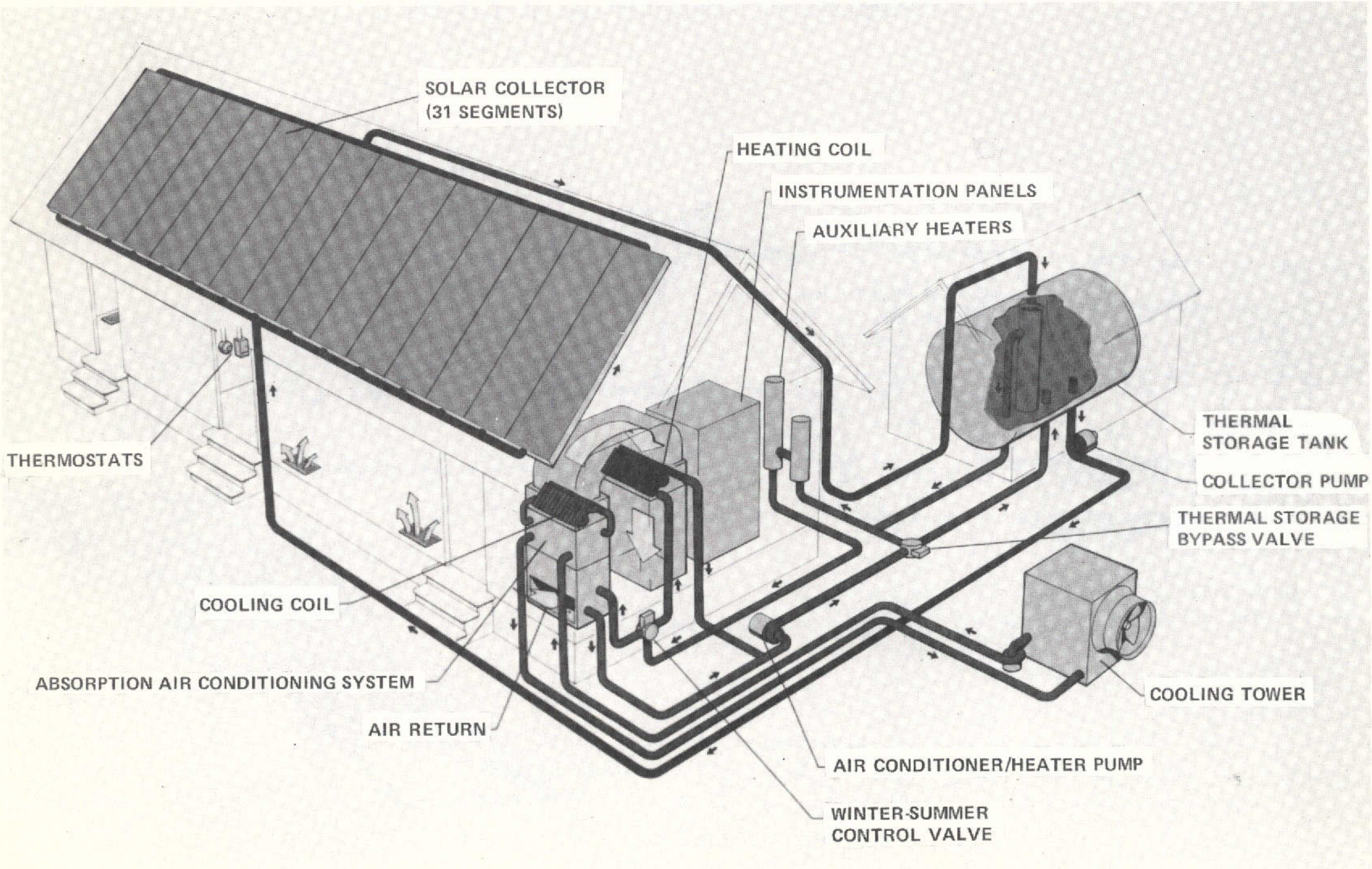

Figure II-2. MSFC solar house site schematic.

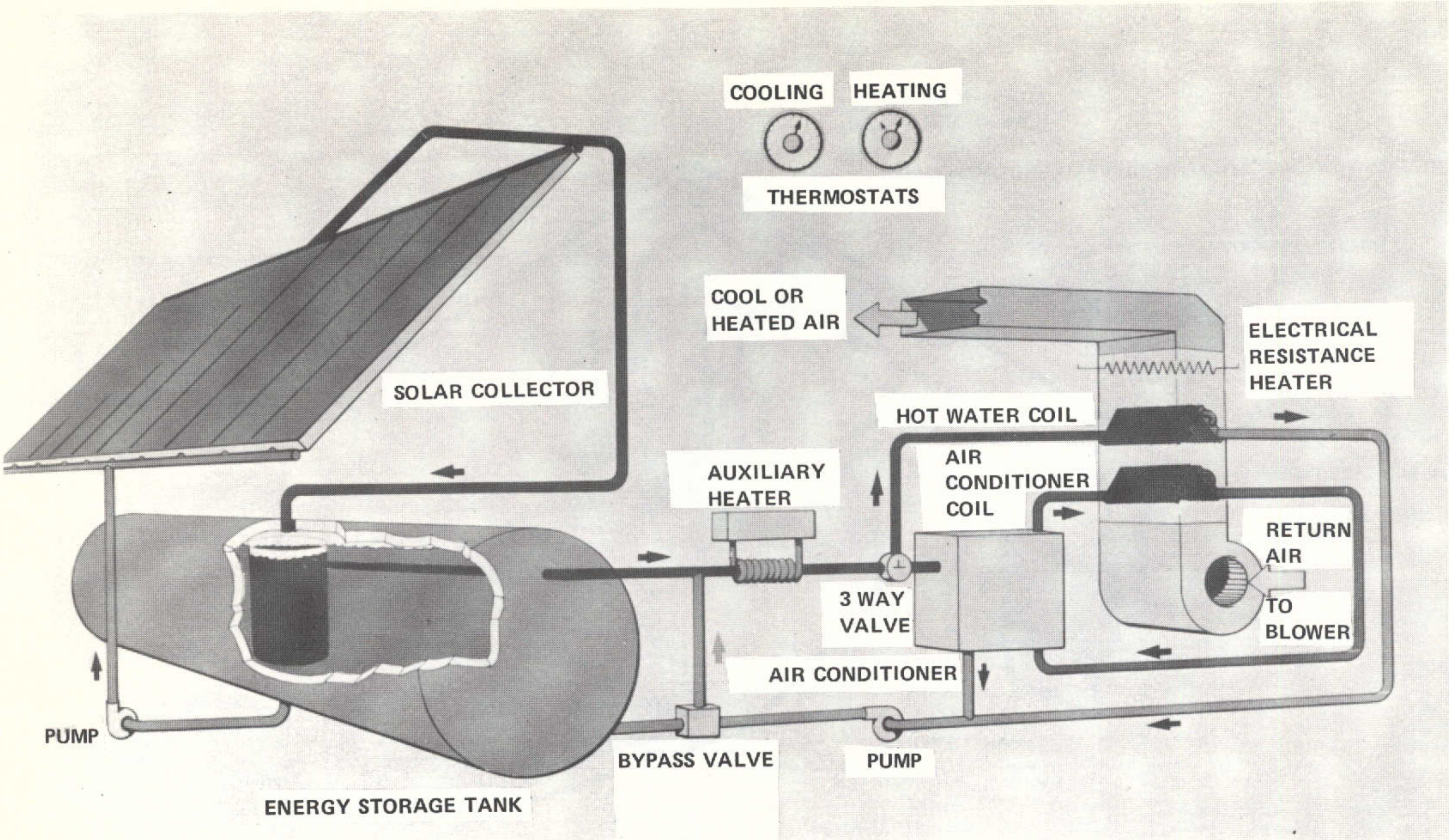

Figure II-3. Simplified system schematic.

fluid control system. The collector control system is also automatically activated and deactivated based on various logic parameters that will be discussed in a subsequent section.

C. Solar Collector

The heart of the test system is the solar collector. Although the projected area of the roof covered by the collector is 1475 ft^2, the collector module surface area is only 1302 ft^2. Because of edge shading only a 1218-ft^2 maximum solar field-of-view is possible from the collector (this excludes shadowing by the wire matrix used to support the Tedlar cover). The collector is situated on the south side of a 45 deg pitched roof with a clear unobstructed view of the southern sky. The collector consists of 31 segments; each segment (Fig. II-4) consists of an aluminum tray containing 7 collector heat exchangers joined in series. These heat exchangers are fabricated from 1100 series aluminum sheets. The heat exchangers are roll-bond type panels formed from two sheets that have been bonded together and expanded in selected areas to form water passages (Fig. II-5). Each heat exchanger is 2 ft by 3 ft in overall size. The heat exchanger surfaces are electroplated with a black nickel material, which has initial approximate solar absorptance and thermal emittance properties of 0.92 and 0.06, respectively. Figure II-6 presents the acceptance criteria used along with a number of actual data points. Although the MSFC black nickel coating has excellent collector initially applied coating properties, it is unstable in the presence of moisture. For this reason, an active purge is incorporated in the collectors to shield the coating from high ambient humidities. The backs of the heat exchangers are insulated using 6-in.-thick household-type fiberglass battens.

The coated outer face of the heat exchangers is covered with wire-reinforced Tedlar plastic. The Tedlar allows transmission of solar energy while retarding convective heat losses from the coated collector surface of the heat exchanger. The Tedlar cover also provides a "greenhouse" effect by trapping solar energy. An added benefit of this cover is the protection it provides against the effect of dust, sand, hail, snow, ice, and other possibly damaging matter. The Tedlar cover is composed of a single 2-mil sheet covering 27 of the 31 segments. The last four segments on the east end of the roof are covered with 4-mil Tedlar. The supporting wire is 0.105 in. diameter galvanized iron (i.e., common welded fence wire) forming 2 in. by 4 in. grids.

The collector heat exchangers are held in the aluminum trays by wooden strips that form channels on two sides of the exchangers. Teflon clips prevent the edges of the heat exchangers from touching the sides of the aluminum trays, thus preventing thermal shorts. The assembled collector weight is approximately 4600 lb, adding 10 percent to the normal roof loading.

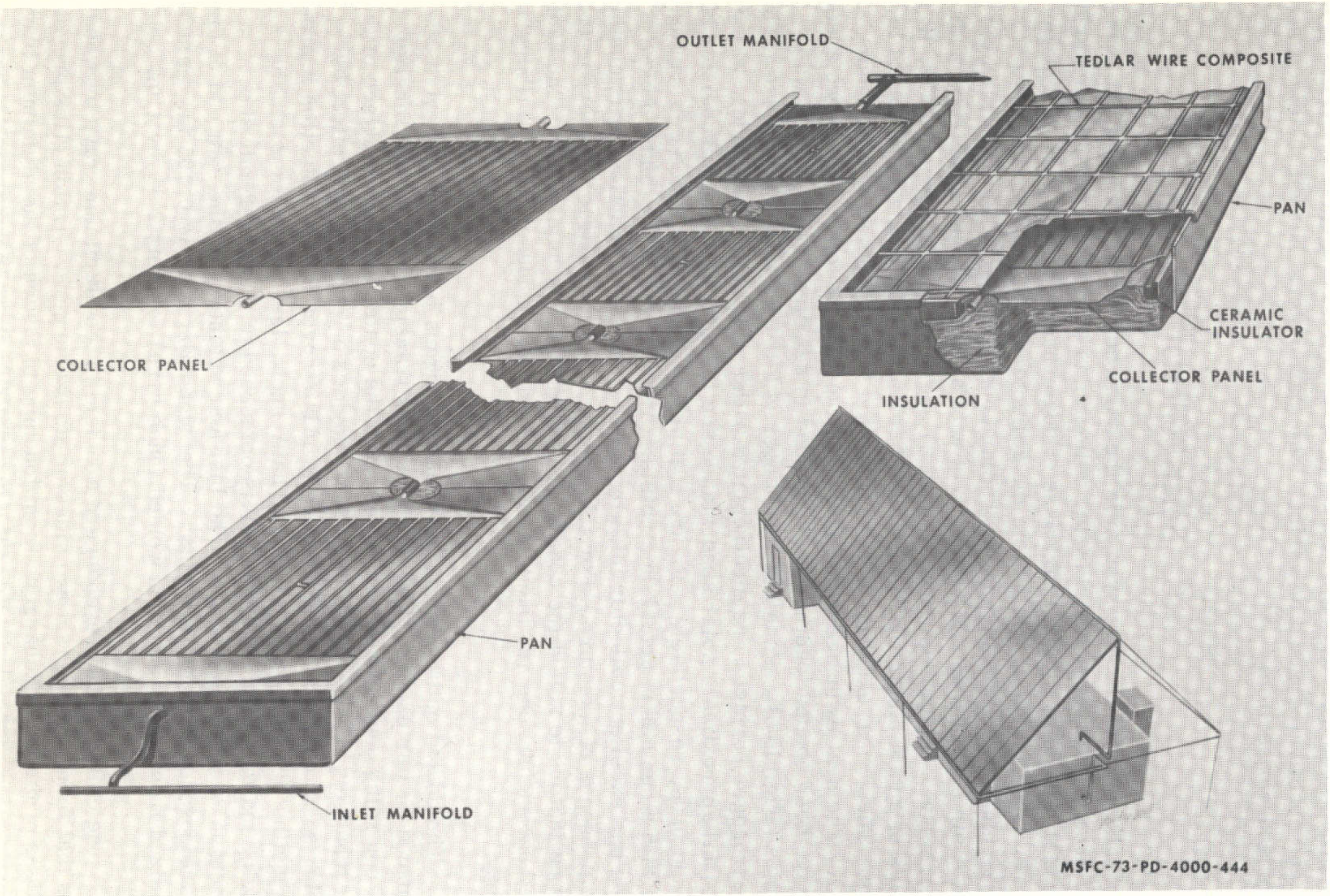

Figure II-4. MSFC collector design features.

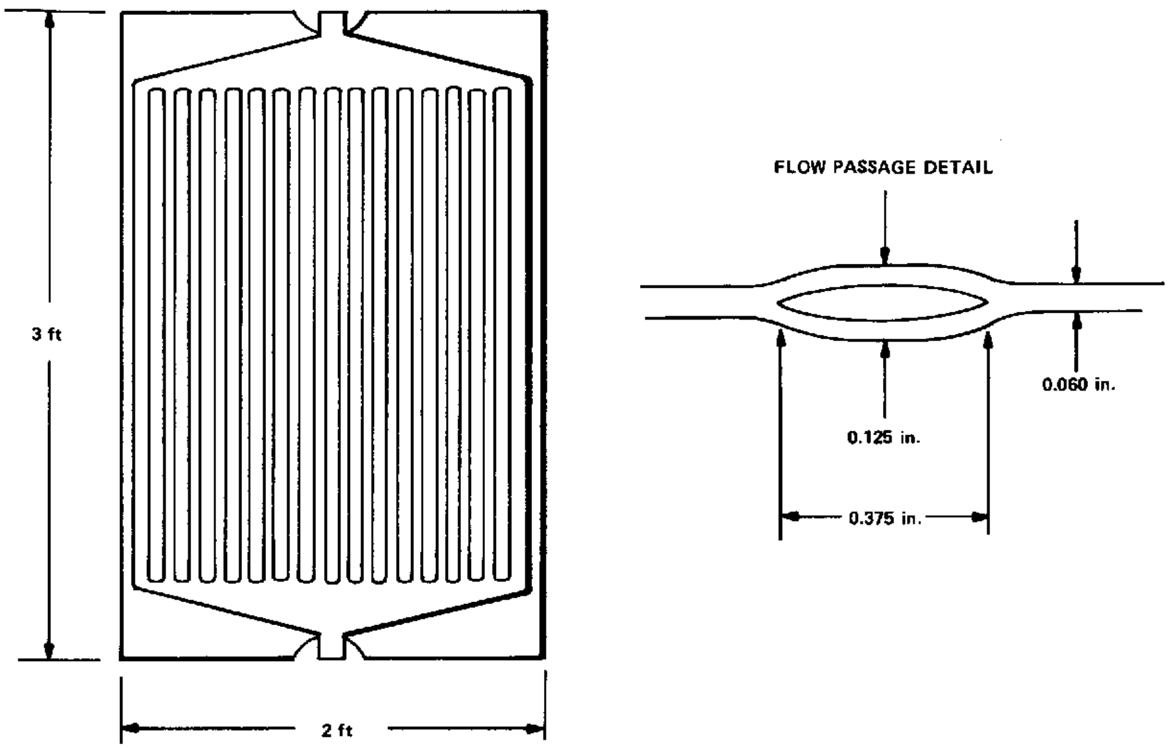

Figure II-5. Roll-bond panel design.

D. Collector Purge

The collector is provided with a dry air purge system to protect the black nickel coating which automatically maintains the humidity level between 15 and 40 percent relative humidity.[1] Two surplus Skylab life support system gas compressors are mounted in parallel to provide a 30 cfm (i.e., 15 cfm each) flow rate capability. The moist return air from the collector flows into a desiccant bed which absorbs the water from the gas stream returning the dried air to the inlet manifold located adjacent to the collector fluid inlet manifold. The desiccant is regenerated continuously by a 700-W heater. The purge gas enters the collector assembly in the center of the insulation portion of the segment. A similar manifold returns the moist air from the top of the collector at the roof apex to the desiccant/compressor assembly located under the roof.

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1. Significant advances have been made in breadboard testing in the area of collector humidity control devices. A completely passive system has been designed and satisfactorily tested and will be incorporated in a second generation collector for long life evaluation with a resulting system power reduction.

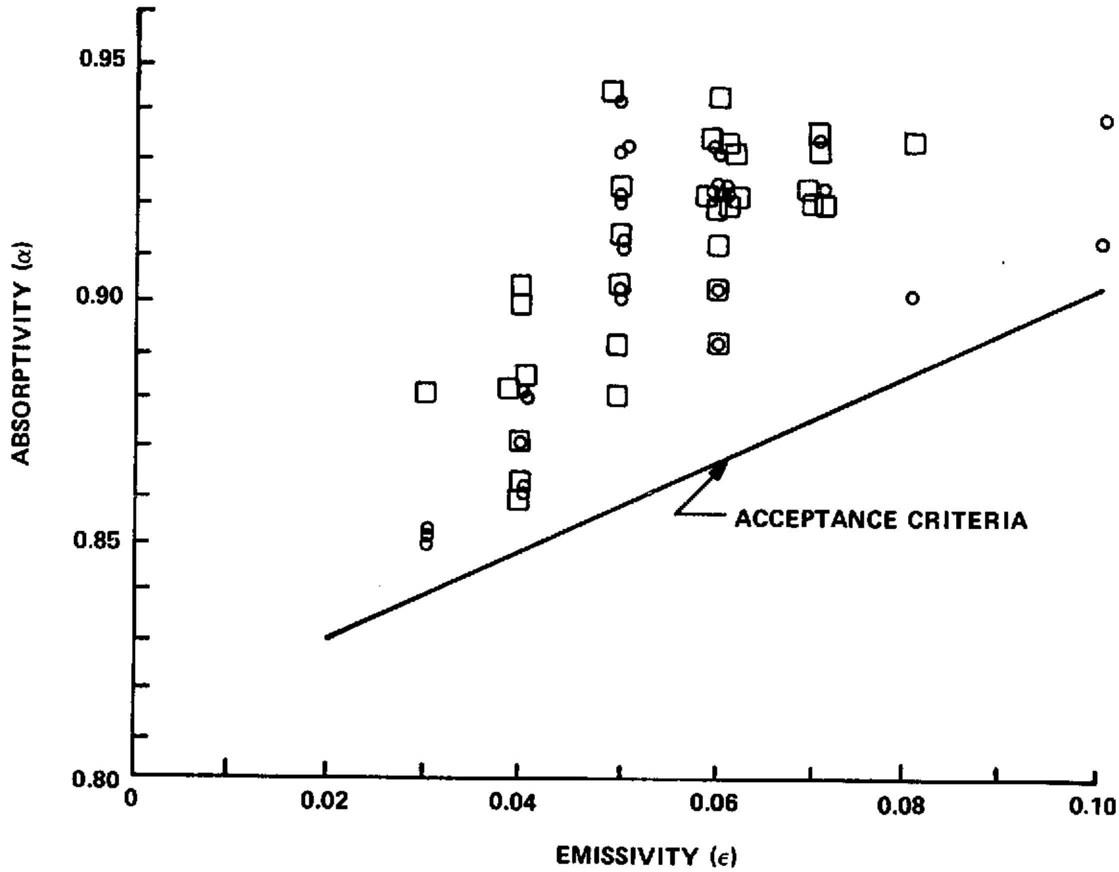


Figure II-6. MSFC collector coating acceptance criteria.

E. Piping System

The interconnecting piping system basically consists of two hot water loops; the collector and air conditioner/heater (AC/Htr) loops, which are actually common since they both return to the water tank, and the cold water loop, which connects the cooling tower to the air conditioner. The collector loop contains a 1/3-hp centrifugal pump that circulates deionized water having approximately 750 ppm by weight sodium chromate additive for inhibiting corrosion. Water is circulated from the tank to the collector inlet manifold. From the inlet manifold, the total water flow is evenly divided between the 31 segments, returned to a common outlet manifold, and then to the water tank.

All piping with the exception of the collector heat exchangers, which are aluminum, and the AC/Htr plumbing in the trailer area, which is stainless steel, are made of galvanized iron. The collector supply and return lines are 2 in. in diameter. The AC/Htr loop has 1 in. diameter galvanized iron pipe except in the immediate vicinity of the house where the AC/Htr loop is constructed of 1 in. diameter stainless steel tubing. Stainless steel tubing was used because of ease in routing and availability at MSFC. Chromium and copper are also present in the fluid loop in the AC control valving and the hot water heat exchanger, respectively. The cooling tower supply and return lines are formed from galvanized iron pipe. Identical 1/2-hp centrifugal pumps are used to circulate fluid through both the cooling tower and AC/Htr water loops. Initial flow rates of 31 gpm through the collector, 11 gpm through the air conditioner, and 10 gpm through the cooling tower were baselined. The pumps were readily available, "off-the-shelf", "low bid" equipment and as such are not performance optimized. Also no special effort was expended in minimizing the system flow resistance.

F. Control/Logic System

An electromechanical logic system was used to control flow initiation and termination to the collector. Similar systems were used to protect the water tank from inadvertent overpressurization and to control fluid and gas flow in the air conditioning and cooling tower loops. Automatic flow bypasses were also incorporated in the air conditioner and heater hot water loop.

1. Collector/Energy Tank Loop. Before the collector loop water pump can be activated, a number of logic checks must be satisfied. The water tank ullage pressure must be less than 25 psig and the collector surface temperature cannot exceed 250° F. In the automatic mode, if the two previous conditions were satisfied and the collector surface temperature was at least 5° F greater than the water tank temperature but not more than 50° F greater, the collector pump would activate for 5 minutes. However, if at any time one of the first two conditions failed to be satisfied or, after the 5-minutes have elapsed, any of the last two conditions are violated, the pump operation will terminate automatically. To operate the pump in the manual mode only the first two conditions must be satisfied. Figure II-7 depicts a flow diagram of this logic.

A protection system in the water tank forces the tank vent valve open above 30-psig ullage pressure (Fig. II-8).

2. Air Conditioner/Heater Loop. In the air conditioning loop, the cooling mode is selected manually by proper positioning of a switch on the thermostat. The selection of the cooling mode positions a three-way valve allowing

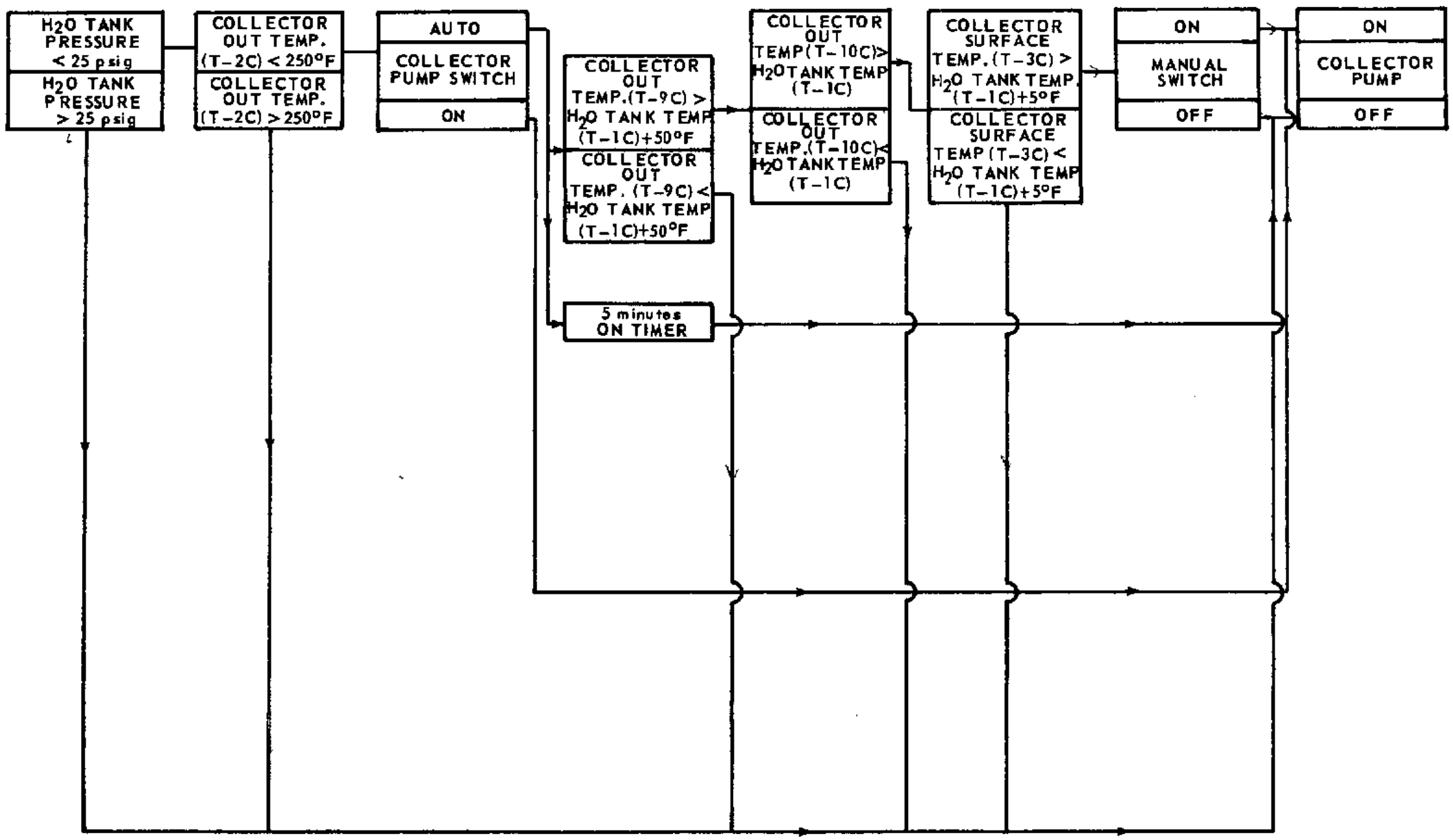


Figure II-7. Collector loop logic.

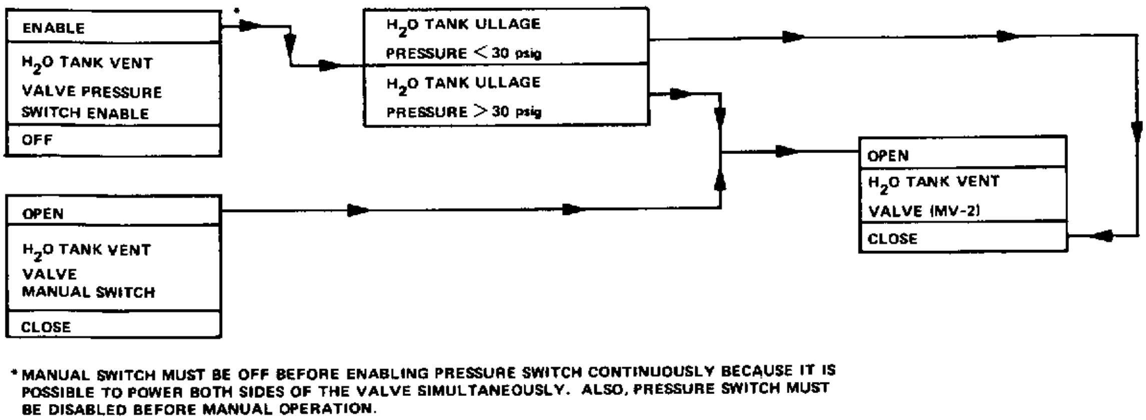


Figure II-8. Energy storage tank logic.

hot flow to the air conditioner. In the event the water returning to the water tank from the air conditioner is warmer than the tank water, a tank bypass valve automatically diverts flow around the water tank to the water heater inlet. In this mode, all the heating necessary to drive the air conditioner comes from two electrically powered hot water heaters.

When the thermostat inside the dwelling is set on the cooling cycle and air temperature within the dwelling increases to the upper thermostat set point, the system is activated as follows (Fig. II-9). The cooling tower pump and fan and the air conditioner fan are energized, the air conditioner loop pump is energized and supplies 11 gpm of water to the system (flow is preset by hand valves HV-11 and HV-12), and hot water then passes through a filter into two heaters (HTR-1 and HTR-2). These heaters are used to either provide all the energy necessary in the tank bypass mode or to supplement energy from the storage tank in the storage tank mode (see paragraph F.4). The hot water then flows into motor valve MV-3 which, on the summer thermostat setting, directs 100 percent of the hot water flow to the air conditioner. To limit the energy input to the generator thereby avoiding the "prevent cutoff" (to be discussed later), a system of flow bypass valves is provided, and they operate as follows. If the control temperature at T-8C[2] is between 210° F and 220° F, all flow passes through the air conditioner generator; if the inlet water temperature at T-8C increases to 220° F, MV-4 opens and 3.7 gpm (flow is initially set by regulating hand valve HV-14) is bypassed around the unit; if the

2. A "C" following instrumentation parameters indicates use for control only.

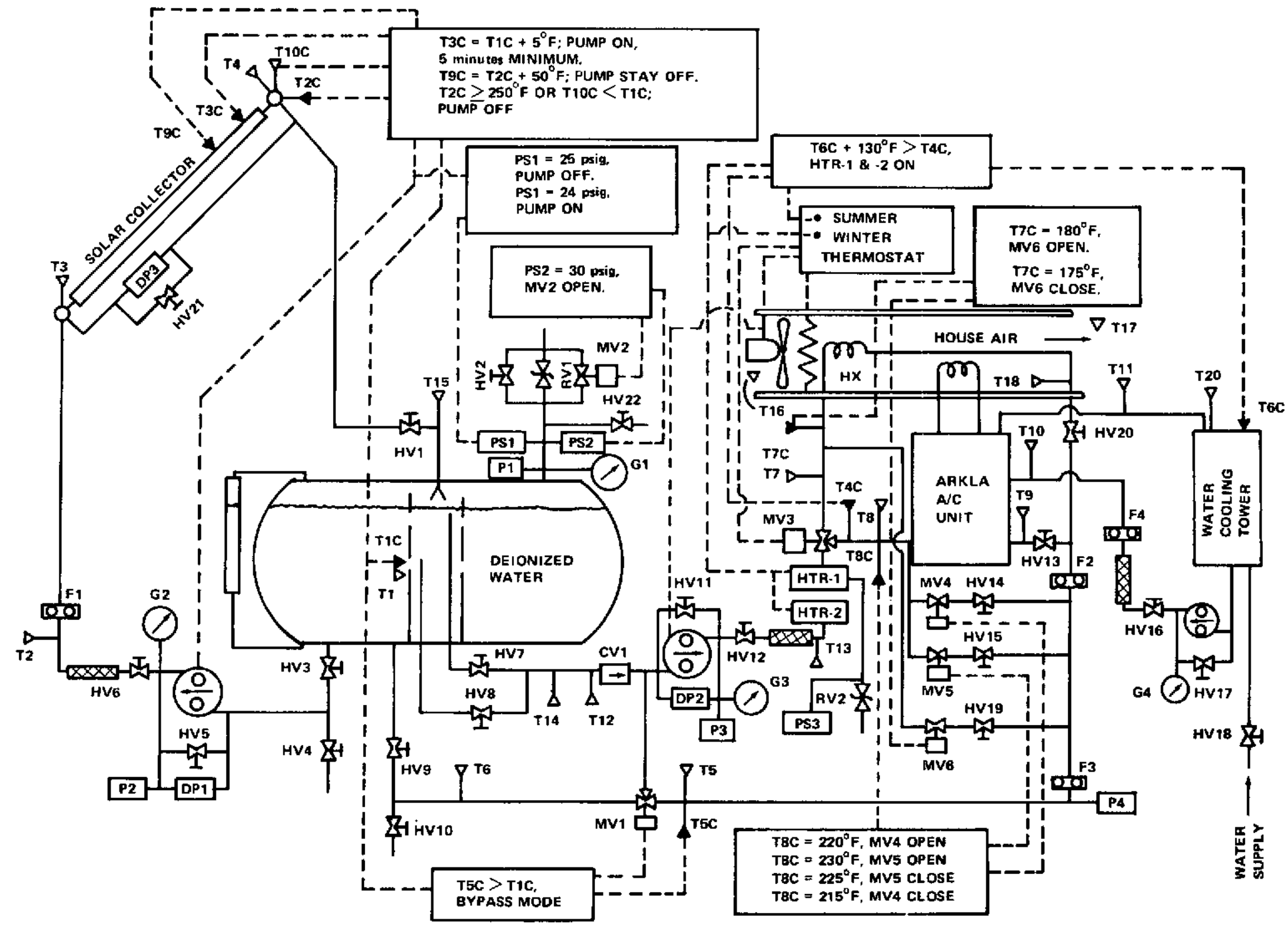


Figure II-9. Fluid system valving schematic.

temperature at T-4C rises to 230° F, MV-5 opens and allows an additional 1.8 gpm (all flow rates were set initially by regulating the appropriate hand valve in the flow loop) to bypass the air conditioner for a total bypass flow of 5.5 gpm. The temperature at T-4C will not exceed 250° F because of collector control logic cutoffs. Flow through the ARKLA unit generator is measured by flowmeter F-2 and total flow is measured by flowmeter F-3, the difference of which gives bypass flow. The hot fluid then passes through HV-9 into the thermal storage tank. If the temperature at T-8C falls to 225° F, MV-5 closes and reduces the bypass flow to 3.7 gpm. When the temperature at T-4C drops to 215° F, MV-4 closes and allows full flow to again be directed through the air conditioner generator and then through MV-1 back to the thermal storage tank.

3. Bypass Mode Flow. Bypass flow is triggered when the return temperature from the air conditioner measured at T-5C is greater than the temperature within the thermal storage tank as measured at T-1C; MV-3 directs 100 percent of air conditioner pump flow to the thermal storage tank bypass. Check valve CV-1 prevents the flow of water from the storage tank. In this mode of operation, all heat required to operate the air conditioner is supplied by electrical energy in the heaters (HTRS). A total of 18 kW is provided by 13.5-kW and 4.5-kW calrod units. Control of electrical heater HTRS is independent of the operational mode; i.e., it may be in operation either in the bypass or storage tank flow mode depending upon storage tank water temperature and outside wet bulb temperature. If the sum of cooling tower wet bulb temperature as measured by T-6C + 130° F is greater than the temperature at T-4C, heaters HTRS will be energized to supply heat. If the sum T-6C + 130° F is less than the pump outlet temperature as measured at T-4C, heaters HTRS will be de-energized. In the bypass mode, the cycling situation described above will continue until the temperature within the storage tank at T-1C reaches a value greater than T-5C, at which time MV-1 will position to provide 100-percent flow to the storage tank. When the air temperature in the dwelling decreases below the thermostat set point, the air circulation fan, AC/Htr water pump, and the cooling tower water pump and fan are simultaneously deactivated. See Figure II-10 for a flow diagram of this logic system.

4. Heating Mode. In the event cool weather operation requires heating, manual selection of cooling on the wall-mounted thermostat causes the three-way heating/cooling valve (MV-3) to divert flow from hot water to the air conditioner to hot water flow to the winter heat exchanger. All subsequent operations are automatic with the logic system providing a nominal 3.5-gpm bypass (MV-6 open) around this heat exchanger should water temperatures rise above 180° F. In the event water temperatures from the tank are too low to provide sufficient heating, three electrical strip heaters located in the gas flow duct are activated. These heaters are cycled on in three stages at 2-deg increments below the thermostat set point to give a total of 20-kW heating capability (Fig. II-11). Since this loop was not activated during these tests, no discussion of tests results is given. However, in the future, tests are planned to verify this system.

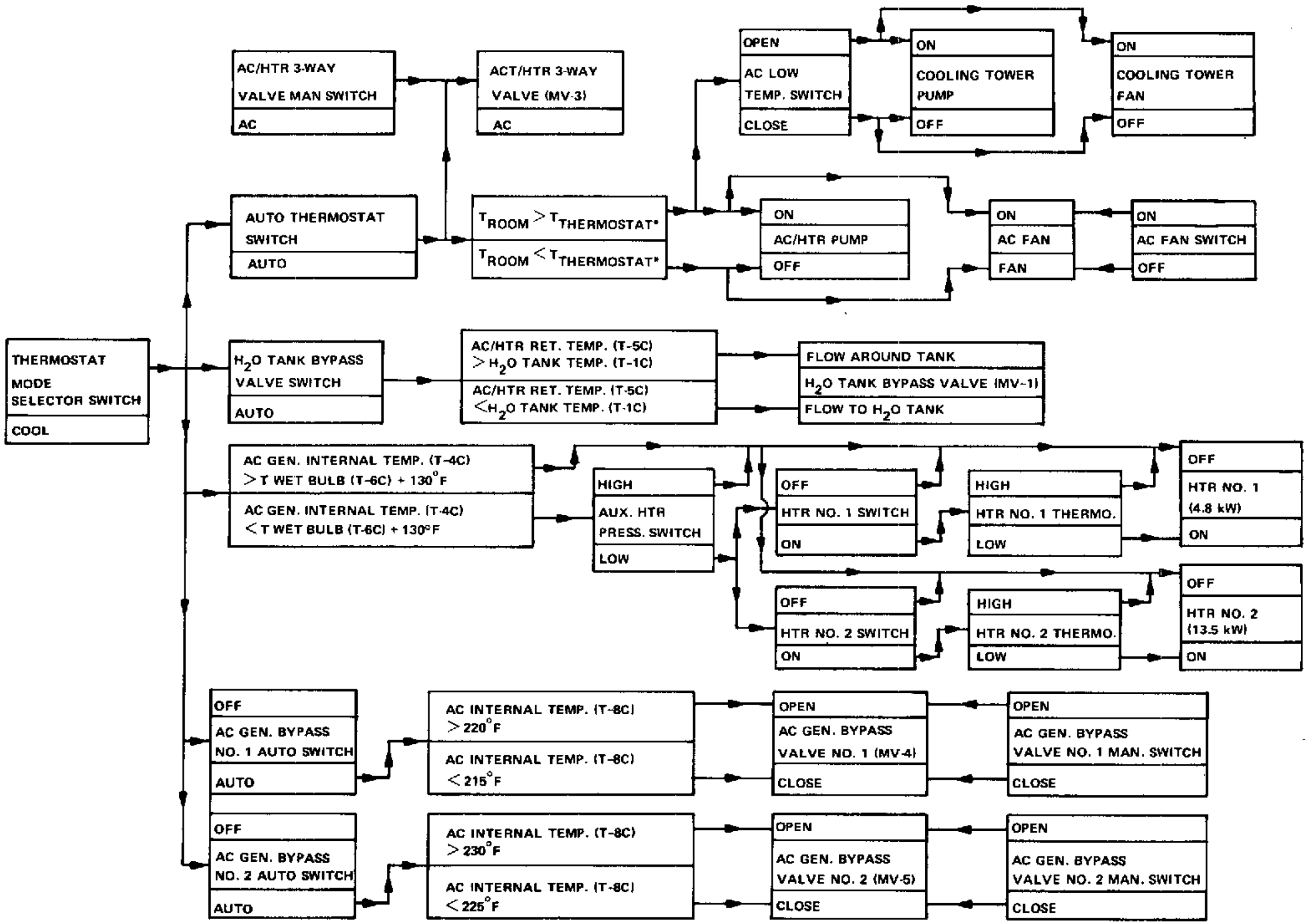
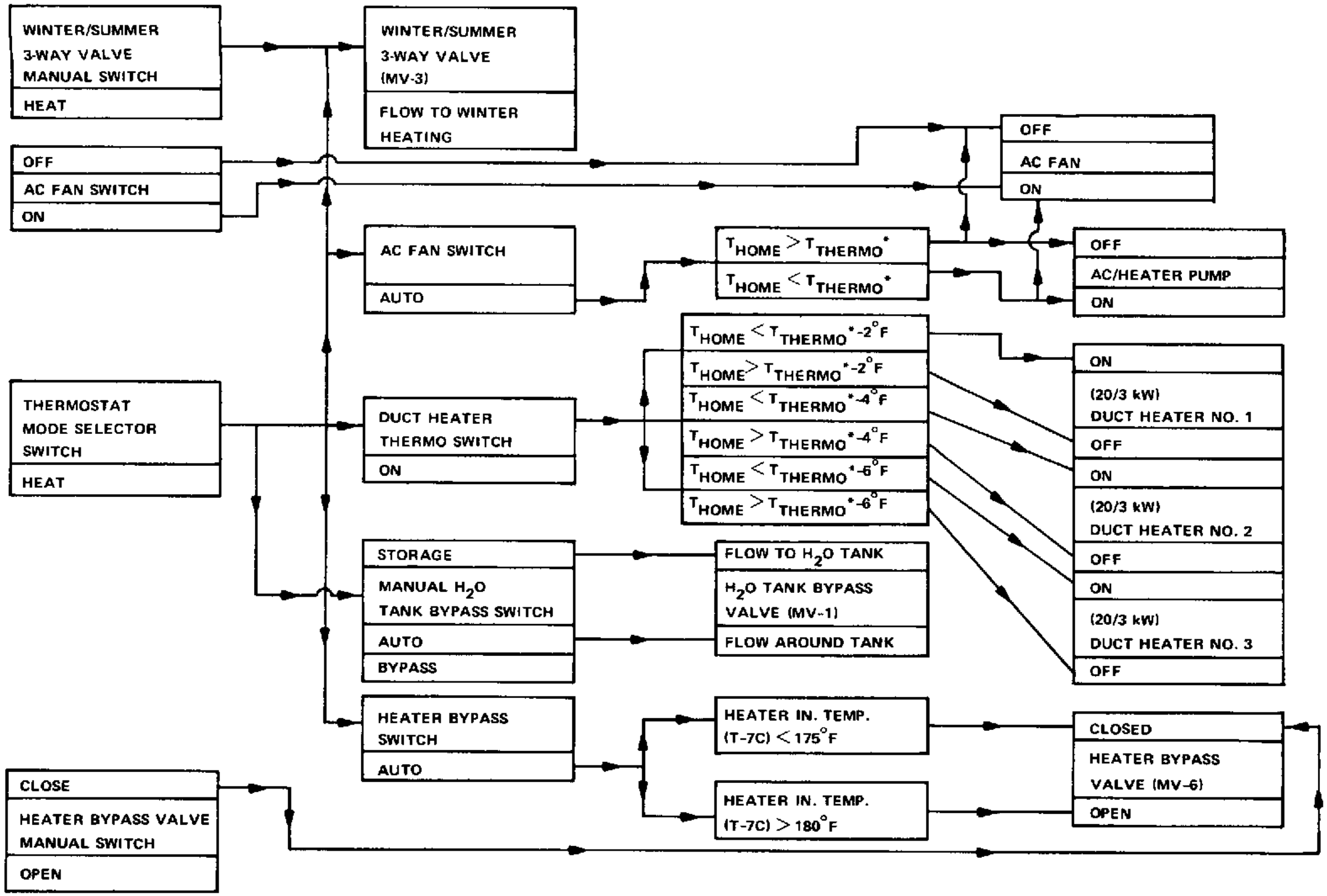


Figure II-10. Original cooling mode logic.



*NORMALLY 68°F FOR HEATING AND 76°F FOR COOLING.

Figure II-11. Heating mode logic.

G. Trailer Complex

The trailer complex, which makes up the "dwelling" to be cooled and heated, is constructed of three surplus office-type house trailers, two 10 ft by 58 ft and one 12 ft by 30 ft, secured together with interconnecting doors to form a 1520 ft^2 floor plan (Fig. II-12). This complex is situated with its long dimension parallel to an east-west line such that one face of the roof faces due south. Since trailer construction does not provide sufficient strength to support a roof structure of even minimum structural requirements, a free standing roof was constructed over the trailers to support the solar collector subsystem and associated plumbing. The roof, which is supported on fourteen 6 in. diameter steel posts set in concrete, has a 45-deg slope and utilizes trussed rafter construction with rafters placed on 24-in. centers.

Although the roof provides a shading effect, no special thermal protection has been intentionally provided to the trailers. Metal skirts are located around the trailer complex periphery at its base; these inhibit air flow under the trailer but were provided for aesthetic reasons. The outer walls and north roof are covered with commercially available white latex paint. The air conditioning ducts are insulated with common 1.5 in. thick fiberglass insulation and are routed under the trailers to the various rooms.

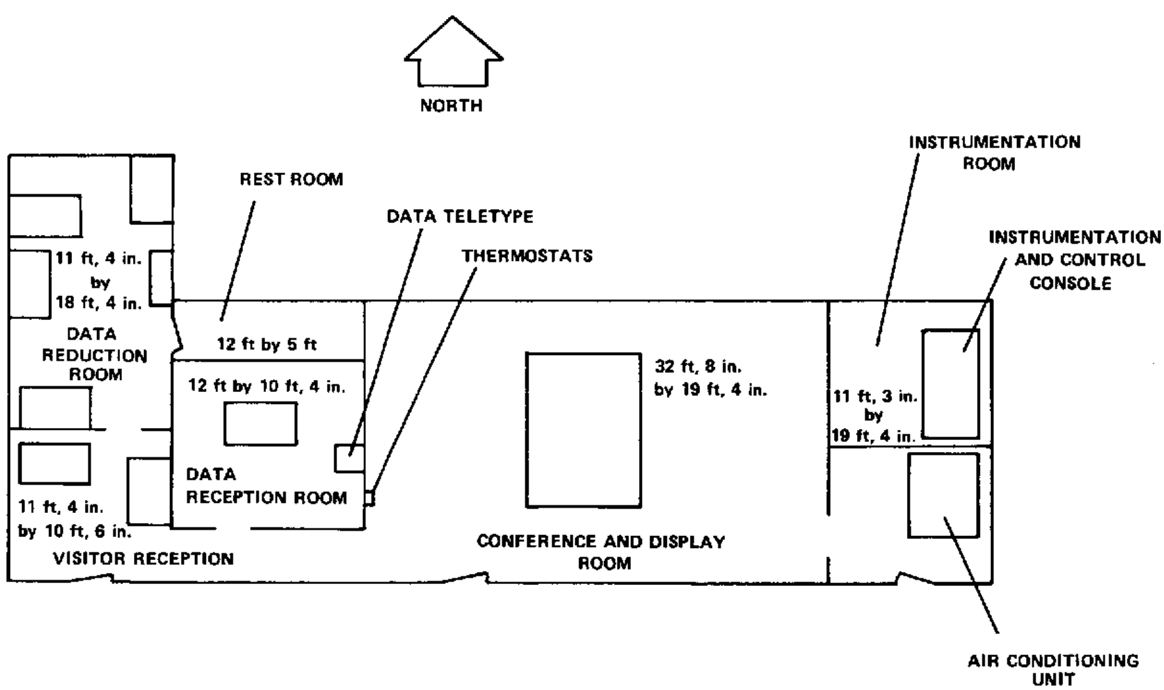


Figure II-12. Solar house floor plan.

H. Energy Storage Tank

The energy storage subsystem is shown in Figure II-13. The basic component is a 4700-gal aluminum storage tank that contains 3550 gal of water. The tank is housed in a wooden "house-like" structure for weather protection. Common household insulation has been installed to achieve a minimum thickness of 12 in. of insulation on all sides (Fig. II-14). The tank, which is a space program surplus item, has 0.5 in. thick walls and is fabricated of 5154 aluminum.

The tank has been modified internally in an attempt to provide the hottest water to the AC/Htr loop while providing the coldest fluid to the collector inlet. To achieve this an 8 in. diameter pipe was welded to the bottom of the tank (Fig. II-15). The hot water from the collector is dumped into the top of the standpipe, while the water supplied to the AC/Htr loop is also taken from the standpipe just below the liquid level. The cold water from the AC/Htr loop is dumped back into the bottom of the tank. The collector supply is taken from the bottom of the tank. Flow to and from the standpipe into the tank proper is through four 2 in. diameter holes equally spaced at each of two levels in the standpipe.

Initial filling of the tank caused the water level to be about 60 in. from the tank bottom. When the collector is not running, water drains from the collector and its manifolds. The water level of the collector system is 4 in. below the inlet manifold.

I. Air Conditioner

Cooling is provided to the trailers by an absorptive air conditioner. This air conditioner is a commercially available unit which was driven by a gas fired generator, having been modified to incorporate a hot water driven generator. The hot water boils the water vapor from the aqueous lithium bromide solution in the air conditioner. The boiling creates a "coffee pot" type percolation which in turn creates circulation in the unit. This percolation allows the air conditioner fluid medium to flow without requiring a mechanical pump and external electrical power. This particular unit was sold commercially by the Arkansas Louisiana Gas Company (ARKLA) through 1967 but its commercial sale has since been discontinued. It was initially designed to be heated directly by a gas fired boiler, but has been modified to incorporate a hot water heat exchanger. This unit has been downgraded from a 4.1 ton unit to provide approximately

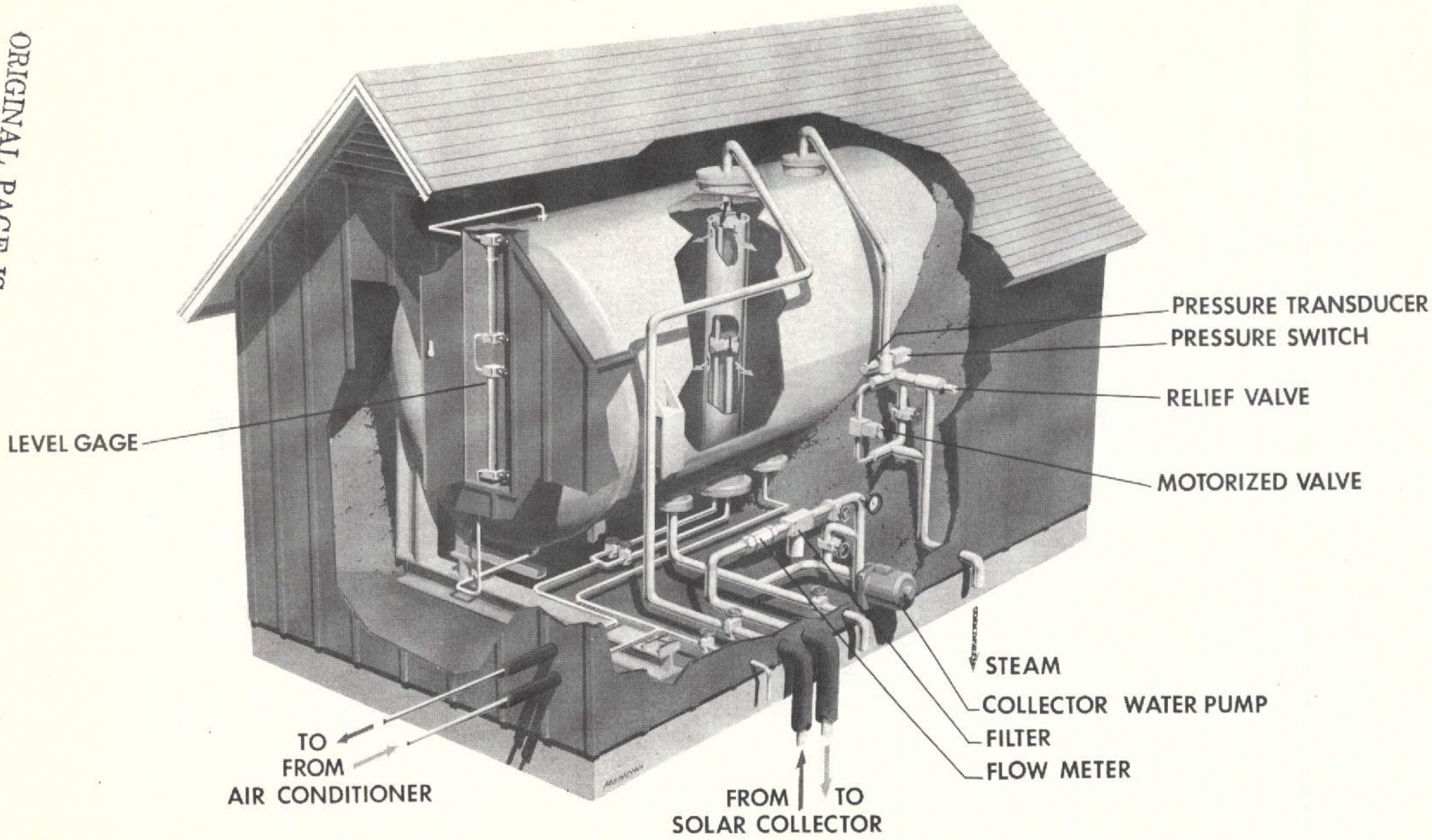


Figure II-13. Energy storage tank arrangement.

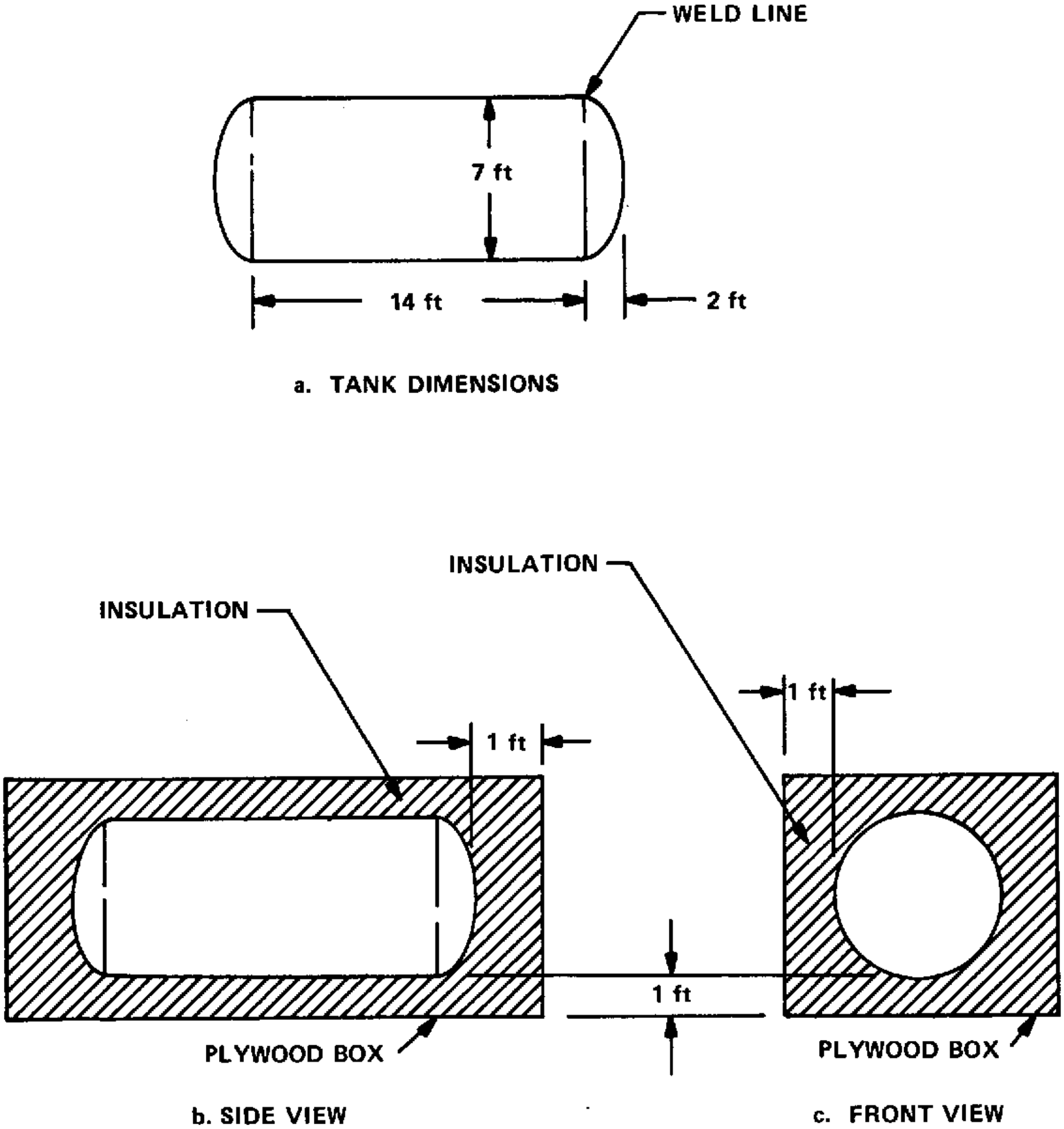


Figure II-14. Energy storage tank insulation scheme.

3 tons of cooling capacity. Hot water is pumped from the energy storage tank into an interface heat exchanger referred to as a generator. The incoming water temperature may be raised using auxiliary heaters in the event the water tank temperature is inadequate.

The lithium bromide/water cycle (Fig. II-16) operates under a reduced pressure, approximately 58 mm Hg at the generator to 8 mm Hg in the evaporator (cooling coil). After the water vapor leaves the mixture in the generator it

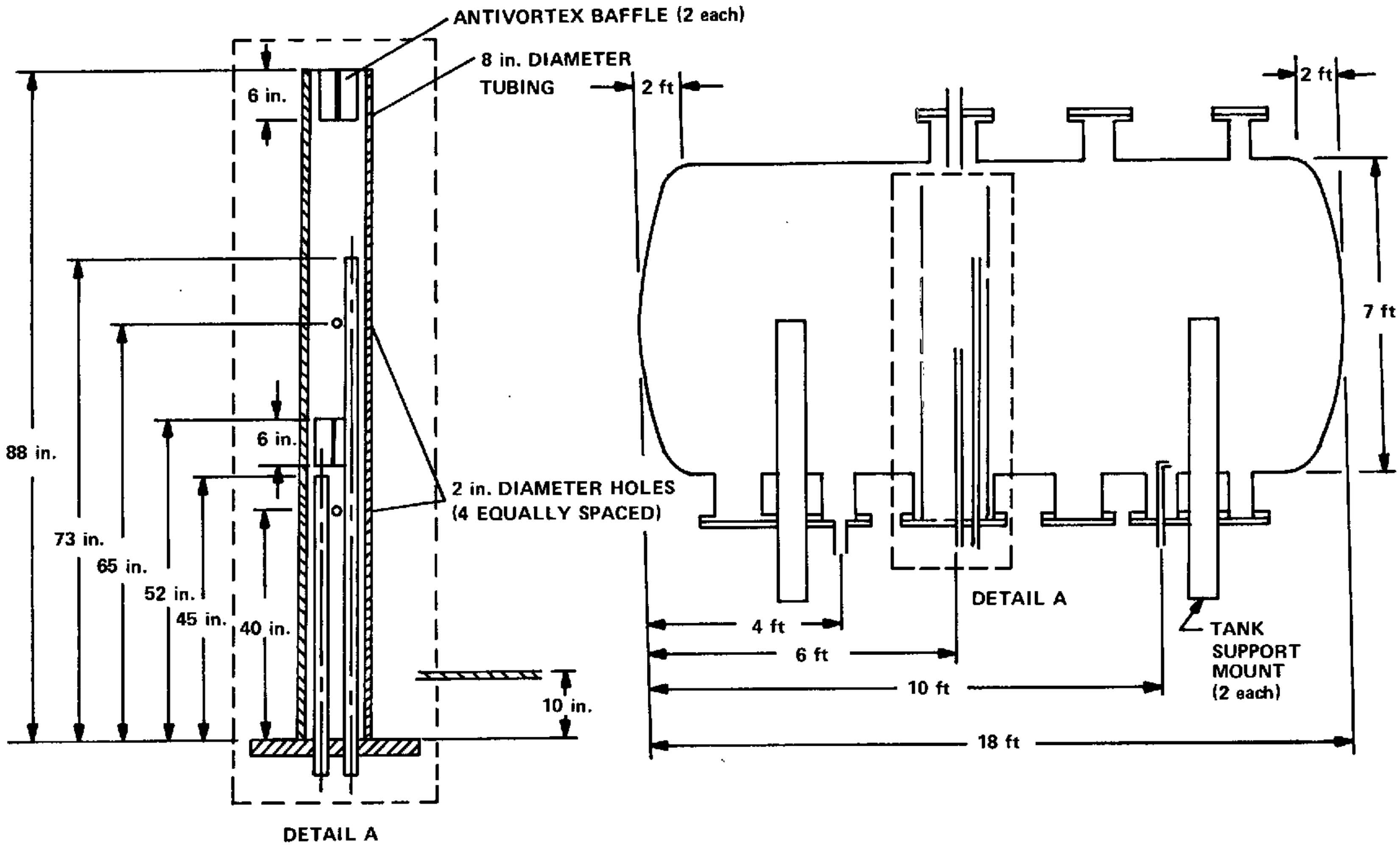


Figure II-15. Energy storage tank details.

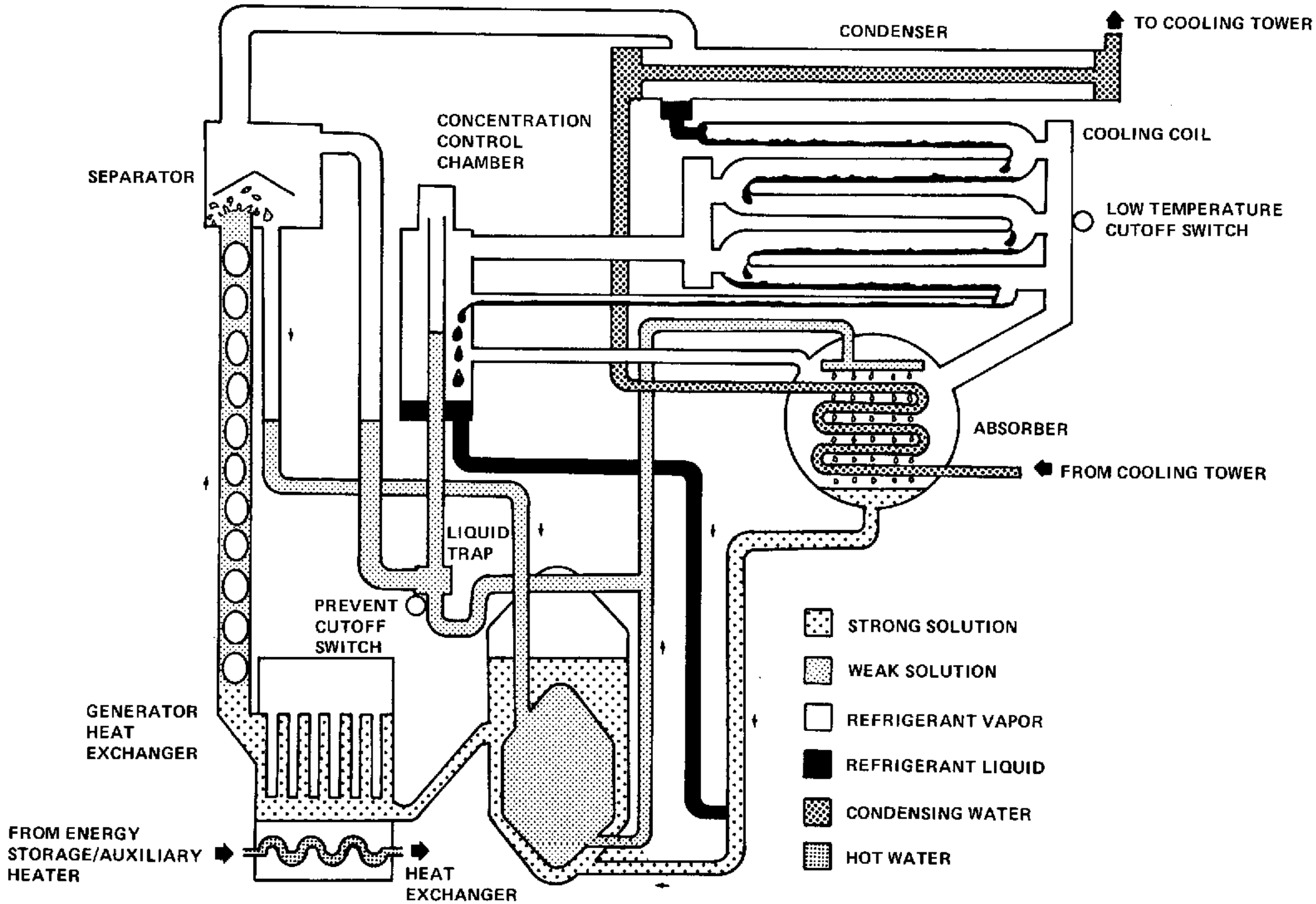


Figure II-16. Lithium bromide absorptive air conditioning system.

passes through the condenser where it is cooled to near ambient temperature and heat is rejected to the cooling tower water loop. Upon leaving the condenser the vapor passes through the expansion valve where it is reduced in pressure and temperature. After passing the expansion valve, the refrigeration effect is obtained in the evaporator by water boiling at the low pressure which normally provides an evaporator surface temperature of from 36 to 50° F. Heat rejection to the evaporator is accomplished by a fan blowing air across the coils at approximately 1200 cfm. The cooling tower provides a heat sink for the air conditioner to cool the water vapor in the condenser and in the absorber (immediately downstream of the evaporator) where the water vapor is reduced to the liquid state. In this state it has a strong affinity for the lithium bromide and consequently recombines. Special protection circuitry is provided by the manufacturer to assure that the water vapor is not overchilled thereby allowing unwanted ice formation. An additional protection system is provided to avoid emptying a liquid trap between the generator and absorber. These protection systems will be referred to as the "low temperature cutoff" and the "prevent cutoff," respectively. In addition to the protection circuits a low temperature water bypass valve is provided to avoid allowing unacceptable cool water to flow through the condenser/absorber combination. This system is designed to bypass approximately one-half of the water from the cooling tower at inlet water temperatures below 75° F.

J. Data Acquisition

The data acquisition system may be divided into two main categories, real time and delayed data acquisition. The first category may be further subdivided into the real time Astrodata system, the Olivetti real time system, and strip charts. The Astrodata system outputs data via a central computer that has been processed to yield information in engineering units on two teletype printers, one of which is located in the solar house. Two digital displays are also available in the solar house which have access to the Astrodata system. Single and dual strip charts are available in the instrumentation room of the solar house.

Data flowing to the solar house teletype is accessed electronically by an Olivetti computer also located in the house. This computer is used to provide calculations on a near real time basis for six parameters:

1. Energy collected.

2. Energy consumed by the air conditioners.

3. Total incident energy.

4. Energy incident while collecting.

5. Energy available in the water tank.

6. Total collector efficiency.

Values for these parameters are automatically printed on a paper tape. The first three parameters are also automatically plotted by an extension of the Olivetti.

Since the Astrodata system cannot print out as much data as it has the capability of recording on magnetic tape, some data are processed only by the MSFC Data Systems Laboratory directly from the tape. In addition to these raw data reductions, the real time information printed on the teletype is also reduced by the Data Systems Laboratory. Thirty other channels of information in the Astrodata system, which may be accessed in real time by a special operation, are also reduced by the Data Systems Laboratory. Reduction of these data consists of plotting and tabulating both raw data and calculated parameters versus time. Some 50 parameters used in system performance evaluation are calculated and plotted using the raw data. Data output from the Data Systems Laboratory is normally received within 48 hours.

Because of Astrodata system limitations and Olivetti data input requirements, only 63 measurements of the total of 110 are automatically printed in real time. Thirty-three of the more important measurements are printed every 250 sec with the remaining 30 printed every 500 sec. The Astrodata system has a nonautomated printout capability for an additional 30 measurements at 50-sec increments. However, these measurements are printed only once per day to avoid excessive interference with the Olivetti. Initially, this left only 17 measurements to be reduced solely by the Data Systems Laboratory. However, 11 measurements giving air conditioner fluid loop temperatures have been added since testing began, increasing the number of measurements that are not accessible in real time to 27 and raising the total number of measurements to 120. Plotted values of all these parameters are combined to form daily solar house data books. Figure II-17 depicts the data flow scheme, and Table II-1 gives a summary of data acquisition.

K. Instrumentation

Table II-2 lists the 120 measurements on the solar house with a brief description of each. All temperature measurements are made using copper-constantan thermocouples. All thermocouples were compared to a 150°F constant temperature reference junction. The 45 deg mounted pyranometer and

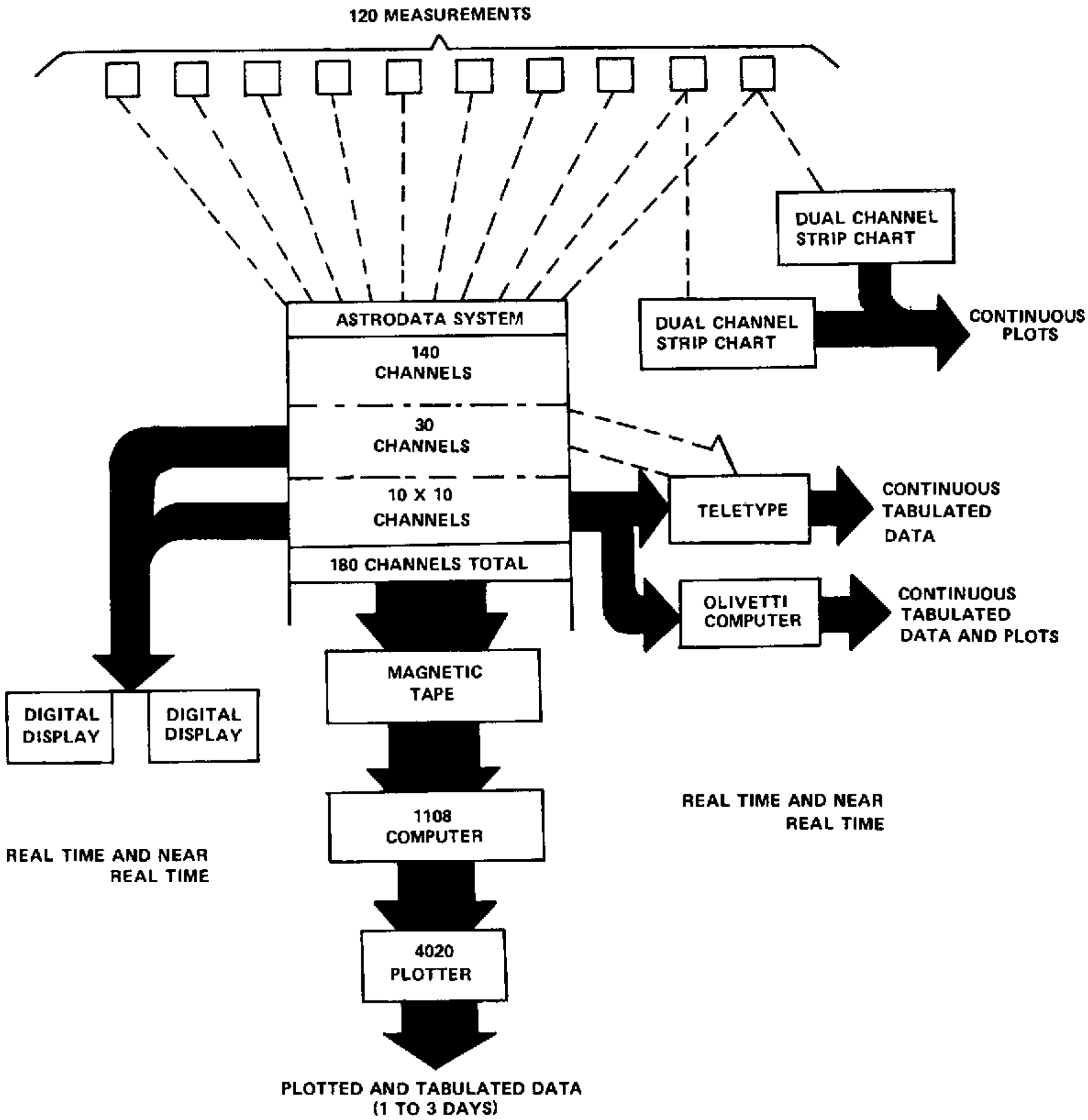


Figure II-17. Data flow chart.

a similar sensor mounted beneath the Tedlar were manufactured by the Weather Measure Corp. Although a horizontal mounted sensor and a diffuse sensor are shown in the measurement list, the sensors did not arrive in time for this series of tests. All liquid flowmeters are Potter Co. turbine type flowmeters. The

TABLE II-1. DATA ACQUISITION SYSTEM SUMMARY

A. Real Time and Near Real Time Data

   1. Astro-Computerized Data

      a. 63 Measurements Automatically Printing Continuously

      b. 30 Measurements Printed as Required (Normally Once per Day)

      c. Two Digital Displays

   2. Olivetti Computer Calculations

      a. Three Plotted Parameters (During Collection Periods)

      b. Six Tabulated Values (At All Times of the Day when the Astrodata System is Functioning)

   3. Strip Charts

      a. Four Measurements on Two Separate Strip Charts

B. Delayed Data (1 to 3 Days after Acquisition)

   1. Raw Data from Astrodata System

      a. Plotted

      b. Tabulated

   2. Calculated Data

      a. Plotted

      b. Tabulated

relative humidity sensors were manufactured by the Phys-Chemical Corp. Pressure transducers, differential and absolute, are MD Electronic Co. strain gage type sensors. All power measurements were made with General Electric watt-meters.

A check was run during these tests to assess the accuracy of thermocouples. From this check, it appears that most temperatures are accurate to ±0.5°F.

TABLE II-2. INSTRUMENTATION LIST

Item	Measurement No.	Description
1	T-001	Storage tank inside stand pipe near air conditioner inlet
2	T-002	Outlet collector water from storage tank to collector
3	T-003	Inlet collector manifold water
4	T-004	Outlet manifold water
5	T-005	Air conditioner tank bypass return water
6	T-007	Winter heat exchanger inlet water
7	T-008	Air conditioner inlet water
8	T-009	Air conditioner outlet water
9	T-010	Air conditioner cooling tower water at air conditioner inlet from cooling tower
10	T-011	Air conditioner cooling tower water at air conditioner outlet from cooling tower
11	T-012	AC/Htr pump inlet water
12	T-013	AC/Htr pump outlet water
13	T-014	Outlet water from storage tank to air conditioner
14	T-015	Inlet water from collector to storage tank
15	T-016	Air conditioner return air

TABLE II-2. (Continued)

Item	Measurement No.	Description
16	T-017	Air conditioner discharge air
17	T-019	Dwelling dry bulb air
18	T-020	Cooling tower water supply to air conditioner
19	T-021	Outdoors dry bulb air
20	T-031	Pumping control bypass line
21	T-032	Weak solution line from absorber
22	T-033	Heat exchanger to generator line
23	T-034	Percolator tube
24	T-035	Separator to heat exchanger line
25	T-036	Strong solution line to absorber
26	T-037	Condensate spill line
27	T-038	Air conditioner evaporator
28	T-039	Condensor housing
29	T-040	Cooling H_2O line to condensor
30	T-101	Storage tank collector water at inlet diffuser
31	T-105	Storage tank water inside stand pipe at bottom
32	T-108	Storage tank water at south end center
33	T-109	Storage tank water at south end bottom

TABLE II-2. (Continued)

Item	Measurement No.	Description
34	T-112	Storage tank water at north end center
35	T-113	Storage tank water at north end bottom
36	T-114	Storage tank surface at south end center
37	T-125	Storage tank east surface, 2 ft south of center
38	T-140	Storage tank support leg, southeast at bottom
39	T-201	Collector segment No. 1 at inlet
40	T-202	Collector segment No. 1 at outlet
41	T-205	Collector Segment No. 3 at inlet
42	T-206	Collector segment No. 3 at outlet
43	T-209	Collector segment No. 5 at inlet
44	T-210	Collector segment No. 5 at outlet
45	T-213	Collector segment No. 7 at inlet
46	T-214	Collector segment No. 7 at outlet
47	T-217	Collector segment No. 9 at inlet
48	T-218	Collector segment No. 9 at outlet
49	T-221	Collector segment No. 11 at inlet
50	T-222	Collector segment No. 11 at outlet
51	T-225	Collector segment No. 13 at inlet

TABLE II-2. (Continued)

Item	Measurement No.	Description
52	T-226	Collector segment No. 13 at outlet
53	T-229	Collector segment No. 15 at inlet
54	T-230	Collector segment No. 15 at outlet
55	T-233	Collector segment No. 17 at inlet
56	T-234	Collector segment No. 17 at outlet
57	T-237	Collector segment No. 19 at inlet
58	T-238	Collector segment No. 19 at outlet
59	T-241	Collector segment No. 21 at inlet
60	T-242	Collector segment No. 21 at outlet
61	T-245	Collector segment No. 23 at inlet
62	T-246	Collector segment No. 23 at outlet
63	T-249	Collector segment No. 25 at inlet
64	T-250	Collector segment No. 25 at outlet
65	T-253	Collector segment No. 27 at inlet
66	T-254	Collector segment No. 27 at outlet
67	T-257	Collector segment No. 29 at inlet
68	T-258	Collector segment No. 29 at outlet
69	T-261	Collector segment No. 31 at inlet
70	T-262	Collector segment No. 31 at outlet

TABLE II-2. (Continued)

Item	Measurement No.	Description
71	T-263	Collector heat exchanger 27-203 outlet
72	T-265	Collector heat exchanger 27-200 outlet
73	T-267	Collector heat exchanger 27-195 outlet
74	T-269	Collector heat exchanger 27-196 west
75	T-270	Collector heat exchanger 27-196 east
76	T-271	Collector heat exchanger 27-196 outlet
77	T-273	Collector heat exchanger 27-207 outlet
78	T-275	Collector heat exchanger 27-205 outlet
79	T-277	Collector segment No. 27 Tedlar wire
80	T-278	Collector segment No. 27 heat exchanger Tedlar cover air gap
81	T-279	Collector segment No. 27 back surface of tray
82	T-280	Collector heat exchanger 28-215 outlet
83	T-282	Collector heat exchanger 28-191 outlet
84	T-284	Collector heat exchanger 28-201 outlet
85	T-286	Collector heat exchanger 28-192 west
86	T-287	Collector heat exchanger 28-192 east
87	T-288	Collector heat exchanger 28-192 outlet
88	T-290	Collector heat exchanger 28-202 outlet

TABLE II-2. (Continued)

Item	Measurement No.	Description
89	T-292	Collector heat exchanger 28-193 outlet
90	T-294	Collector segment No. 28 Tedlar wire
91	T-295	Collector segment No. 28 heat exchanger Tedlar cover air gap
92	T-296	Collector segment No. 28 back surface of tray
		Pressure
93	P-001	Storage tank ullage pressure
94	P-002	Collector pump outlet pressure
95	P-003	Air conditioner heat exchanger pump outlet pressure
96	P-004	Air conditioner heat exchanger outlet water pressure
		Pressure Differential
97	DP-001	Collector pump pressure rise
98	DP-002	Air conditioner heat exchanger pump pressure rise
99	DP-003	Collector pressure drop
		Flow Rate
100	F-001	Collector pump flow
101	F-002	Air conditioner hot water flow

TABLE II-2. (Continued)

Item	Measurement No.	Description
102	F-003	Air conditioner heat exchanger loop flow
103	F-004	Cooling tower water flow
104	F-005	Dehumidifier air flow
		Heat Flux
105	Q-001	Solar flux, horizontal
106	Q-002	Solar flux, diffuse components only
107	Q-003	Solar flux, 45 deg south
108	Q-004	Solar flux, 45 deg south inside collector segment No. 31
		Humidity
109	RH-001	Relative humidity in collector segment No. 31
110	RH-002	Relative humidity of outdoor air
111	RH-003	Relative humidity of dwelling area air
112	RH-004	Relative humidity of air conditioner discharge air
		Power
113	W-001	Collector pump power
114	W-002	Air conditioner heat exchanger loop pump power

TABLE II-2. (Concluded)

Item	Measurement No.	Description
115	W-003	Dehumidifier power
116	W-004	Air conditioner and cooling tower power
117	W-005	Auxiliary heater power
118	W-006	Duct heater power
	Wind	
119	WD	Wind direction
120	WV	Wind velocity

Figure II-18 shows a schematic of instrumentation in the fluid and gas loops with power measurements also shown. Figure II-19 depicts instrumentation on the collector. Figure II-20 represents location of online sensors added to the air conditioner on June 24, 1974, and Figure II-21 shows instrumentation within the energy storage tank.

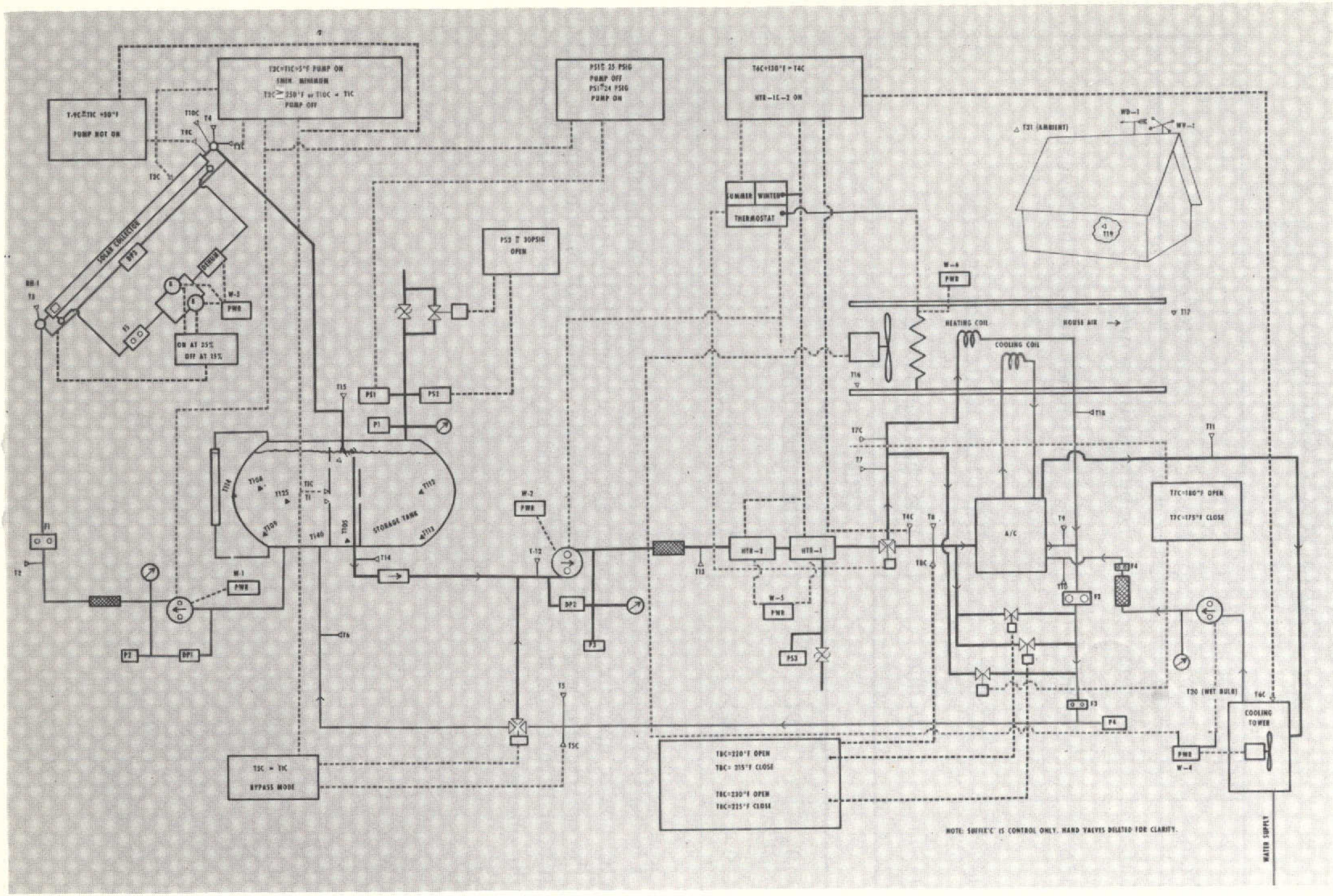


Figure II-18. Solar house fluid loop instrumentation and control schematic.

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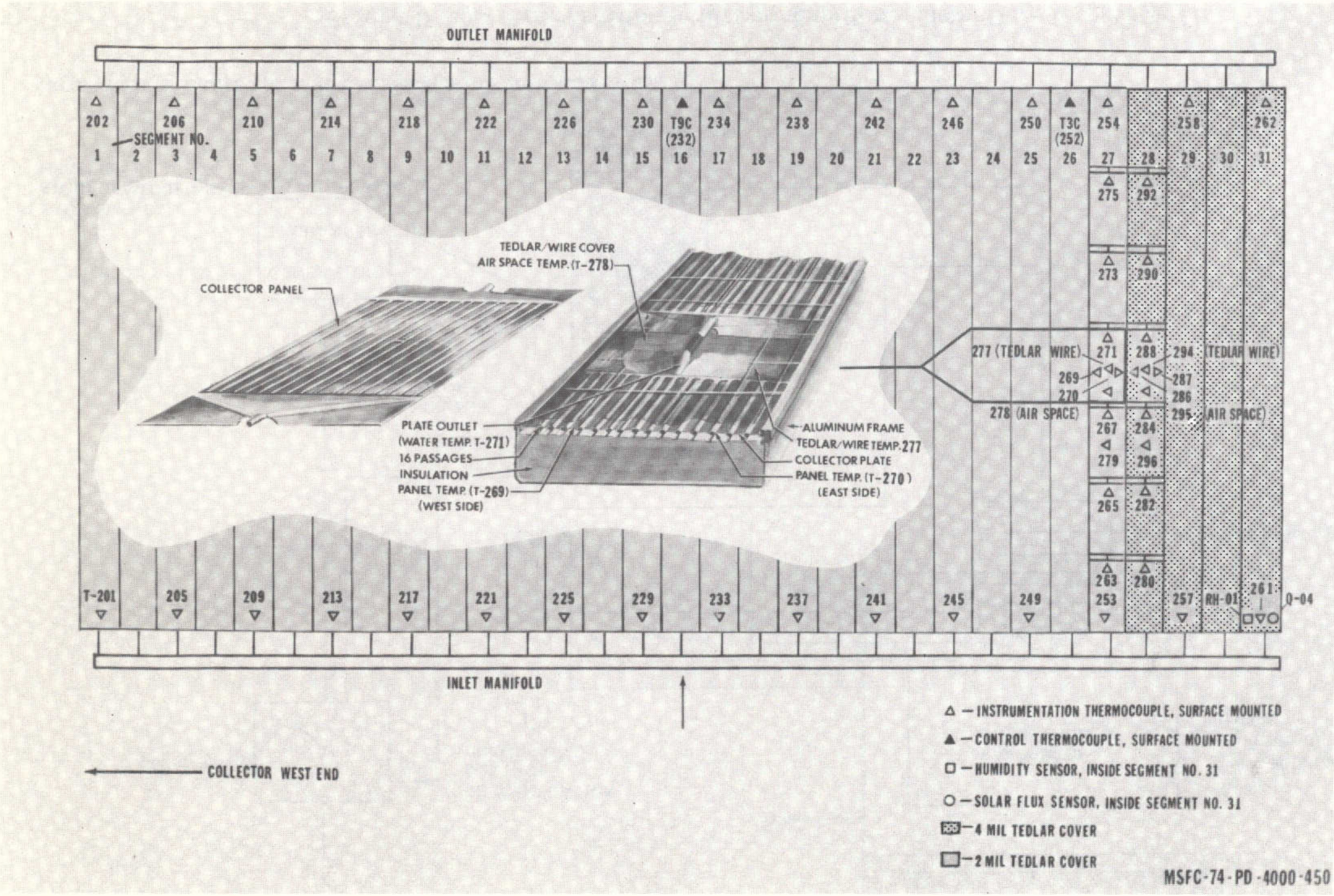
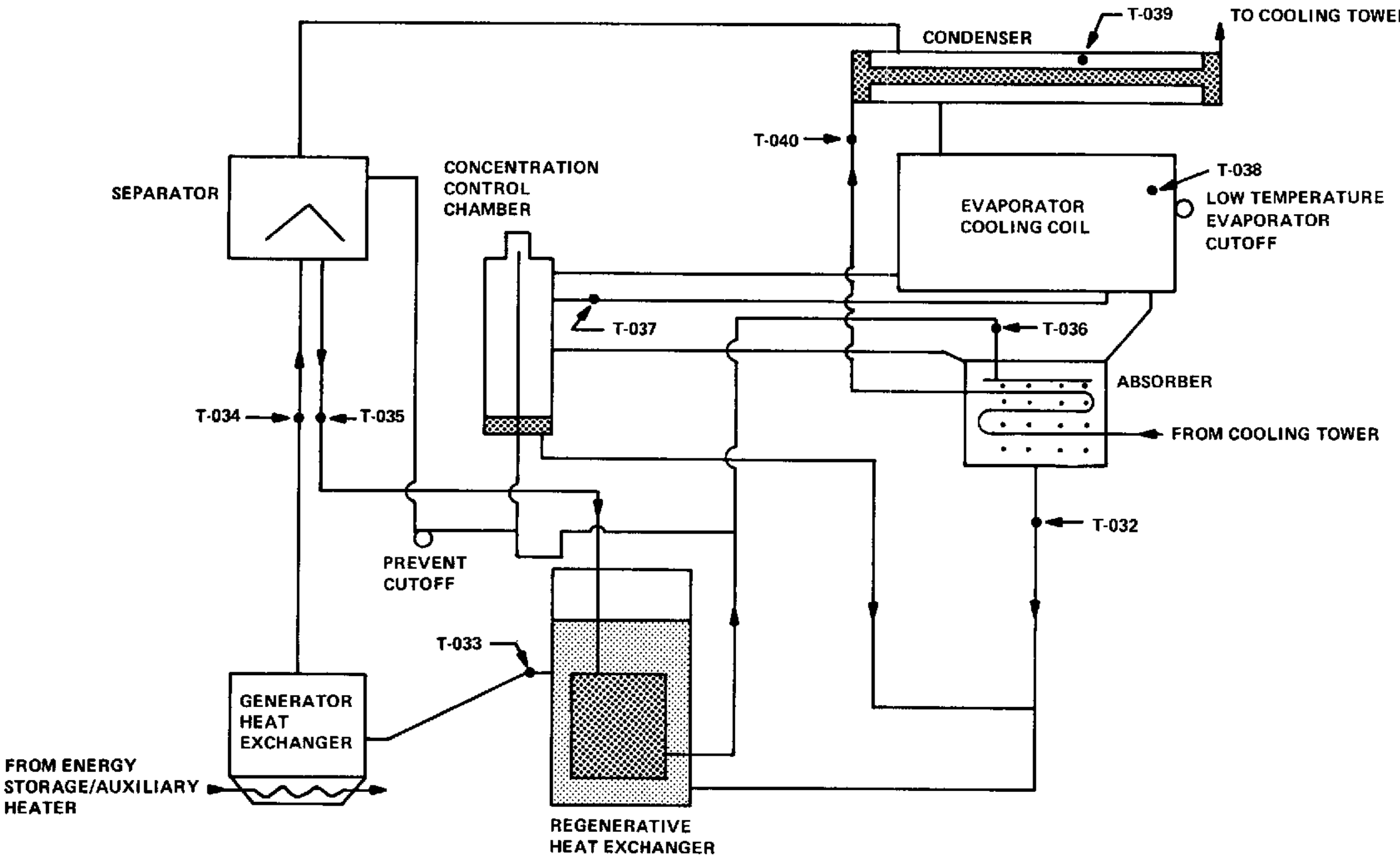


Figure II-19. Identification and location of instrumentation on solar collector.



NOTE: T-031 IS LOCATED ON NONCONDENSABLE GAS PUMPING SYSTEM.

Figure II-20. Air conditioner instrumentation schematic.

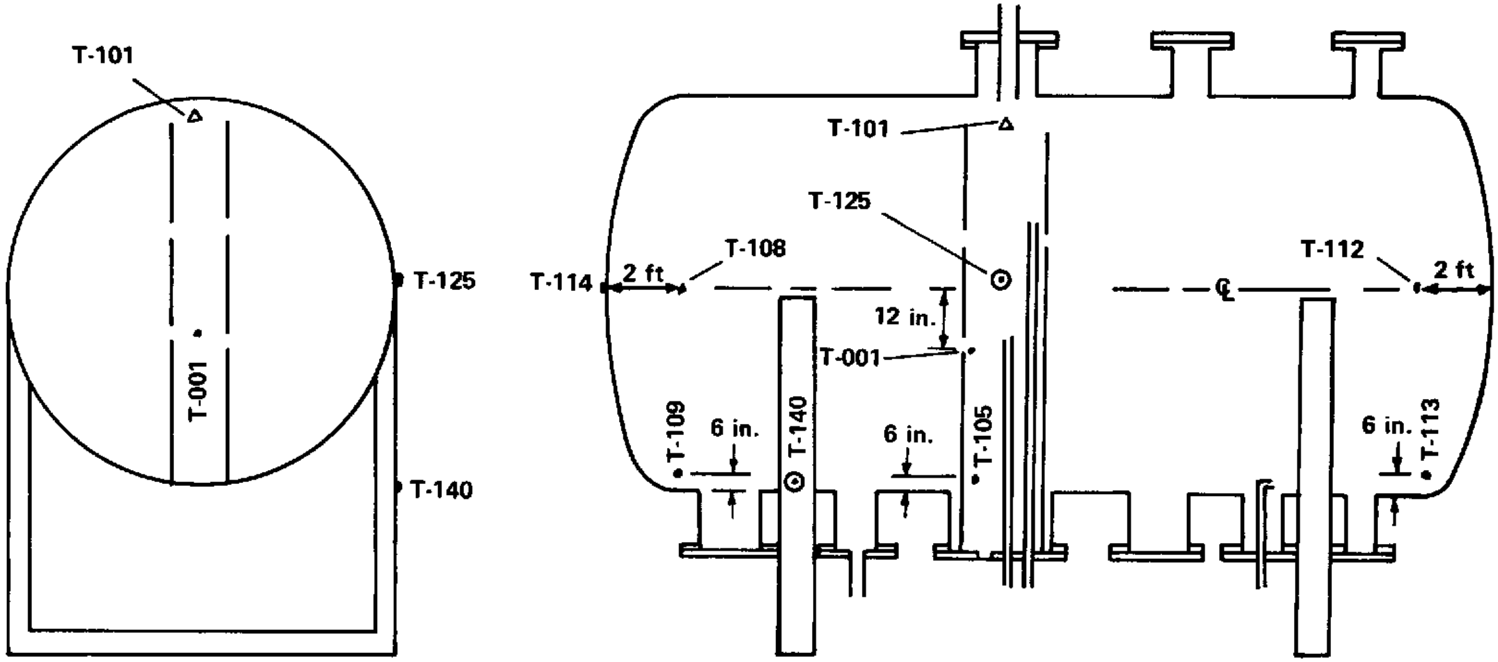


Figure II-21. Energy storage tank instrumentation.

III. TEST DATA AND EVALUATION

A. General

The philosophy of this test program was to construct a solar heated and cooled facility using the best designs that could be accomplished with readily available, current state-of-the-art equipment. This goal was accomplished within a tight schedule. As a result, it was impossible to realize a high degree of system optimization or to achieve complete simulation of typical home application hardware.

The object of the program was to gain practical insight into continuous operation that would be applicable to commercial and residential facilities of this type. The facility was operated over a broad spectrum of weather conditions and random seasonal variations imposed by the north Alabama climatic zone.

This report covers only the 3 months from initial test activation of the facility, in late May, to the first of September. During this period of testing the system configuration was changed several times to exploit information obtained from the tests and to correct system problems. In general, an attempt was made to make no more than one major change at a time, so that the effect of each change could be properly isolated and assessed. Most changes occurring in this phase of testing affected the collector loop and air conditioner/heater loop logic systems. Minor collector modifications and repairs were also performed during this period. The collector flow rates were also changed a number of times.

Numerous electrical and mechanical difficulties were experienced during these tests. However, since some of these problems were traceable to the age and qualification of the hardware used, a discussion of all these failures is not included. Because of the significant variation in day-to-day results, most performance assessments are made based on average values. In the interest of brevity, all parameter variations, occurrences, and test data are not presented. Table III-1 gives a cursory summary of major test changes and occurrences along with weather conditions for the report period. The bulk of the data contained in this section addresses problems that apply in general to any system of this type, with emphasis placed on assessing subsystem and overall system performance.

TABLE III-1. TEST SUMMARY

Test No. (Date)	Description	Weather
5-27-74	Changed heater logic system from 130° F bias to 125° F bias	Clear with broken clouds
5-28-74	Initial test data. Using auxiliary heaters and solar collector heated up H_2O tank to 227° F night before	Clear with broken clouds
5-29-74	Normal operation	Sunny with haze
5-30-74	Changed collector flow rate from nominal of 31 gpm to nominal of 16 gpm	Partly cloudy with sun
5-31-74	Added insulation to all collector inlet segments	Overcast
6-1-74	Heated water using auxiliary heaters and collector	Overcast with rain
6-2-74	Modified heater logic system so that < 195° F both heaters on, < 200° F 13.5 kW heater on and > 205° F both heaters off	Partly cloudy
6-3-74	Found that sealant around collectors is allowing water to leak into collector	Partly cloudy
6-4-74	Normal operation	Partly cloudy with haze
6-5-74	Found all temperatures on thermocouple reference junction No. 7 were low by 1.5° F (i.e., T-005, T-007, T-008, T-009, T-010, T-011, T-012, T-013, T-016, T-017, T-018, and T-020)	Overcast with rain

TABLE III-1. (Continued)

Test No. (Date)	Description	Weather
6-6-74	Added 60-sec time delay to auxiliary heater to allow temperatures to stabilize before logic checks	Overcast with rain
6-7-74	Changed collector pump activation — deactivate thermocouple reference from T-001 to T-105; changed bypass logic from AC/Htr loop to reference T-108 in tank rather than T-001	Overcast with rain
6-8-74	Normal operation	Partly cloudy
6-9-74	Normal operation	Partly cloudy
6-10-74	Changed logic so that both air conditioner bypass valves (MV-4 and MV-5) open if refrigerant spillover line temperature is below 50° F	Scattered clouds
6-11-74	Changed refrigerant spillover line trip temperature to 55° F; recaulked roof	Scattered clouds
6-12-74	Found that air conditioner inlet relative humidity gage (H-003) is reading 20 percent too low behind instrumentation cabinet; will move to air conditioner inlet	Overcast
6-13-74	Normal operation	Partly cloudy
6-14-74	To increase dehumidification flow in collector, drilled out flow orifice to 0.5 in. diameter; made air conditioner duct velocity profile run today; insulated air conditioner on bottom panel	Clear

TABLE III-1. (Continued)

Test No. (Date)	Description	Weather
6-15-74	Normal operation	Overcast and partly cloudy
6-16-74	Normal operation	Clear
6-17-74	Air conditioner inlet relative humidity gage relocated to air conditioner inlet	Clear
6-18-74	Normal operation	Overcast
6-19-74	Normal operation	Partly cloudy
6-20-74	Normal operation	Clear
6-21-74	Normal operation	Overcast
6-22-74	Logic amplifier failed inhibiting collector operation	Partly cloudy
6-23-74	Normal operation	Overcast with rain
6-24-74	Defective collector amplifier replaced; collector flow rate raised to 44 gpm; F-02 failed today and was repaired	Overcast
6-25-74	First in series of special air conditioner tests, 10-gpm temperature decay; collection system inhibited	Partly cloudy with haze
6-26-74	Second air conditioner test varying flow rate from 10 to 2 gpm; collection system inhibited	Overcast and partly cloudy
6-27-74	Normal operation	Scattered clouds

TABLE III-1. (Continued)

Test No. (Date)	Description	Weather
6-28-74	Third air conditioner test, 7-gpm temperature decay; collection system inhibited	Partly cloudy
6-29-74	Normal operation	Partly cloudy
6-30-74	Collector pump failed to shut off automatically; manual shutoff at 12 p.m.	Partly cloudy
7-1-74	Fourth air conditioner test, 3-gpm temperature decay; collector system inhibited	Partly cloudy and hazy
7-2-74	Fifth air conditioner test, 7-gpm temperature decay; collector system inhibited	Clear
7-3-74	Sixth air conditioner test, 7- and 10-gpm temperature decay; collector system inhibited	Clear
7-4-74	Seventh air conditioner test, 5-gpm temperature decay; collector system inhibited	Partly cloudy
7-5-74	Eight air conditioner test, 3-gpm temperature decay; collector system inhibited	Overcast with rain
7-6-74	Collector would not come on because of logic system set point change	Partly cloudy
7-7-74	Collector would not come on because of logic system set point change	Partly cloudy

TABLE III-1. (Continued)

Test No. (Date)	Description	Weather
7-8-74	Ninth air conditioner test, 3-gpm temperature decay; collector system inhibited	Overcast
7-9-74	Tenth air conditioner test, 1.7-gpm temperature decay; collector system inhibited	Partly cloudy
7-10-74	Eleventh and final air conditioner test, 1.7-, 5- and 10-gpm temperature decay; collector system inhibited	Clear
7-11-74	Thermostat set at 74° F; changed MV-4 logic to following: T-008 ≥ 192° F — open, 5 gpm through air conditioner T-008 < 192° F — closed, 10 gpm through air conditioner	Partly cloudy
7-12-74	Flowmeter F-002 failed; system powered down at 12 a.m. for logic system modification	Partly cloudy
7-13-74	Installed storm window covers for jalousie windows; added flashing over inlet and outlet collector manifolds	
7-24-74	System back in operation with new logic	Overcast
7-25-74	Collector flow rate changed to 31 gpm	Overcast, rain, and partly cloudy

TABLE III-1. (Continued)

Test No. (Date)	Description	Weather
7-26-74	Lightning damaged data acquisition system and control system	Overcast with rain
7-27-74	Data system down for repair	Overcast with rain
7-28-74	Data system down for repair	Partly cloudy with haze
7-29-74	T-5 in error	Hazy and clear
7-30-74	Replaced T-001 with T-105 for collector off logic	Clear
7-31-74	Normal operation	Partly cloudy and overcast
8-1-74	Collector pump failed to shut off; winter/summer valve (MV-3) found to have been leaking for 2 to 3 days	Partly cloudy
8-2-74	Repaired faulty three-way valve	Overcast
8-3-74	MV-3 reinstalled incorrectly causing low air conditioner flow rate and intermittent percolation loss in air conditioner	Overcast
8-4-74	Air conditioner operation erratic	Overcast and partly cloudy
8-5-74	Air conditioner thermostat set back to 72°F	Clear
8-6-74	Found MV-1 stuck in tank bypass mode for 2 days	Partly cloudy

TABLE III-1. (Continued)

Test No. (Date)	Description	Weather
8-7-74	Replaced MV-1	Overcast with rain
8-8-74	Added logic timer	Overcast
8-9-74	Water boiled out of lithium bromide because of electrical system mal-function	Overcast and partly cloudy
8-10-74	Normal operation	Partly cloudy
8-11-74	Normal operation	Partly cloudy
8-12-74	Data system down	Partly cloudy
8-13-74	MV-5 valve fuse burned out and valve stuck open; MV-1 failed in bypass mode again	Partly cloudy
8-14-74	Shut down to replace MV-5; found MV-4 stuck in open position; found extreme rust in MV-5	Overcast
8-15-74	MV-5 replaced and filter removed and cleaned	Clear
8-16-74	Normal operation	Overcast
8-17-74	Normal operation	Partly cloudy
8-18-74	Normal operation	Partly cloudy
8-19-74	Normal operation	Clear
8-20-74	Normal operation	Clear to partly cloudy

TABLE III-1. (Concluded)

Test No. (Date)	Description	Weather
8-21-74	Found that 13.5-kW heater coming on too high at T-008 value of 193° F	Clear to partly cloudy
8-22-74	Manually placed MV-1 in bypass mode to raise energy tank temperature	Clear and hazy
8-23-74	Normal operation	Clear
8-24-74	Normal operation	Clear
8-25-74	Manually deactivated collector pump because T-3 > T-4 at 4:40 p.m.	Clear and hazy
8-26-74	Normal operation	Overcast and partly cloudy
8-27-74	Normal operation	Partly cloudy
8-28-74	Normal operation	Partly cloudy
8-29-74	Lightning struck nearby and data system knocked out; lost instrumentation and control system	Overcast with rain
8-30-74	Air conditioner loop manually put in tank bypass mode since controller not functioning properly because of control system malfunction; lithium bromide had crystalized, restarted OK	Overcast with rain
8-31-74	Repaired lightning damage	Overcast with rain

B. Flow Balance of Collector

Prior to the start of testing, action was taken to assure an equal flow rate in the collector segments. This was necessary because of the longer flow paths to and from the single inlet/outlet manifold. Also this adjustment was necessary to overcome any flow resistance mismatch between segments because of tolerance buildup or manufacturing flaws. Individual hand valves were placed in the tubes leading to each segment to allow this adjustment.

A 1-in. line size, hand-operated iron gate valve was used for adjusting flow at the inlet of each collector segment. Prior to installation, each valve was modified by placing shims between the valve gate and the stem for the purpose of removing "looseness" in valve adjustment, so that the valve gate could be precisely positioned with the stem. Additionally, a bench test was performed on each valve to determine differential pressure across the valve versus flow rate through the valve for various valve stem settings.

After installation in the collector subsystem, each of the collector segment inlet valves was preset (based upon bench test data) to provide the differential pressure values shown in Table III-2 for a 31-gpm water flow rate into the collector inlet manifold. Following this, a flow balance test was run to verify equal flow distribution in the segments. The collector segment outlet temperatures were compared while flowing 31 gpm through the collector on a clear day and these compared within $\pm 2°$ F which was acceptable; therefore, no further adjustment to the preset valve positions was necessary.

C. Collection Subsystem

During initial testing water was noted to have leaked under the Tedlar cover. This rain water leaked into the collector pans by flowing through the Tedlar cover-to-frame caulking joint, permeating part of the insulation protection system. After the water was drained, the caulking was found to be a type that had been substituted during fabrication and would not bond to Tedlar. The joints were then recaulked with the proper caulking compound. After this repair was completed the collector subsystem was operated in the rain after which previously wet insulation was rechecked and found to be dry. No further difficulties with rain water leakage have been experienced.

On May 30 the collector flow rate was decreased from 31 gpm to 16 gpm in an attempt to overcome flow channeling in the energy storage tank (see subsection E). At the lower flow rate the rate of energy collection was found

TABLE III-2. SOLAR COLLECTOR VALVE DIFFERENTIAL PRESSURE

Collector Segment No.[a]	Inlet Valve Differential Pressure (psid)[b]
1 through 5	0
6 and 7	0.01
8	0.02
9	0.03
10	0.04
11	0.05
12	0.06
13	0.08
14	0.10
15	0.13
16	0.15
17	0.13
18	0.10
19	0.08
20	0.06
21	0.05
22	0.04
23	0.03
24	0.02
25 and 26	0.01
27 through 31	0

a. Numbered 1 through 31 beginning at west end of collector.

b. For a 31 gpm total water flow rate to inlet manifold.

to have decreased significantly. As a result the collector flow was increased back to the original 31 gpm flow rate on June 10. Later on June 27 the collector flow rate was boosted to 44 gpm (the peak flow rate attainable with the existing pump) in an effort to increase collection. However, test data indicated a slight decrease (Fig. III-1). A decrease in collection because of a

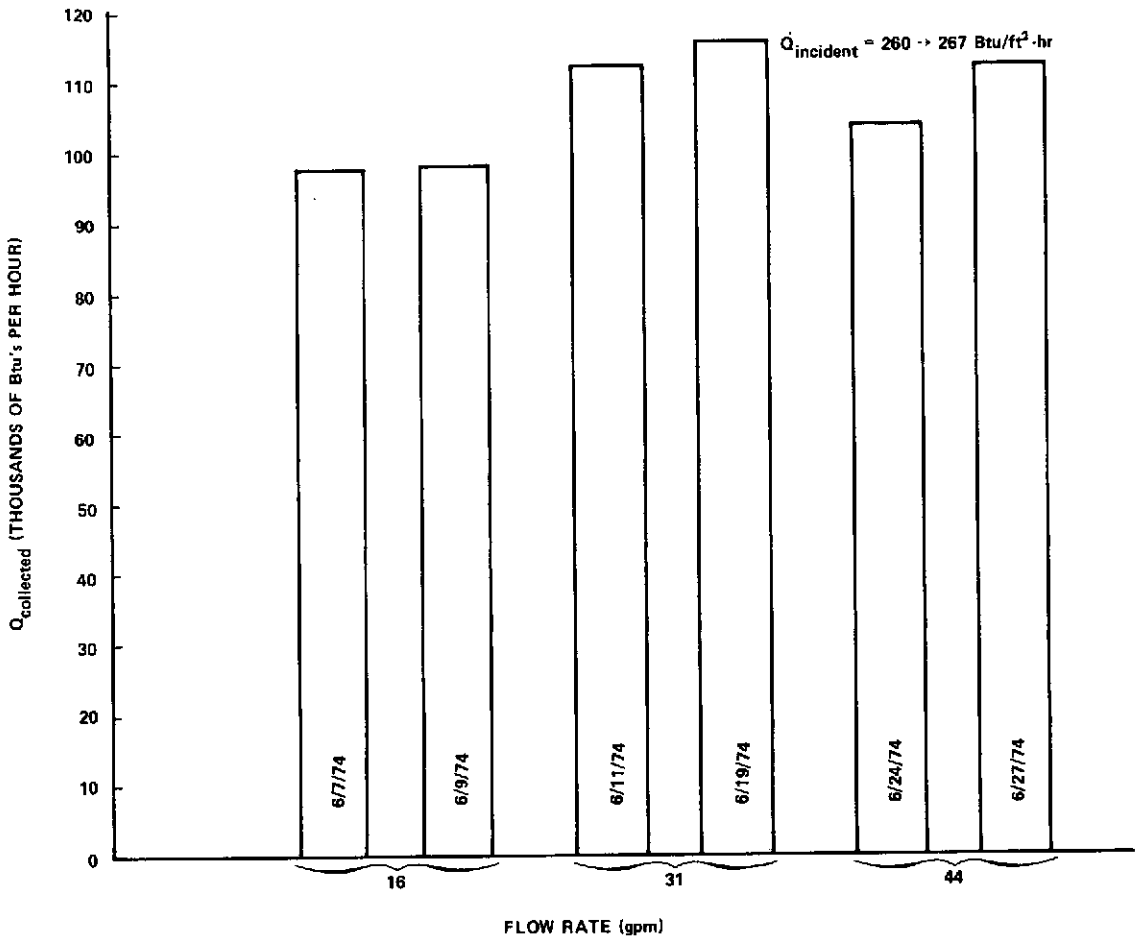


Figure III-1. Flow rate effect on energy collection.

flow rate increase is not justifiable; consequently, the slight performance loss is conjectured to be a result of other causes (i.e., variation in collector surface temperature, incident flux, ambient temperature, etc.). However, it was apparent that no appreciable performance enhancement resulted from the flow rate increase. Since the higher flow rate required more pump power, the flow rate was returned to 31 gpm on July 25.

The collector segments were mounted on the roof in March; they were later removed for 2 weeks for minor repairs and replaced. During this initial 8 to 12 week period after installation, noticeable discoloration or "clouding" of the Tedlar occurred. Figure III-2 shows a sample of Tedlar which had been exposed from December 1973 to June 1974 in a prototype test segment under similar test conditions. For comparison, a clear, unused sample is also shown on the right side of Figure III-2. From this photograph, the transmissivity degradation is obvious from the white appearance of the degraded sample. The degradation was determined by chemical analysis to have resulted from the depositing on the tedlar of condensable offgassing products from a binder used in the fiberglass insulation battens. Tests of two samples taken from the prototype segment indicated transmittance values of 82.6 percent and 73.4 percent. These values may be compared to the initial undegraded transmittance of 94 percent. Alcohol cleaning of the samples restored the original transmittance. In the future, this problem should be avoided by either using fiberglass qualified at higher temperatures than the common household material used or by driving off the offgassing products in an oven prior to use. Heating the fiberglass to 800°F for five minutes or to 500°F for 30 minutes will remove the undesirable products.

A close scrutiny of the cover reveals darkened areas around the wire mesh. These darker areas are the epoxy glue used to attach the mesh to the Tedlar. This glue, which is relatively transparent on initial application, darkens after sun exposure. This darkening has the unwanted effect of obscuring more collector surface than the simple projected area of the wire. Observations indicate that up to 15 percent of some segments are shaded by the wire/glue combination.

In an effort to determine the performance of the collector subsystem, the collector pump was twice run at night. During both these tests, the energy loss rate from the collector stabilized at 125 000 Btu/hour. Using this value, the loss flux rate was calculated to be 102.6 Btu/hour-ft^2, which is near the predicted value.

Collector performance is very difficult to accurately assess. This is primarily because it is a function of a number of variables which vary with time. Typically, for a given collector geometry, the system performance is a function of the average surface temperature, the incident flux, and the ambient conditions (temperature and wind velocity). Figure III-3 illustrates the typical effect of these parameters, excluding wind velocity. Variations in incident flux not only depend upon whether the weather is inclement or clear but also upon the degree of cloud cover. Figure III-4 shows typical examples of cloudy, partly cloudy, and clear days. As seen in this figure, the incident flux varies with time of day.

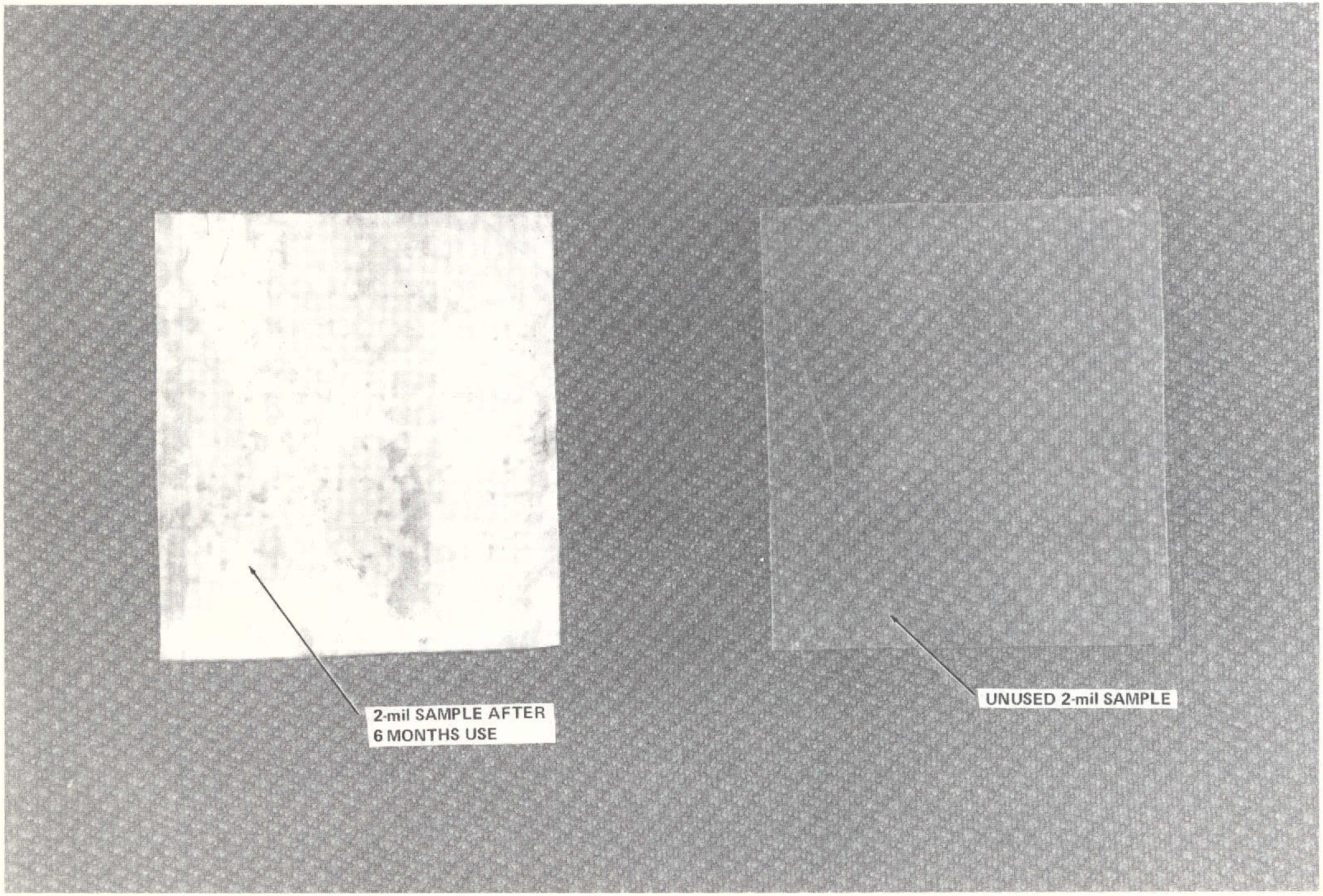


Figure III-2. Contamination of Tedlar resulting from outgassing.

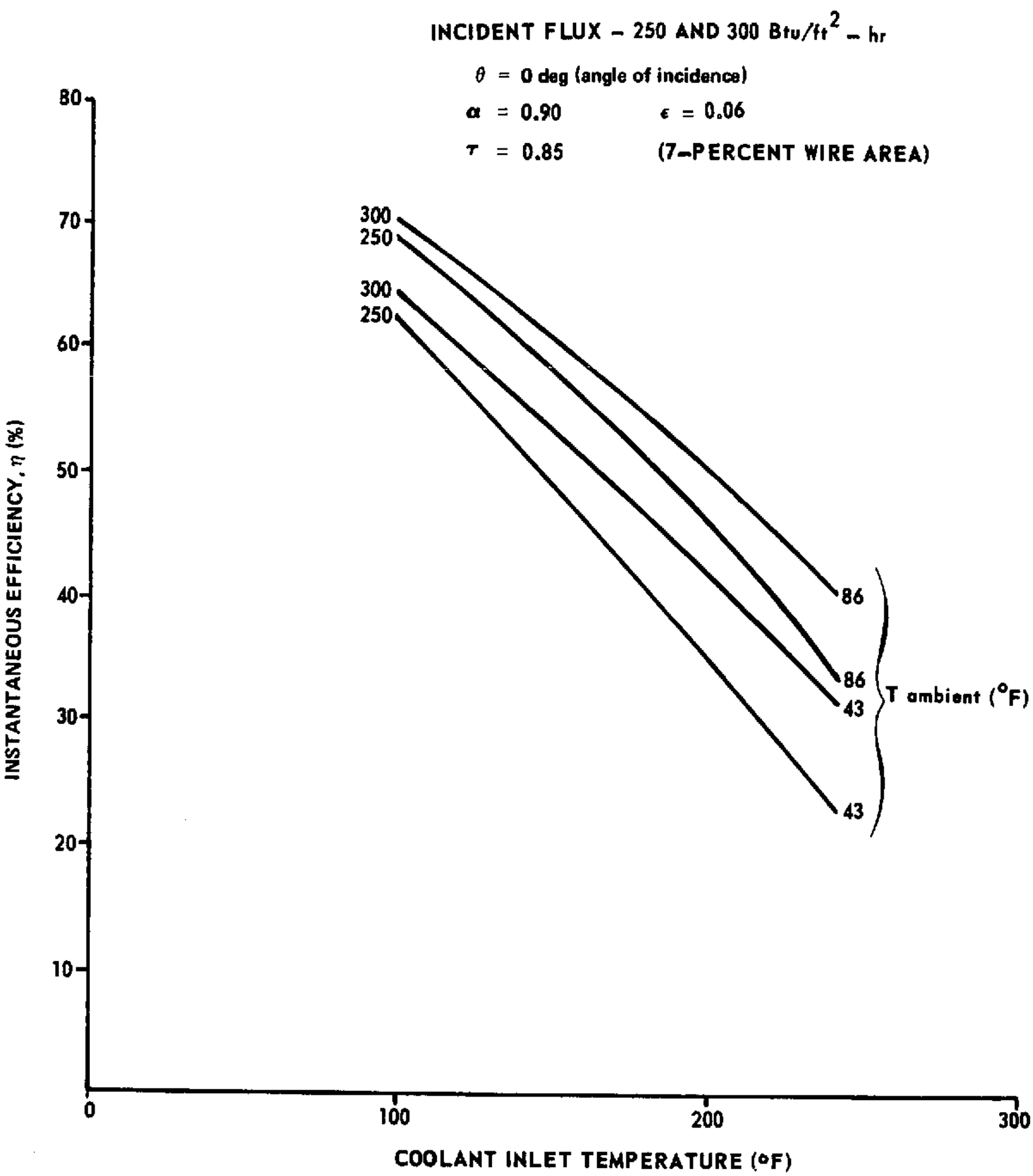


Figure III-3. Instantaneous collector efficiency.

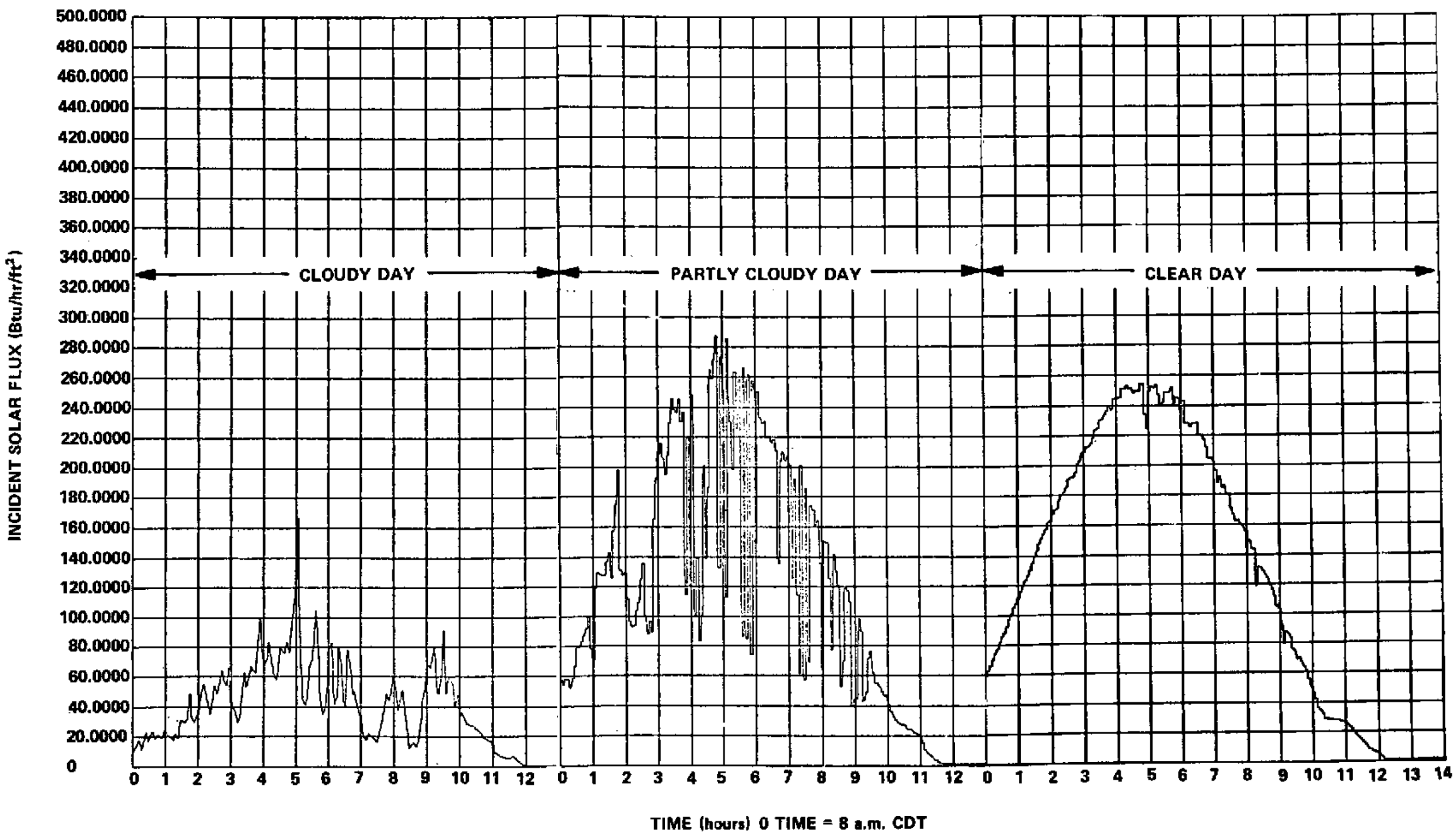


Figure III-4. Incident solar flux for various weather conditions versus time.

Defining the collector efficiency, η , based on instantaneous rates by

$$\eta = \frac{\dot{Q}_{collected}}{\dot{Q}_{incident}} ,$$

it is obvious that this instantaneous value varies also with time of day (Fig. III-5). A more meaningful efficiency from a system performance viewpoint is the integrated efficiency over a day defined by

$$\eta_T = \frac{\int_{day} \dot{Q}_{collected}\, dt}{\int_{day} \dot{Q}_{\substack{incident \\ while \\ collecting}}\, dt} .$$

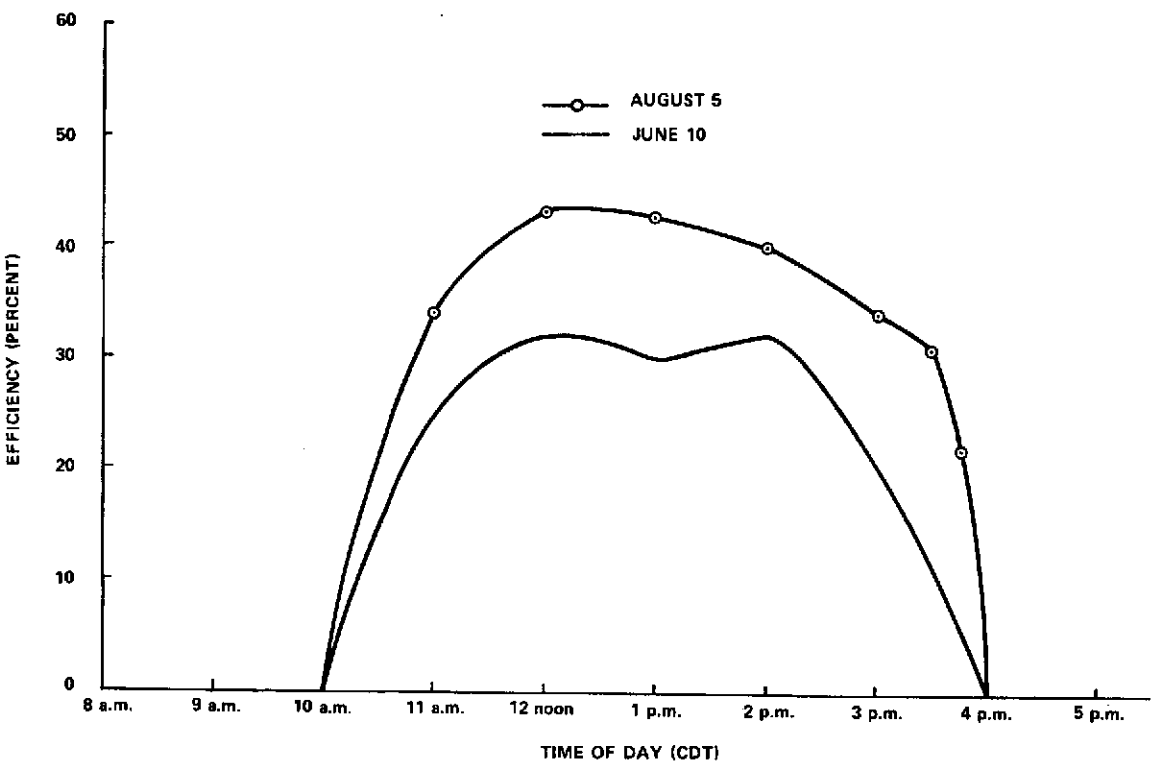


Figure III-5. Collector efficiency comparison.

Values of η for peak fluxes (i.e., taken around solar noon, 1 p.m. CDT) for selected days along with parameters affecting η are given in Figure III-6. Figure III-7 tabulates similar daily integrated efficiencies, η_T, with average daily values of significant parameters. Large excursions in these parameters are a result of the transient nature of the incident energy. Rapid changes in the incident flux produce quick changes in surface and fluid temperatures which result in cyclic operation of the collection system. Long off-times allow the fluid in the supply lines to cool significantly causing bursts of hot and cold fluid through the collector. These bursts cause anomalous collection rate values to be indicated. These false indications are due to the cold fluid which is trapped in the supply lines being pushed into the collector prior to the warm fluid from the storage tank. As the cold fluid passes the collector inlet thermocouple, a rapid drop in temperature occurs while the outlet thermocouples continue to be heated by the incident flux. Therefore, if an intermittent data sample is taken as this occurs, an abnormally high collection will be indicated. Conversely, as the warm fluid from the hot storage tank enters the collector, there is a period when a wave of hot fluid is on the collector inlet while cooler line fluid is still passing the outlet causing low or negative collection rates to be indicated.

An examination of Figures III-6 and III-7 indicates that a marked increase in collector efficiency occurred after the logic system modification. The only readily available explanation for this is the decrease in solar incident angle that occurred from early June to early August of approximately 6 deg (Fig. III-8). However, an assessment of the expected enhancement resulting therefrom does not fully justify the increase in energy collected. This is substantiated when comparing two typical clear days for June and August. Pertinent values for these two days are given in Table III-3. From these data, it can be seen that a 15-percent increase in incident energy (apparently a result of the more favorable solar incidence angle and the clearer day) is attended by a 54-percent increase in collected energy from June to August. The average surface temperature is only slightly lower for the June day. Differences in other parameters which affect collection efficiency do not justify the large increase in collection. However, should the surface coating properties or Tedlar transmissivity values be functions of incident angle, this phenomenon might be explained, especially if the collector surface has specular qualities at the lower sun incidence angles that could result in lower apparent absorptivity with incident angle or if the Tedlar degradation causes the cover to be more opaque to inclined rays. This phenomenon is currently under study.

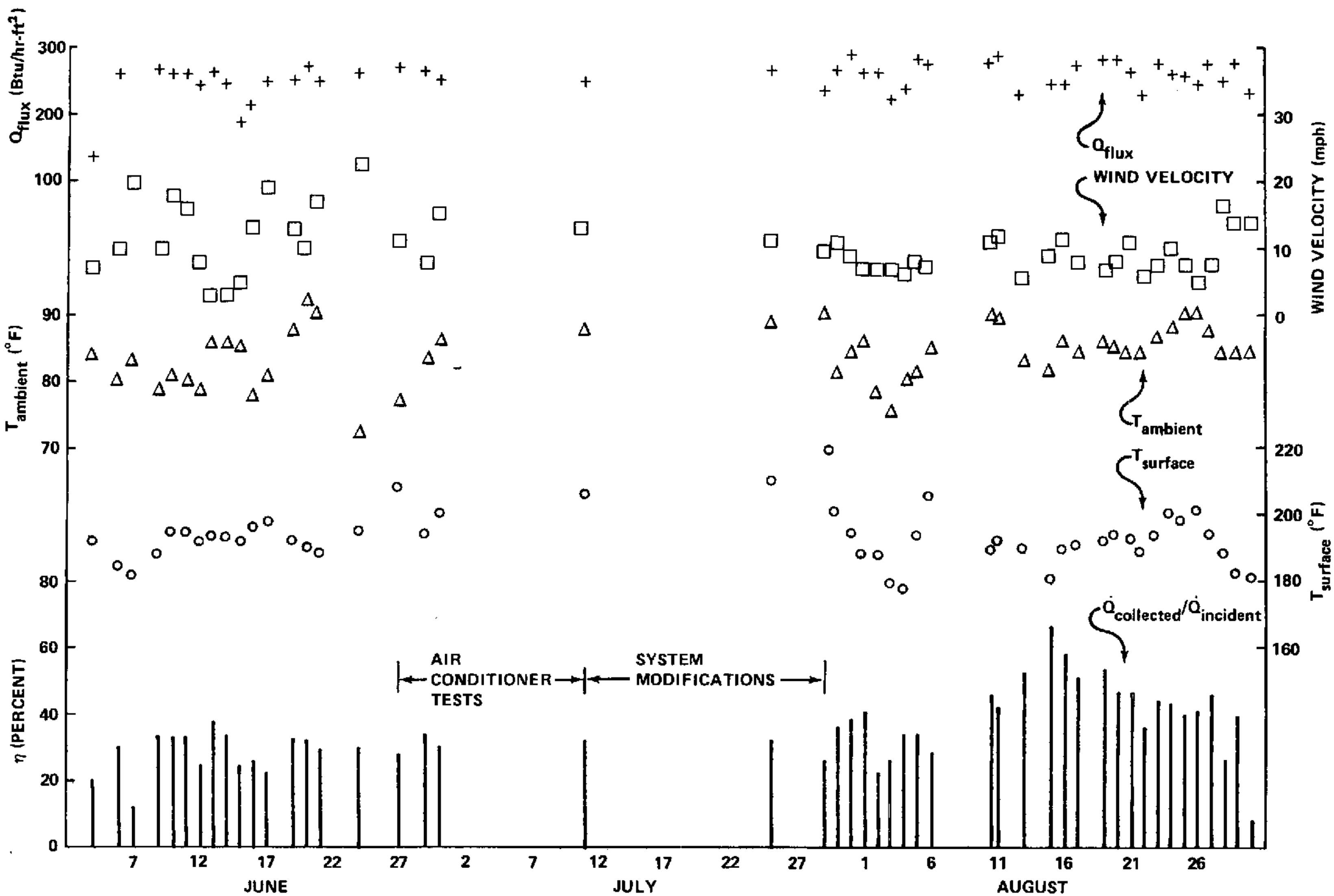


Figure III-6. Integrated daily collector efficiency.

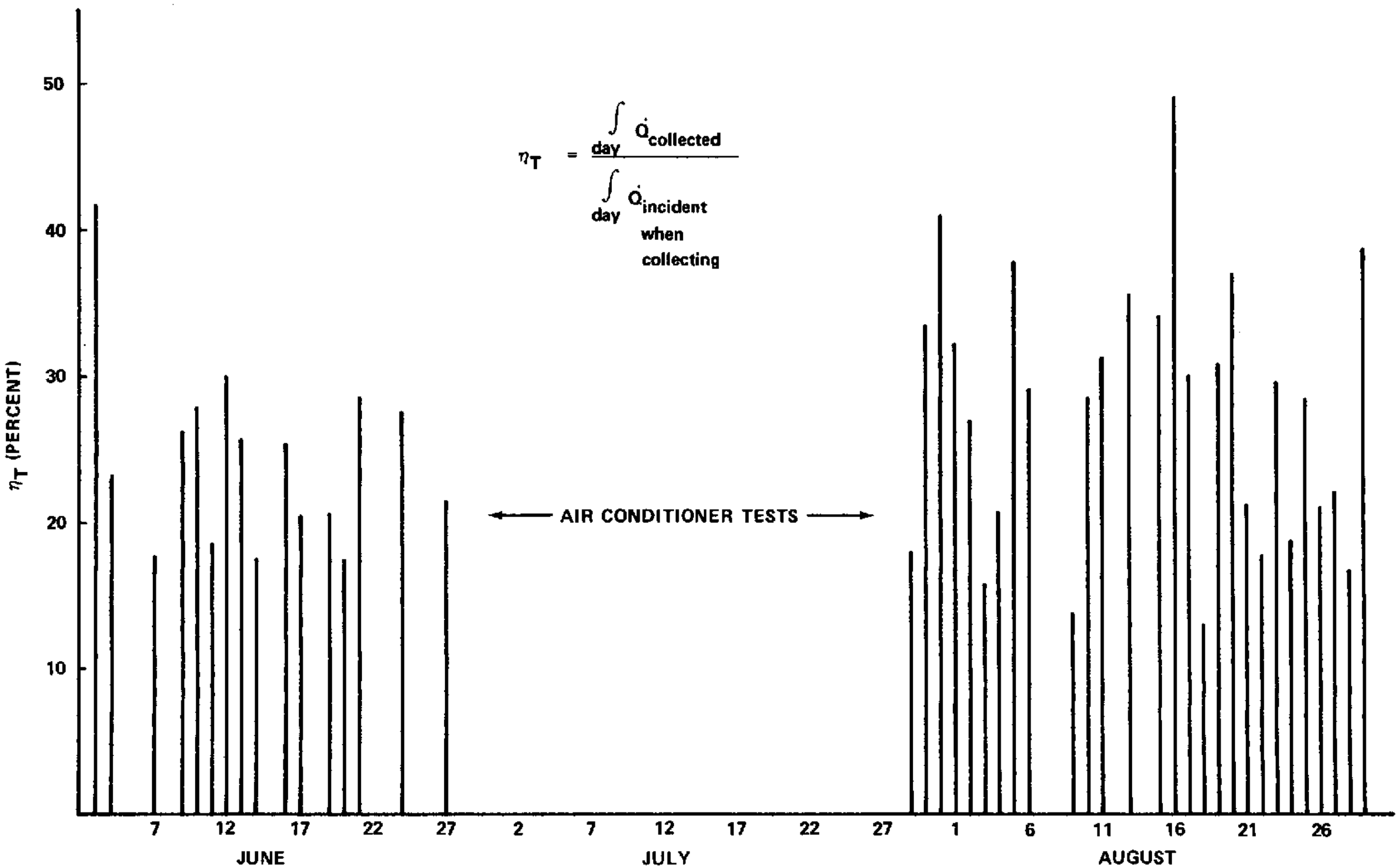


Figure III-7. Peak collector efficiency.

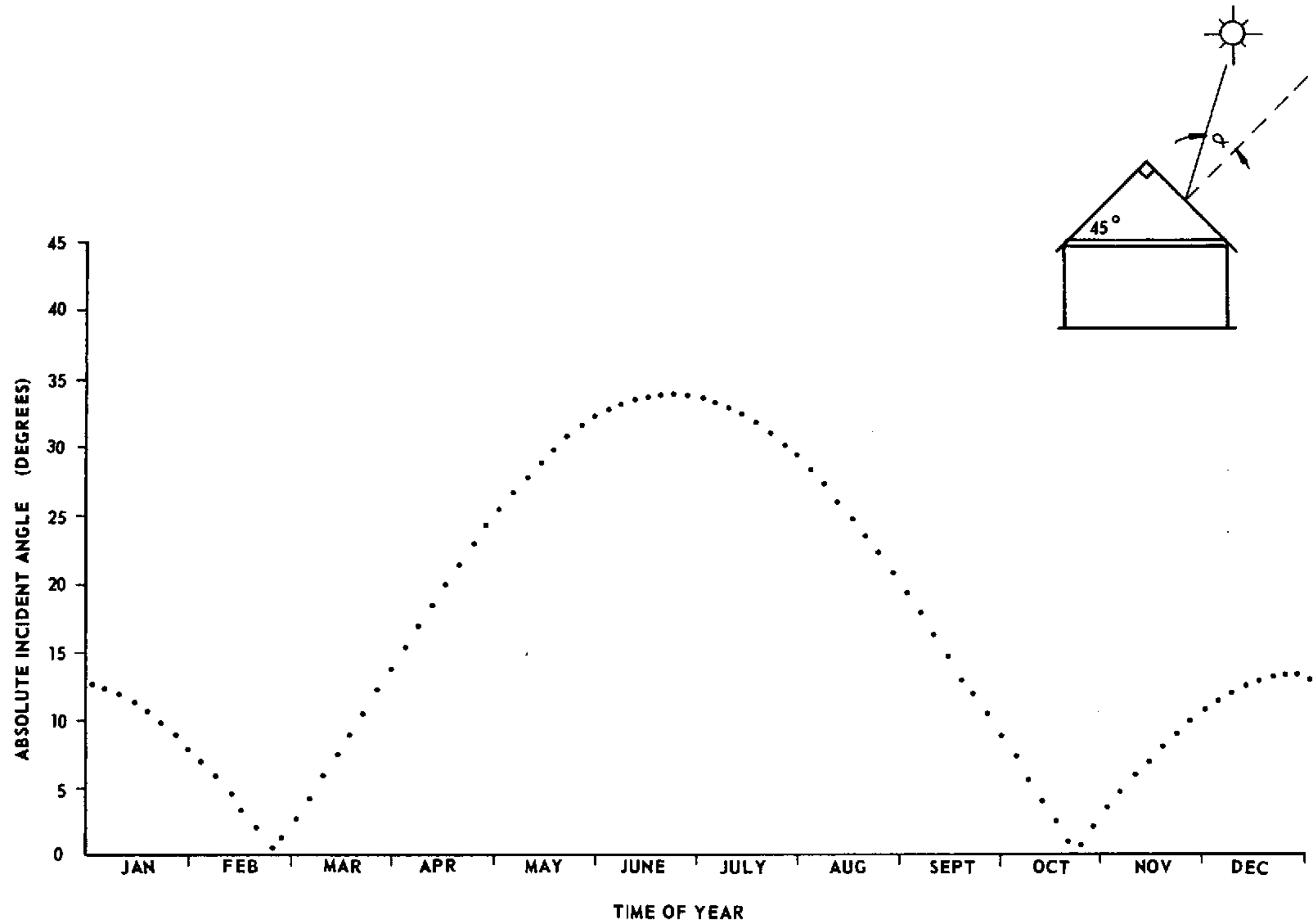


Figure III-8. Absolute incident angle of sun, $|\alpha|$, at solar noon versus time of year.

TABLE III-3. JUNE/AUGUST COLLECTION COMPARISON

	June Collection	August Collection
Date	June 10	August 5
Total Incident (Btu)	2 150 000	2 474 000
Total Collected (Btu)	397 500	612 712
Average Collector Surface Temperature (°F)	187.2	185.7
Average Ambient Temperature (°F)	75	74
Average Wind Velocity (mph)	10.5	6.5

Figure III-9 depicts a typical collector surface inlet and outlet temperature profile. These values are given for days with active and inactive collectors. It is interesting to note that typically when the collector pump initially activates, a drop in surface temperature occurs. This drop is a result of the cooler stagnant fluid trapped in the supply line flowing through the collector. As soon as warmer tank water reaches the collector, a rapid rise in temperature takes place. This initial burst of cold fluid also flows into the air conditioner loop because of the plumbing arrangement. If this occurred when the air conditioner was operating, a momentary loss of air conditioner percolation usually resulted. However, since this cold fluid burst normally lasted only a few minutes, no action was taken to avoid its effects.

During the period when special air conditioner tests were being run, the collection system was disarmed. This caused very high surface stagnation temperatures to occur with the high value recorded being 311°F on the outlet of segment No. 27 on July 3. During periods when the collector is armed, no high surface temperatures are experienced because of the cooling effect of the flowing water.

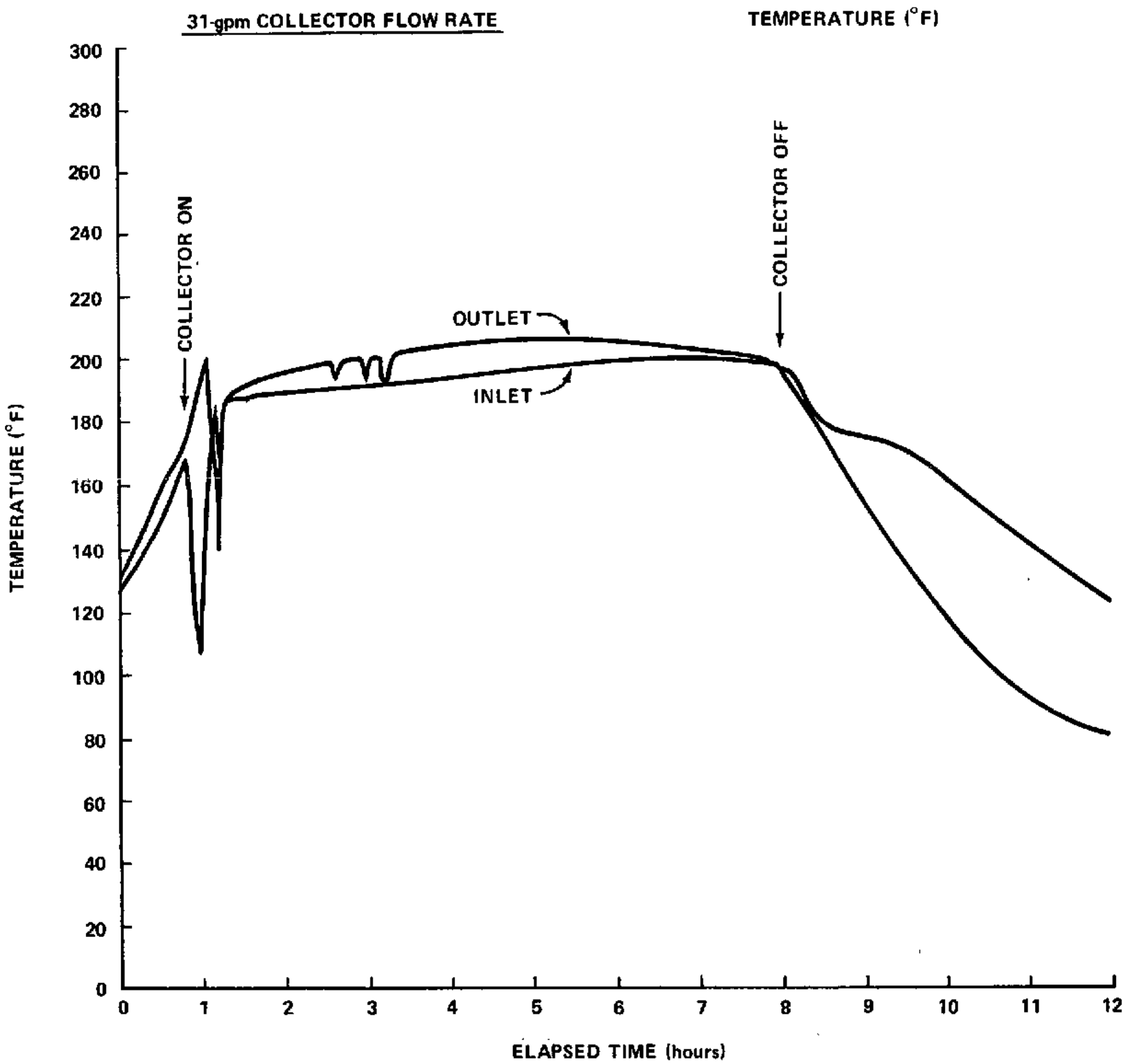


Figure III-9. Collector surface temperatures with collection.

D. Trailer Complex

When making relative comparisons, the energy input to the trailer is extremely important. In determining the comparative size of this system in relation to a typical residential dwelling a number of contradicting factors must be considered. Some factors indicate the cooling load of the trailer complex is higher than an equivalent floor plan dwelling in this locale, and others indicate that this load is less. The shading provided by the collector offers thermal protection from direct sunlight from which a typical trailer or conventional dwelling would not benefit. However, the metal sides, thinner walls, unusually high waste heat levels, and high activity levels within the trailer induce larger energy inputs than a typical residential dwelling of this size would experience. The waste heat generated within the dwelling is excessive for a 1500-ft^2 dwelling.

During the period covered by this report, the high activity level within the trailer complex resulted in numerous openings and closings of the two access doors. A 2-week survey of the frequency of entries indicates an average of 73 entries and exits per day by an average of 28 persons. This high incidence of entries and exits is easily explained, since the complex was used as a work area by test and analysis groups and as a conference area, and was open to large and small tour groups. University cooperative students, who assisted in data reduction and performed facility monitoring tasks, inhabited the facility 24 hours per day. An estimate of 120 man-hours of utilization per day of the trailer is reasonable. Added to this, instrumentation and control equipment housed in the trailers averaging approximately 700 W was operative 24 hours per day. A summary of the lighting and miscellaneous equipment loads shows that a maximum load of 3710 W is possible. Table III-4 summarizes these values.

TABLE III-4. WASTE HEAT SUMMARY

Item	Maximum Wattage	Estimated Percent of Usage	Average Wattage
Lighting	2190	80	1752
Miscellaneous Equipment	320	100	320
Air Conditioner Blower	500	40	200
Instrumentation and Controls	700	90	630
Totals	3710		2902

Integrating the total cooling provided by the air conditioner for a number of representative days and plotting against the average difference in the ambient temperature and the thermostat set point gives the results presented in Figure III-10. The negative values on the abscissa result from using the average ambient temperature for the entire 24-hour day. The fact that the air conditioner came on during cool nights is probably a result of the high internal loads discussed earlier and the baseline of operation. This baseline armed the thermostat 24 hours per day with all windows and doors closed regardless of the outside temperature.

E. Energy Storage Tank

During initial testing, it was noted that in certain flow configurations the temperature delivered to the air conditioner was considerably below the bulk temperature of the tank (as indicated by T-108). From Figure III-11 it is obvious that in the air conditioner only mode the rates of change in the fluid

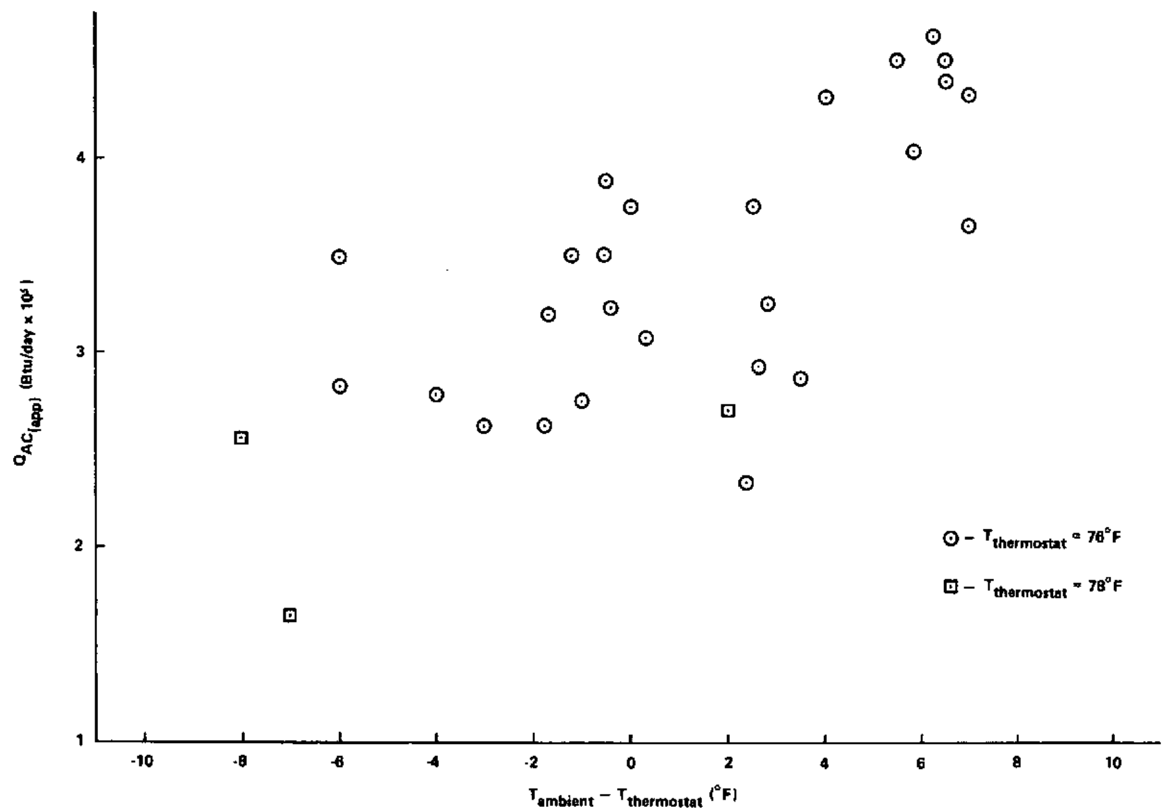


Figure III-10. Trailer cooling load.

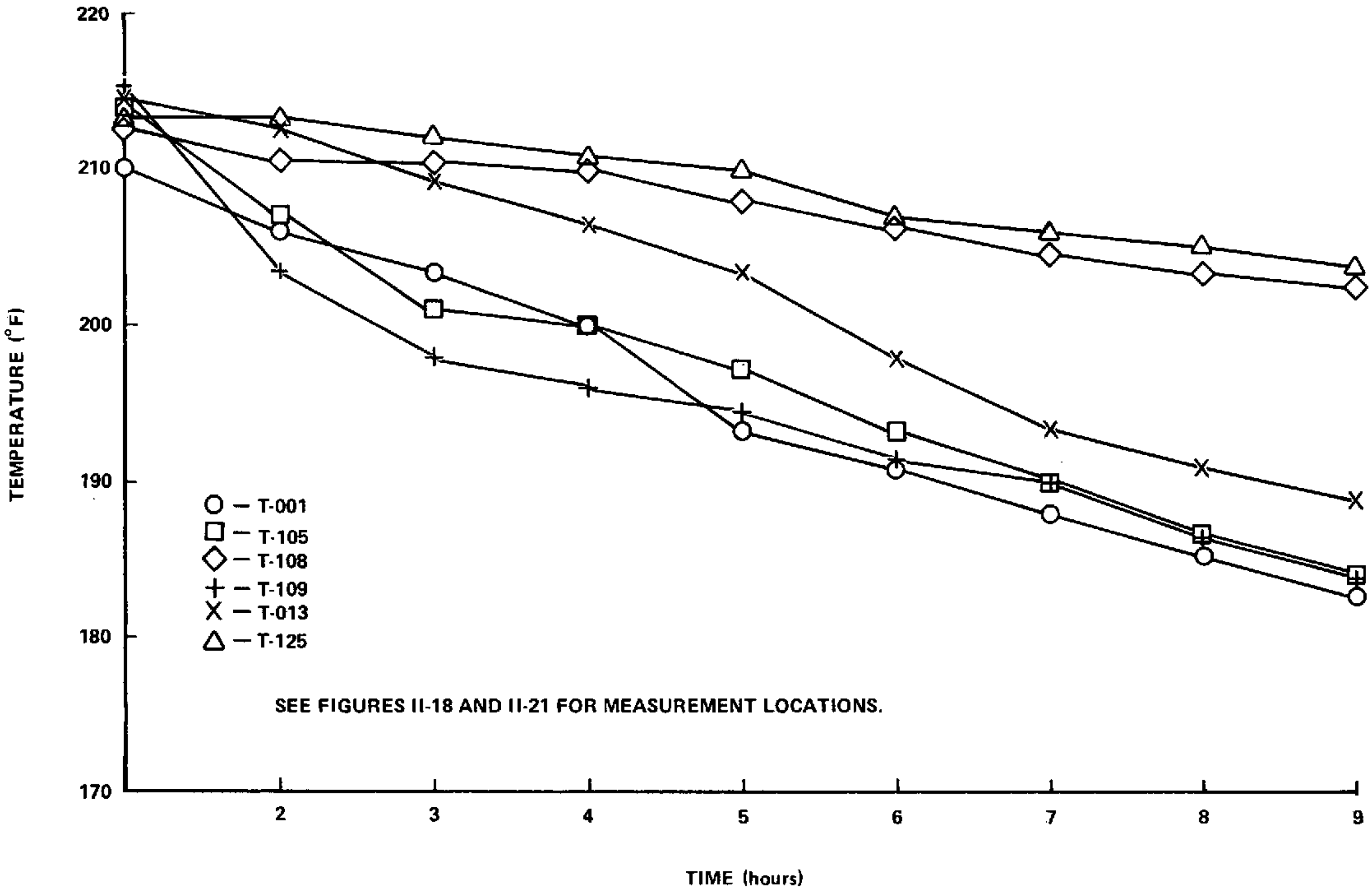


Figure III-11. Energy tank fluid temperature profiles.

temperature near the tank walls (T-108 and T-125) were vastly different from those in the bottom of the tank (i.e., T-105, T-113, and T-109) and near the air conditioner inlet (T-001) (Table III-5). The fact that the temperature gradient of the thermocouple in the fluid line to the air conditioner (T-013) is approximately that of T-001, T-105, T-113, and T-109 indicated that only the cold fluid in the lower tank is involved in the flowing system with the outer fluid having little involvement. The fact that T-001 is cold, for its position in the tank, suggests that the cold air conditioner return flow is "short circuiting" or "channeling" back into the air conditioner inlet (Fig. III-12). The severity of the "short circuiting" is a direct function of time; thus, for cases where the air conditioner runs only for short periods of time, "short circuiting" is not significant.

TABLE III-5. FLUID TEMPERATURE GRADIENTS FOR DATA OF 6-28-74

Measurement	Gradient (°F/hour)
T-001	3.6
T-013	3.1
T-105	3.6
T-108	1.1
T-109	3.9
T-112	1.0
T-113	3.5
T-125	1.0

Note: Flow was 10 gpm for 8 hours. Gradient = $\Delta T/\Delta t$.

However it could be argued that the resulting temperature histories could be partially explained by standpipe conduction. The standpipe, if colder than the surrounding fluid (by conduction), could cause T-001 to be colder. Another partial, yet plausible, explanation for the cold air conditioner supply temperature is that warm fluid is drawn into the air conditioner supply standpipe being cooled as it flows from the tank through the cooler stratified fluid layer in the lower tank. Future tank designs may avoid any recooling by exiting the air conditioner line from the tank top or tank side rather than from the bottom where the colder fluid is located.

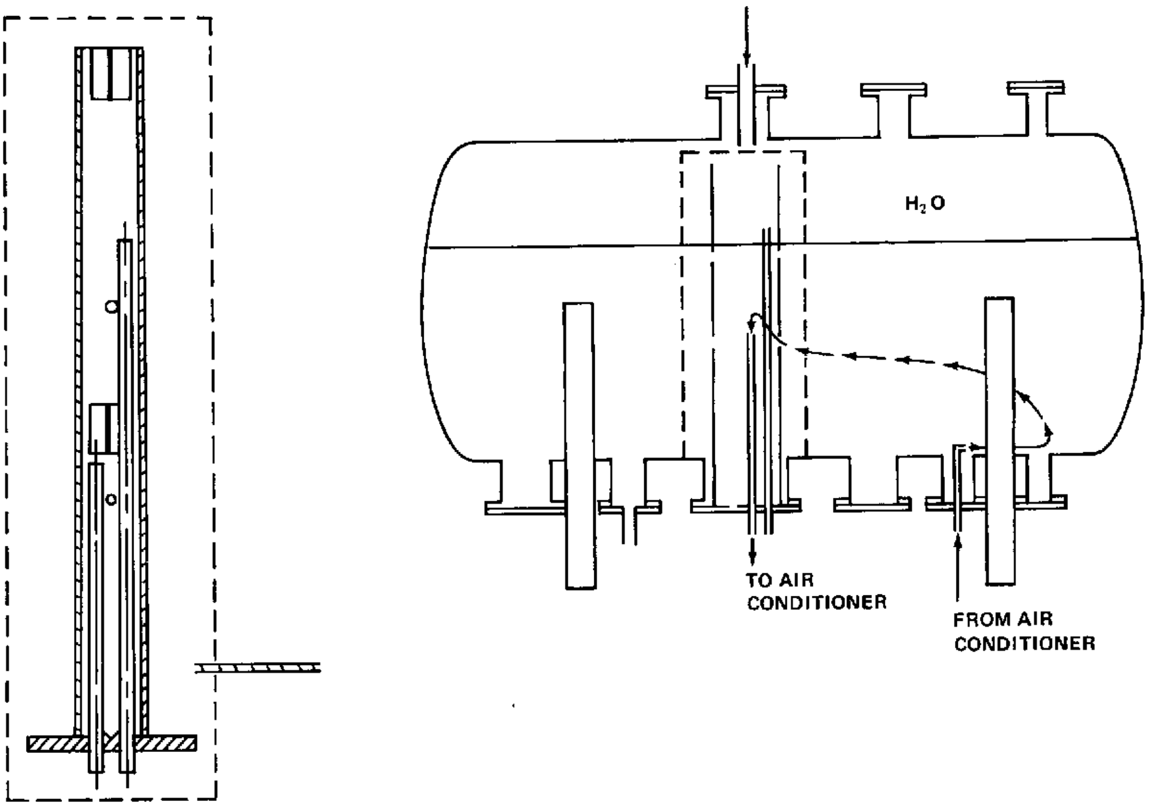


Figure III-12. Energy tank short circuiting — air conditioner only mode.

Since an effort was made in initial design to avoid "short circuiting" (i.e., standpipe-enclosed supply and elbowed return away from supply), the occurrence of this phenomenon was unexpected. Although evidence indicates this phenomenon exists in all possible flow modes, it is partieularly undesirable in the air conditioner only mode. In this mode much of the cold air conditioner outlet return fluid flows directly back into the air conditioner supply line, in effect rendering the inactive warm fluid in the tank useless. Schematics of the "channeling" phenomena for each of the other two possible modes of collector only and collector plus air conditioner are given in Figures III-13 and III-14.

In the case of collection only, the "short circuiting" is less pronounced, with the colder lower tank temperatures strongly influencing the fluid being supplied to the collector. In the collection plus air conditioner mode, "short circuiting" is present between the collector outlet to the air conditioner inlet and the air conditioner outlet to the collector inlet. This is a desirable situation since the colder fluid is being supplied to the collector and the hotter fluid

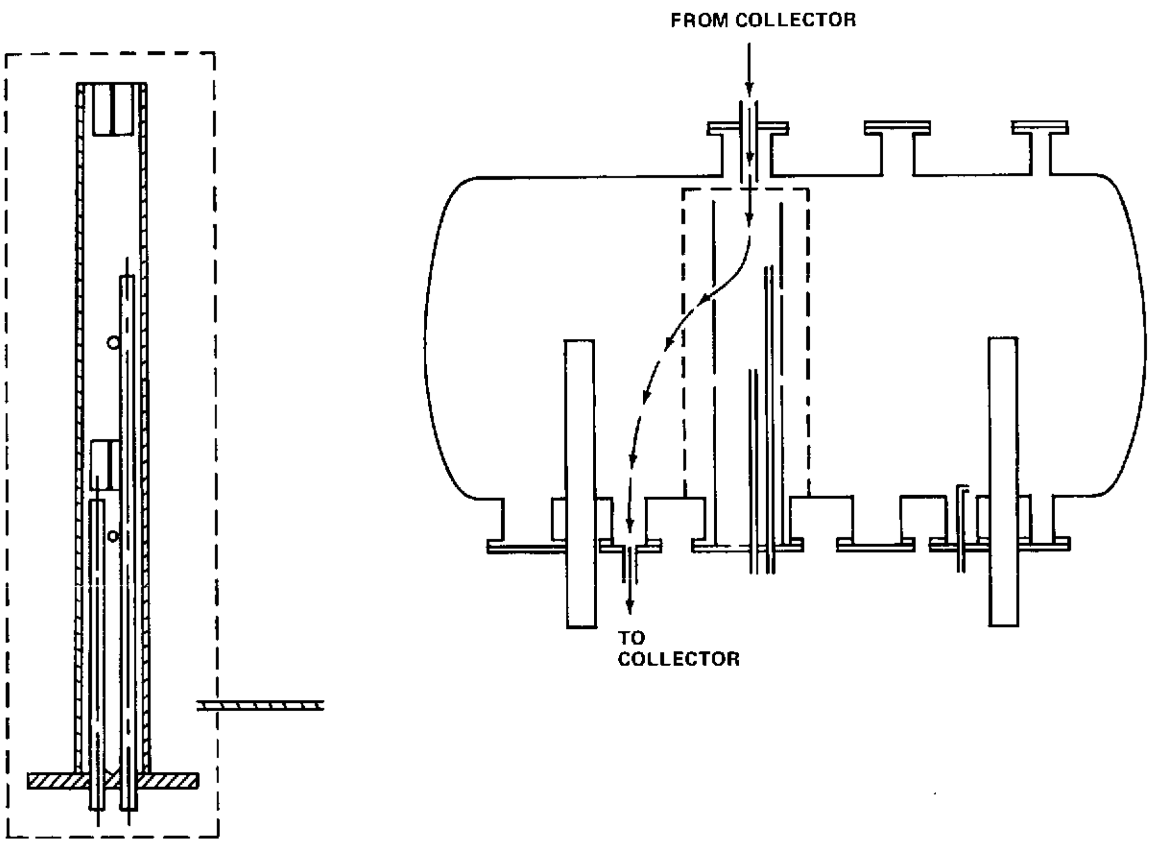


Figure III-13. Energy tank flow path — collector only mode.

is being supplied to the air conditioner inlet. This flow configuration is apparently a result of the counter flow from the collector suppressing the channeling between the air conditioner inlet and outlet.

For the case represented by Figure III-14, the cold fluid entering the collector caused premature cutoff of the collector system during early tests. T-001, in the tank standpipe, was initially used to reference the collector outlet temperature for collector pump deactivation. When the collector outlet temperature reached 5° F above T-001, the pump activated, deactivating as the outlet temperature fell to the same value as T-001. However, since the temperatures in the tank were stratified with the cooler fluid at the collector inlet and because of the "short circuiting" from the air conditioner return to the collector inlet, fluid much colder than T-001 was being supplied to the collector. As a result, although the fluid was being warmed as it passed through the collector, outlet temperatures were less than T-001, causing premature pump cutoff. This cutoff was undesirable because any increase in fluid temperature in

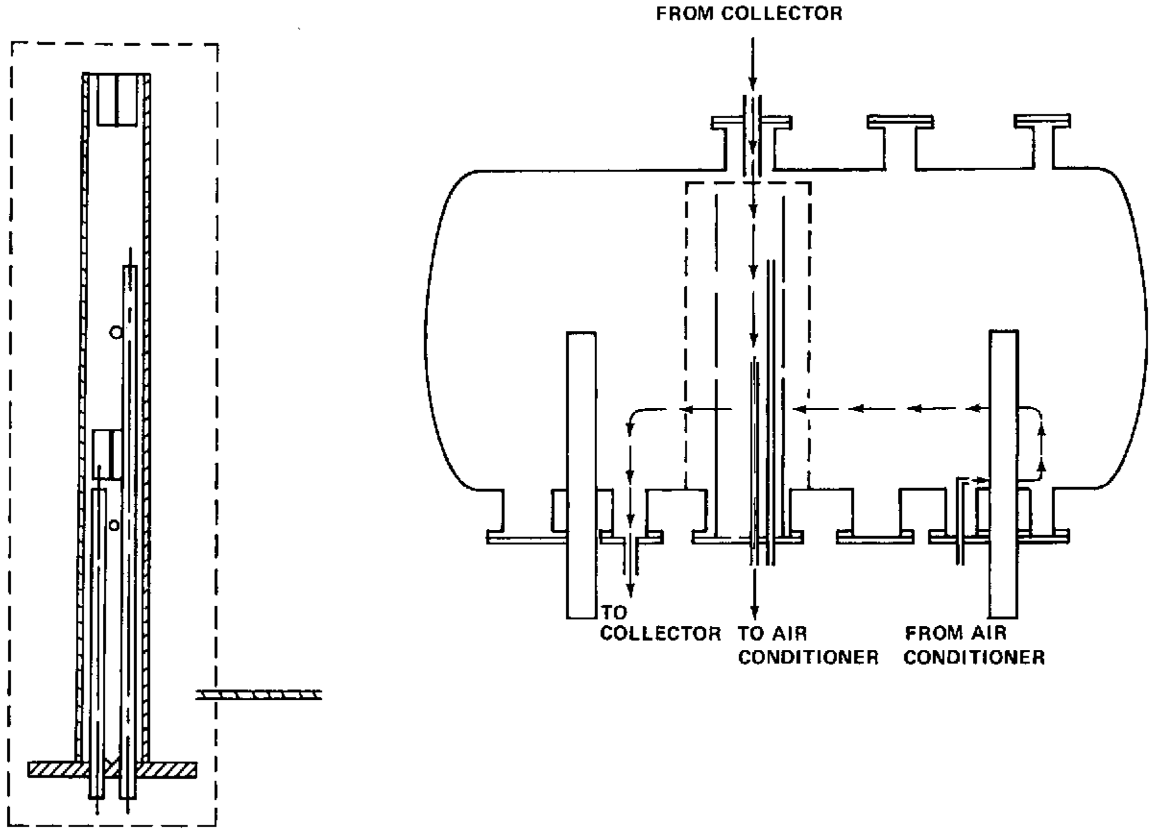


Figure III-14. Energy tank flow path — air conditioner and collector combined mode.

the collector represents an energy addition to the system (i.e., if T-004 > T-003, then $Q_{stored} > 0$). As a result, the tank reference temperature was switched to T-105, which tracked the collector supply temperature better allowing the collector to flow at colder supply temperatures. However, in the future it would be desirable to install a thermocouple at the collector inlet to assure maximum collector on-time.

Temperature stratification was observable in the tank when the air conditioner was running without the collector. Stratification is a result of the colder fluid flowing from the air conditioner outlet into the bottom of the tank and not mixing with the warmer fluid in the upper tank. Stratification was evident at all flow rates, but was more pronounced at lower flow rates. Using the lowest temperature, T-105, the temperature decrease after 1 hour of air conditioner on-time without collector operation and at each of four air conditioner flow rates is given in Figure III-15. Since the energy being removed by

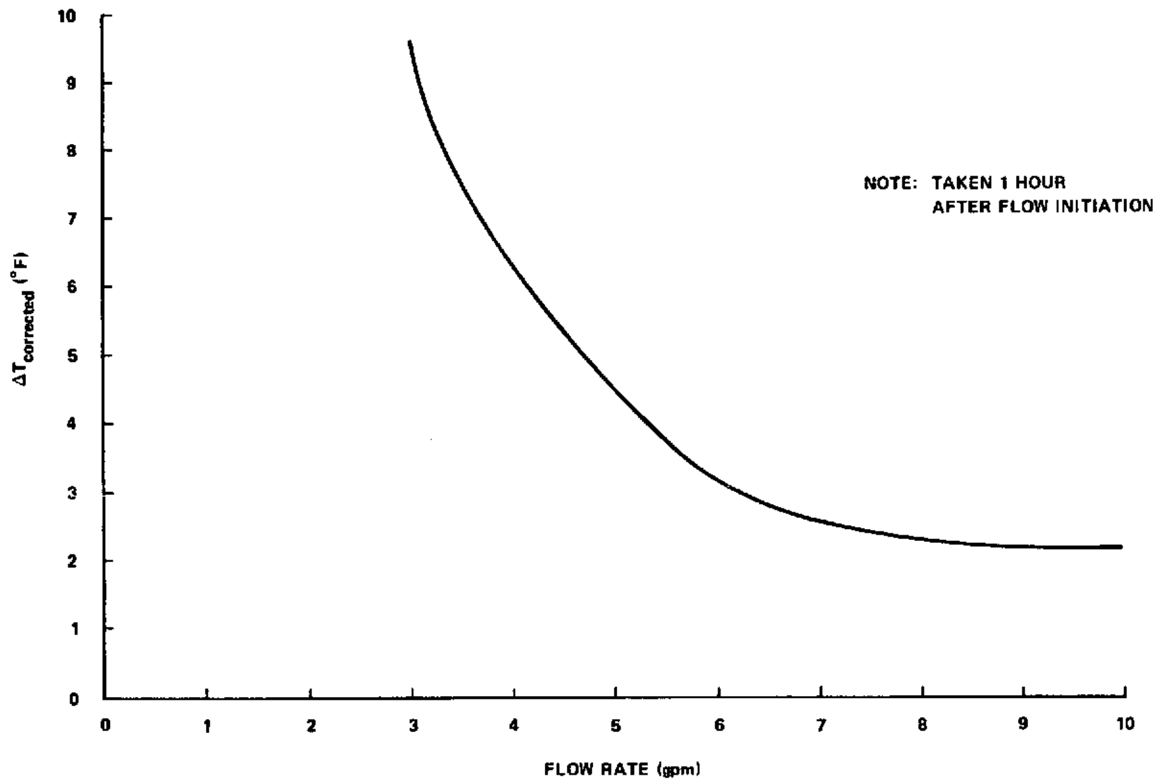


Figure III-15. Corrected stratification temperature differential versus air conditioner flow rate.

the air conditioner was lower for lower flow rates, different temperature gradients were to be expected; consequently, the actual temperature drop was corrected for each case by subtracting the temperature drop that would have been caused by the energy loss if the tank acted as a uniform mass, $\Delta T_{corrected} = \Delta T_{actual} - \dot{Q}_{gen}/\dot{m}c_p$.

From these data it is obvious that stratification is a desirable phenomenon for this plumbing configuration because it causes the warmest fluid to be supplied to the air conditioner and the coolest to be available at the collector inlet. As a result, operating at lower air conditioner flow rates has the added benefit of causing colder fluid to be supplied to the collector.

Future studies are planned to determine the advisability of adding a direct flow path between the air conditioner outlet and the collector inlet. This would allow the colder air conditioner return fluid to enter the collector when both are operating. However, the advantage of such a modification must be weighed against the cost of added hardware.

F. Hardware Performance

No significant leakage problems occurred in the deionized water subsystems after startup of the system in late May 1974. Prior to system startup, however, leakage occurred during proof pressure tests at several galvanized pipe joints, at a weld in the storage tank, and at rubber hose connections between the solar collector heat exchangers. The original thread die used on the galvanized pipe produced poor quality threads so it was necessary to rethread several galvanized pipe connections to repair the leaking areas. The storage tank was weld-repaired in place. The hose connections on the solar collector were replaced with aluminum nipples welded between the heat exchangers; the collector segment inlet and outlet hoses were replaced with 0.375 in. outside diameter aluminum tubing expansion joints. These changes corrected the leakage problems in the deionized water subsystem.

Water leaking into the collectors also occurred during early testing as was discussed previously in subsection III.C.

On four occasions during the period covered by this report the instrumentation system was "knocked out" (i.e., amplifiers burned out) by lightning strikes in the vicinity of the house. On an overcast evening, one of the test personnel observed an electrical discharge between a low cloud and the metal surface of the collector which caused a low popping sound. Lightning arresters have been installed on the roof, but insufficient time has elapsed since installation to properly assess their effectiveness.

The external insulation on the supply and return lines interconnecting the solar collector and storage tank developed longitudinal cracks after initially heating the deionized water to approximately 210° F. The polyurethane insulation distorted enough to break the external mastic seal. The above-ground portion of the lines was repaired by applying a fiberglass cloth wrap on the lines sealed with a mastic coat. Further reinforcement was provided with 1 in. wide metal bands applied approximately 1 ft apart along the length of pipe. No further insulation cracks have developed. The underground pipe insulation may be cracked also, but no significant heat loss has been observed; therefore, excavating and repairing the lines is not warranted.

Several electrical control problems occurred. A relay meter "stuck" twice, on one occasion preventing the solar collector pump from starting and on a second occasion causing the solar collector pump to remain on after the cutoff point was reached. Replacement of the relay meter corrected the problem. Two relays failed in a thermoelectronic controller that controlled the auxiliary heaters ON-OFF points.

Numerous control system anomalies occurred as the result of electronics controller set point changes. These controllers have manually adjustable dial set points which trip causing functions to be performed, such as valve or pump cycling. Controllers are used to execute all logic control. The set point changes are the result of displacement of the set points by either vibration, inadvertent physical contact, or electronic drift. In all cases corrective action was simple readjustment of the set point dial. The age of the equipment and its previous use in other testing is considered to be a contributing factor to the high frequency of failures.

G. Corrosion

Even though sodium chromate inhibitor was added to the water loop, a number of valve failures occurred that have been attributed to corrosion buildup interference with normal valve operation. A total of nine valve malfunctions occurred in the AC/Htr loop during the testing period. All these valves were 1-in. Hydromatics aluminum valves with carbon steel ball type plugs. These valves were in-house available hardware and were at least 8 years old. The failure mode of all valves was either in the form of burned out actuators or failure of the valve operator to rotate. Post-test failure inspection of the internal parts of the valves indicated severe corrosion (Fig. III-16), especially on the carbon steel ball. The extensive corrosion observed in all valving within this loop indicated that most operator electrical failures were a result of excessive friction in the valve inducing high current loads within the actuator motors. Failure analysis indicated that a galvanic corrosive action occurred between the valve's aluminum body and the chrome-plated steel ball. New valves are being ordered to replace existing hardware. Inspection of water samples and fluid filters indicates corrosion has been confined to these valves.

H. Heat Losses

In a system this size, large heat losses may be encountered. Even in the case where systems are well insulated, heat shorts inherent in structural support and large surface area to volume ratios are significant. The exact value of heat losses in lines and components is difficult to obtain directly because the losses are reflected by only very small fluid temperature changes. As a result, the total heat loss for this system was assessed by using a total energy balance given in the following equation:

$$Q_{loss} = Q_{col} + Q_{\substack{\text{out of energy} \\ \text{storage tank}}} - Q_{\substack{\text{absorbed by air conditioner} \\ \text{from energy storage tank}}}$$

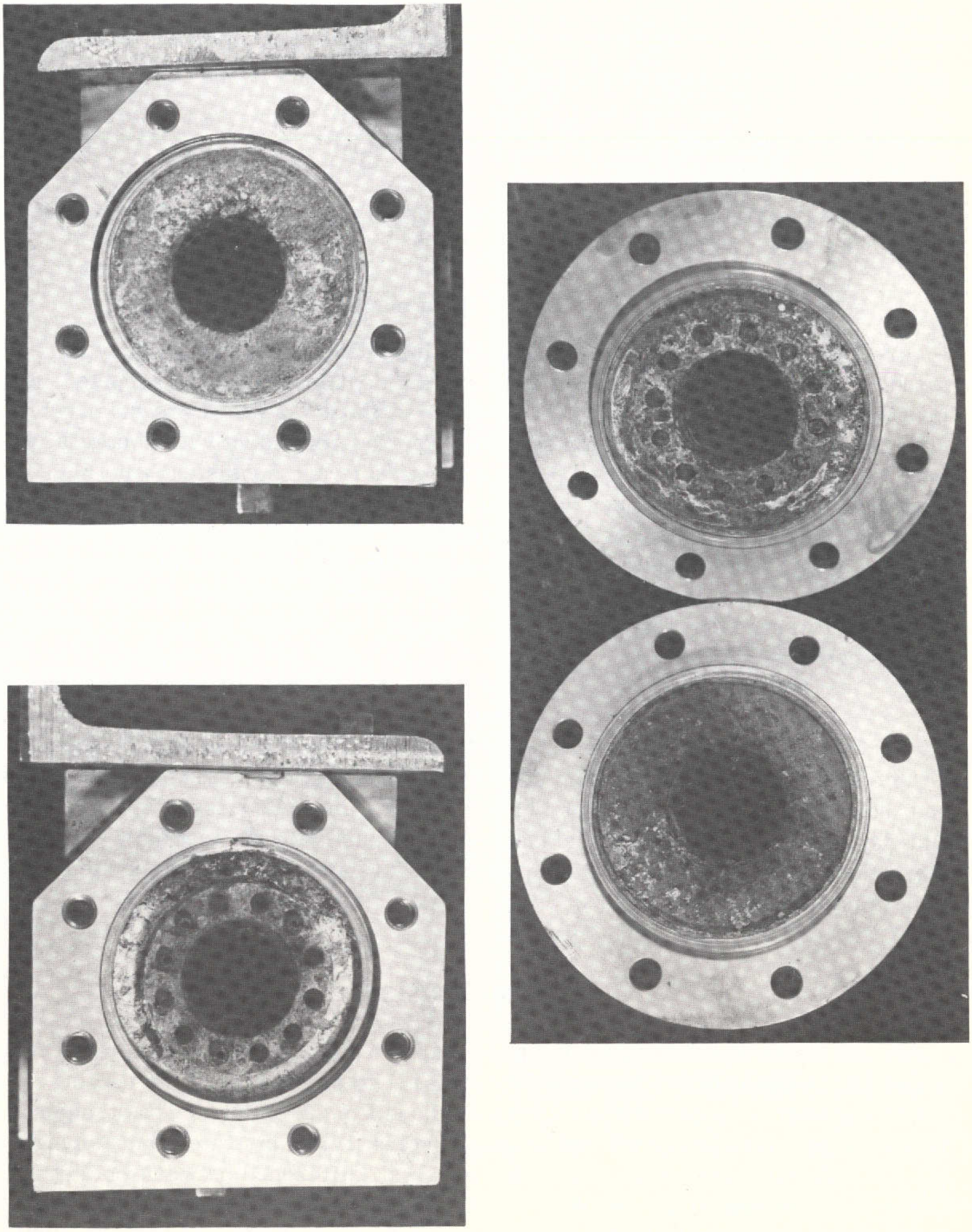


Figure III-16. Air conditioner fluid loop valve corrosion.

The energy equation terms will be designated by Q_{col}; $Q_{\text{out of energy storage tank}}$, energy loss from the water tank (i.e., $mc_p \Delta T$ of water in tank); and $Q_{\text{absorbed by air conditioner from energy storage tank}}$ (i.e., $Q_{gen} - Q_{aux\ htr}$, since energy is transferred into the air conditioner from heat supplied by the energy storage tank and/or the supplemental auxiliary heaters).

A plot of daily values of Q_{loss} for a number of days is given in Figure III-17. From this figure, a typical loss rate of 180 000 Btu/day is found. For the 48 days for which total system heat loss data are reported, the loss for 12 days exceeded 200 000 Btu/day. Of these days 11 were relatively high collection days, and 8 of these days occurred coincident with abnormally long air conditioner run times. Both of these facts are consistent with high heat leakage. However, there were days for which high collection and long air conditioner run times occurred in which the indicated heat leakage was not excessive. Unfortunately the instrumentation is not sufficiently accurate to determine exact losses and locations. However, the addition of high fidelity instrumentation to air conditioner and collector lines would allow more definitive assessment.

By using energy tank temperatures during a number of dormant nighttime periods where tank energy loss is easy to assess, the heat loss can be found (Table III-6). This loss is from 40 000 to 65 000 Btu/day. Pretest calculations had indicated a total loss rate of 160 000 Btu/day and a tank loss rate of 45 000 Btu/day, both of which are in very close agreement with the test data. Using these tests and analytical data comparisons, it is apparent that the system heat leak is not excessive.

I. Logic System

Prior to initial testing, the heater control temperature was altered from a wet bulb temperature reference (Fig. III-18) to one that used the cooling tower water return temperature to the air conditioner, T-020. This temperature was used because of the uncertainty involved with accurately measuring the wet bulb temperature and the expected small difference between cooling tower outlet temperature and the wet bulb temperature. This temperature was found to be, in general, higher than the wet bulb temperature so that the auxiliary heaters were operating at a higher hot water temperature than necessary. This resulted in air conditioner hot water inlet temperatures averaging 210° F to 215° F. As a result, the control system was altered to use a 125° F bias on T-020 rather than the 130° F bias initially designed when a wet bulb reference was used. This change was incorporated on May 27.

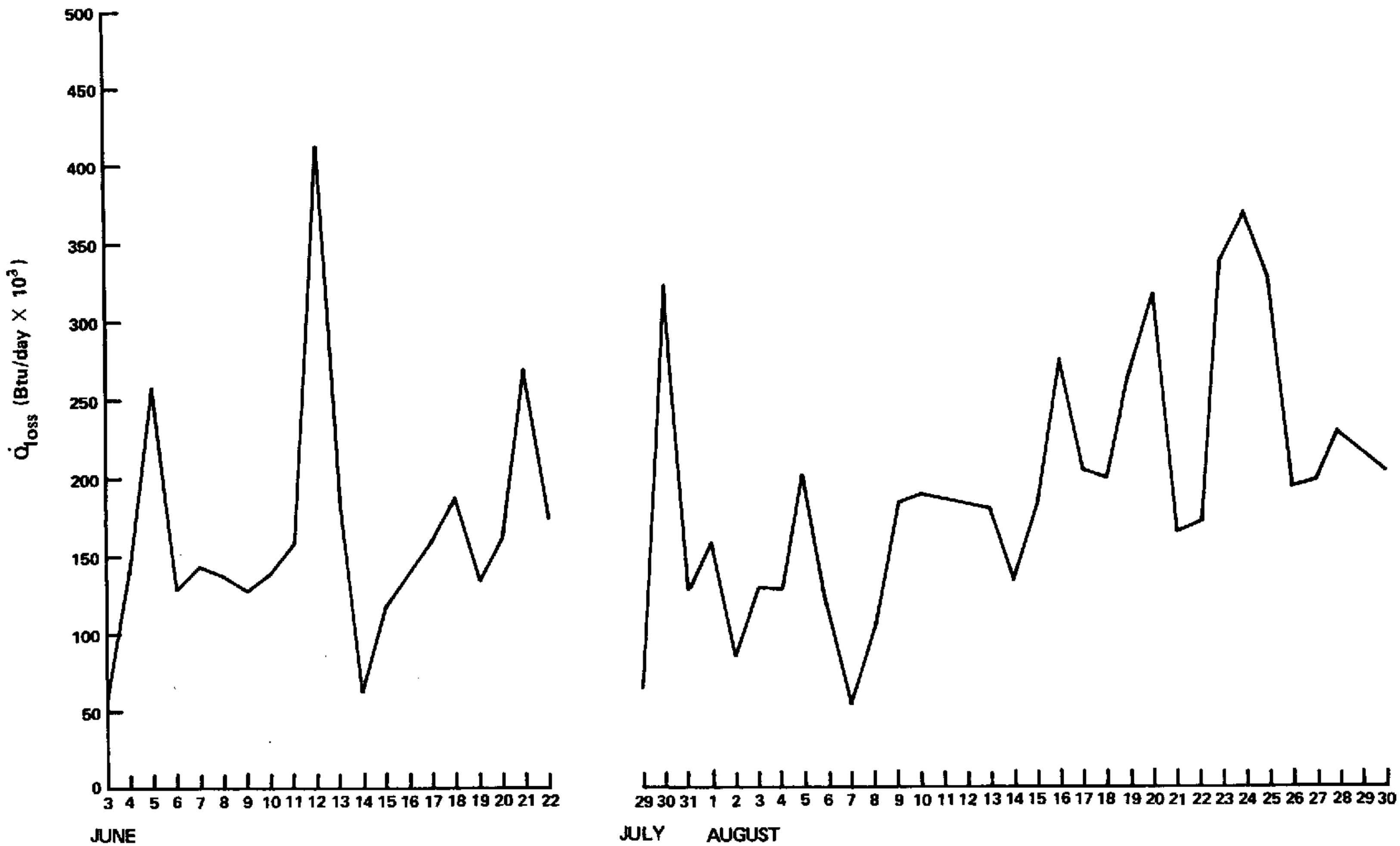


Figure III-17. System heat loss.

TABLE III-6. ENERGY TANK HEAT LOSS

Date	Heat Loss (Btu/day)
6-4	65 300
6-24	40 000
7-3	43 000
7-4	60 000
7-17	43 077

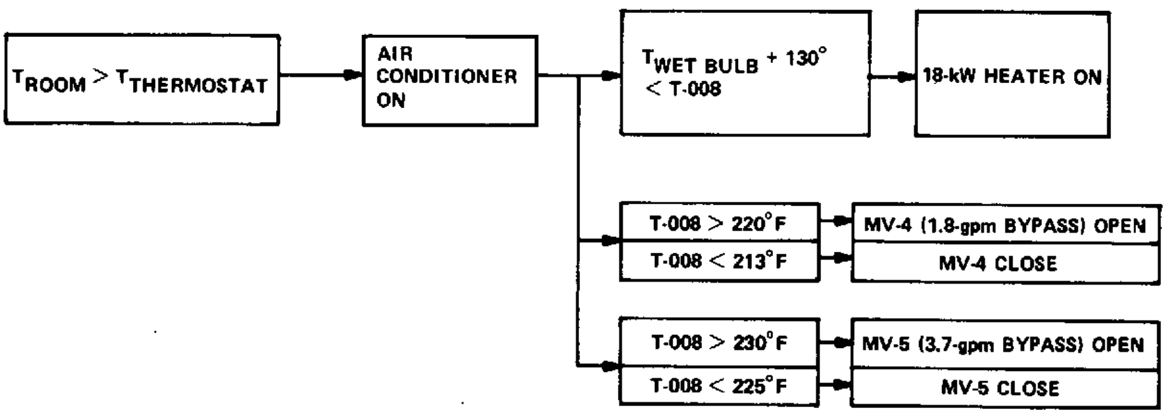


Figure III-18. Original logic system design.

Afterward, testing indicated high energy consumption by the air conditioner. Following this, the manufacturer was contacted to determine an absolute minimum temperature at which the unit could function properly; a value of 195° F was recommended. From this a new heater control logic (Figs. III-19 and III-20) was incorporated in the system on June 2. Four days later, a 60-sec delay was added to avoid startup transients associated with cool, stagnant, low temperatures occurring at startup. The improvement resulting from the control change can be seen by comparing the average power input by the auxiliary heaters at a 195° F tank temperature before and after the change (Fig. III-21).

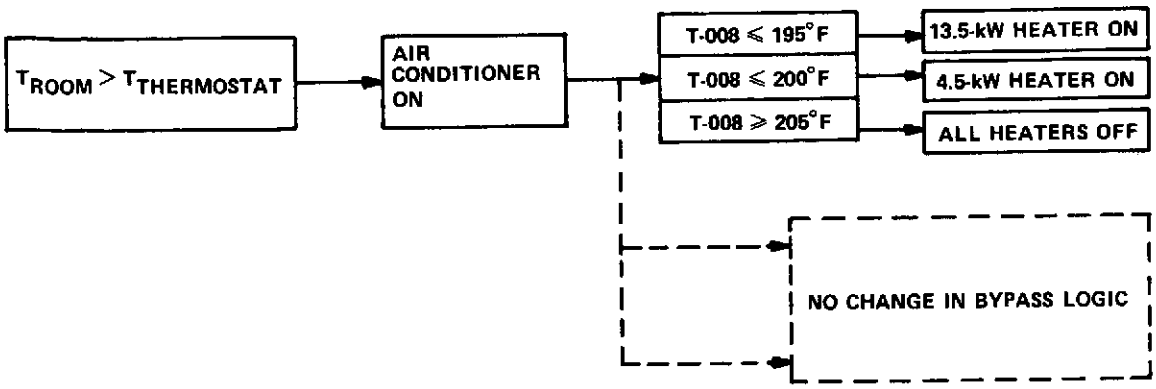


Figure III-19. First logic system modification.

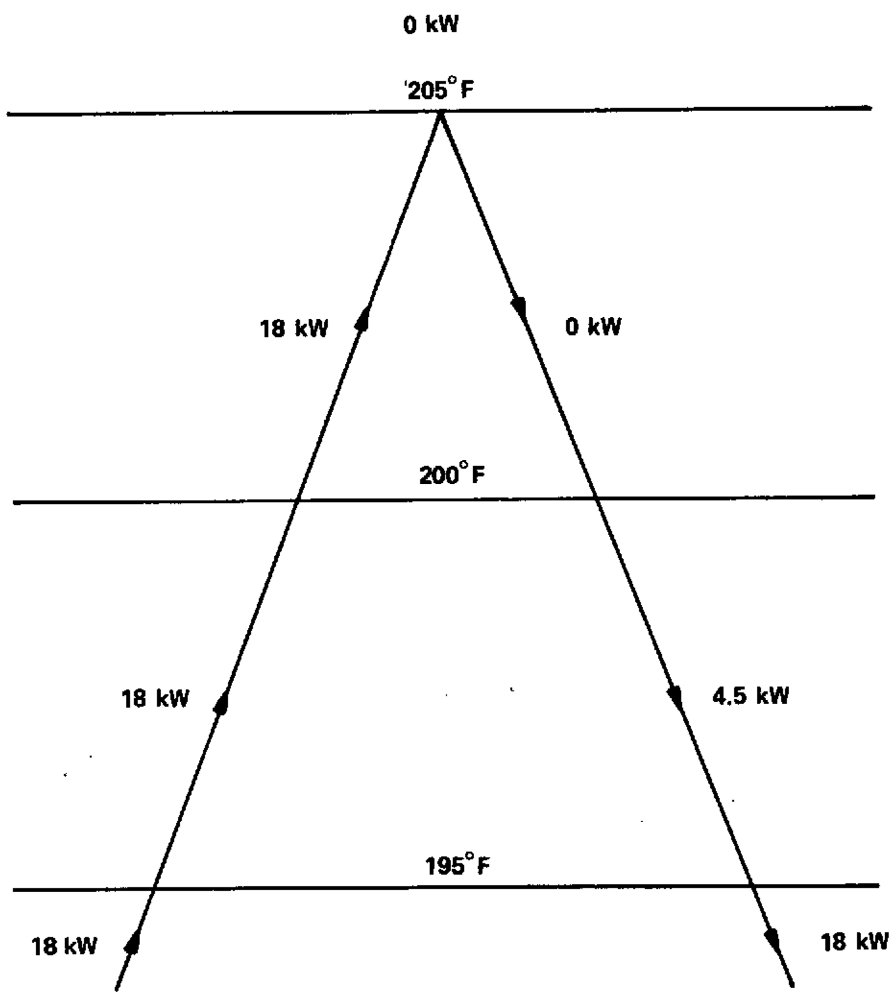


Figure III-20. Heater control logic scheme 1.

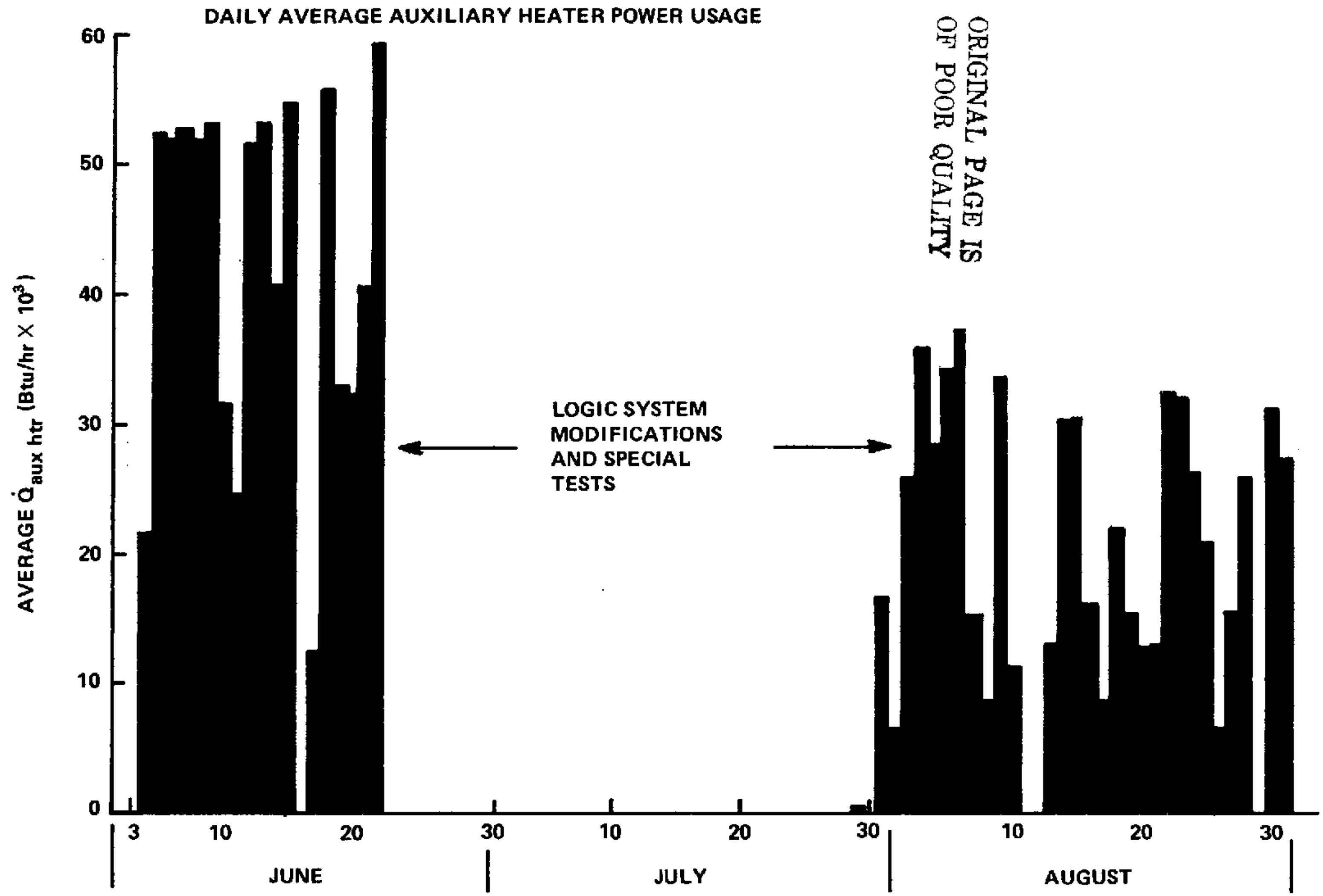


Figure III-21. Power useage before and after modifications.

On June 10, a modification was performed to the logic controlling the air conditioner bypass flow. This modification was performed to avoid excess energy usage because of reheating spillover cooling capability. The logic was controlled by a temperature sensor (T-037) on the condensate (cold water) spillover line (see Fig. II-20). As excess cooling capability is produced in the air conditioner, cold water spills over back into the regenerative heat exchanger. This excess water results when the boiling rate and, consequently, the expansion valve upstream pressure increases causing an increased water vapor circulation rate. This cold fluid must be reheated in the generator. As a result, any flow through this line represents wasted cooling capability. These enhanced cooling conditions cause more cool water to flow through the evaporator than can be boiled by the warm air passing through the evaporator coils. For this reason, a sensor was placed on this line which activated both bypass valves open (i.e., MV-4 and MV-5). This caused lower generator flow, less energy input through the generator into the air conditioner loop, and, consequently, less production of excess cold water. The sensor was initially set to trigger bypass at a spillover line temperature of 50° F; on June 11 this temperature was raised to 55° F. This bypass tended to occur for only a short period of time, usually at initial air conditioner startup. The energy savings from this modification proved to be small.

On June 17, additional steps were taken to further reduce the auxiliary heater operating time. Figures III-22 and III-23 depict the operating control mode.

After these modifications, the energy utilization continued to be unacceptably high. As a result, a major examination of the air conditioner performance was found to be necessary. The new logic system resulting from the findings of these tests is discussed in subsection III. L.

J. Air Conditioner Performance

The primary problem encountered with the air conditioner system previously described was that the unit used an excessive amount of energy, causing exorbitantly high auxiliary heater power usage. The rate of energy usage was found to be a strong function of hot water inlet temperature to the air conditioner. These high levels were unexpected since manufacturer's data indicated an energy consumption rate of only 52 000 Btu/hour. Because of uncertainty as to the accuracy of these rate measurements, an instrumentation accuracy survey was performed. This survey proved these rate values to be accurate within ±5000 Btu/hour.

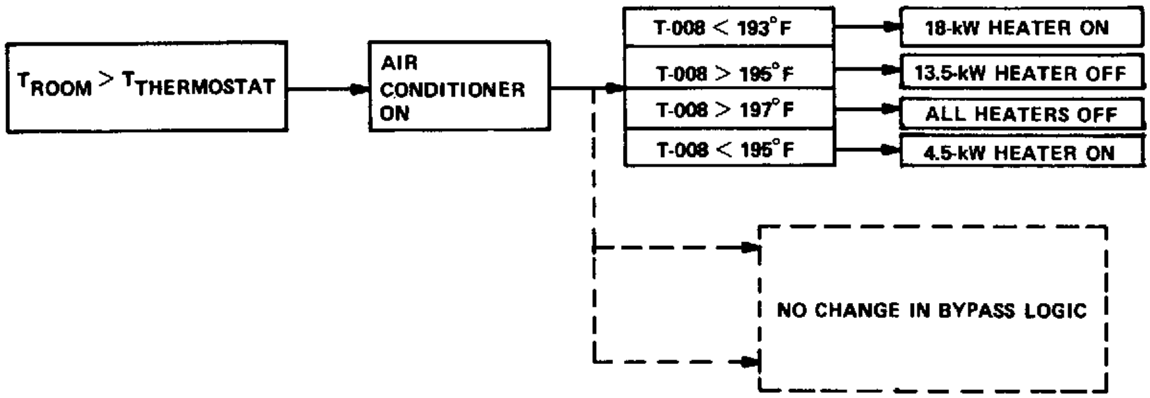


Figure III-22. Second logic system modification.

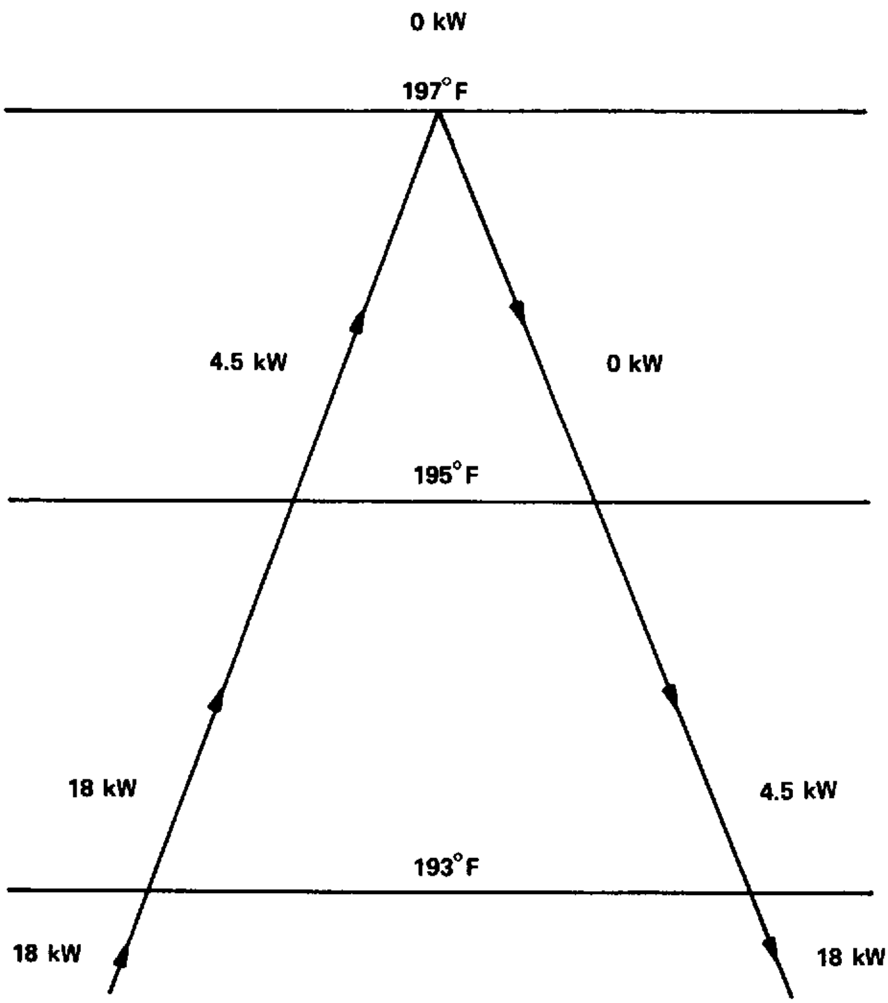


Figure III-23. Heater control logic scheme 2.

One of the strongest evaluation tools for air conditioner performance assessment is the coefficient of performance, COP, defined by the equation

$$\mathrm{COP} = \frac{\dot{Q}_{AC}}{\dot{Q}_{gen}}$$

where $\dot{Q}_{AC}$ is the cooling rate output from the air conditioner, and $\dot{Q}_{gen}$ is the heating input to the air conditioner. The manufacturer's data (i.e., based on a 60-percent relative humidity, a dry bulb temperature of 80° F, with a cooling tower temperature of 85° F) reported a COP of 0.67. However, using test data from test start through June 1, COP's ranging from 0.41 to 0.43 were found.

From these tests, it was also noted that the cooling rate of the air conditioner is a function of evaporator inlet temperature and relative humidity as well as cooling tower supply temperature to the air conditioner. Cooling rates for the initial inlet temperatures tested ranged from 30 000 Btu/hour to 41 000 Btu/hour.

Because of the indication of cooling rates lower than the 36 000 Btu/hour nominal rate, a gas flow rate check was performed. The results of this check indicated the nominal flow rate was 1650 cfm, compared to the value of 1200 cfm quoted by the manufacturer.

In an effort to minimize energy loss from the air conditioner, it was insulated with common household fiber glass insulation around the absorber and generator. Although it is believed that this decreased the heat leak, it was not possible to quantitatively assess the change in performance.

In determining the rate of cooling from the air conditioner, it was found that the relative humidity indicated by the gage located immediately behind the instrumentation panel, H-003, and adjacent to the air conditioner was considerably lower than the humidity level existing at the air conditioner inlet. Relocation of this gage into the inlet duct improved the steady state performance estimates using the evaporator parameters. However, during cyclic operation of the air conditioner it was noted that the relative humidity sensors lagged the actual profile considerably, causing significant errors in cooling calculations when using this measurement. For this reason, the air conditioner performance was determined using the following equation

$$\dot{Q}_{AC_{(app)}} = \dot{Q}_{cooling\ tower} - \dot{Q}_{gen} \quad ,$$

where $\dot{Q}_{AC_{(app)}}$ is the cooling rate using the above equation with (app) indicating an apparent value assuming no external heat gain other than that of the generator and evaporator. $\dot{Q}_{cooling\ tower}$ is the heat removed by the cooling tower measured at the air conditioner. This equation results from the fact that the cooling tower removes all energy inputs to the air conditioner.

During the June 26 test, a number of low evaporator temperature cutoffs were experienced. These all occurred at a low room temperature (i.e., less than 68°F) and low cooling tower temperatures. Consequently, repetition of this event is not expected. On July 10, at elevated generator inlet temperatures (i.e., 223°F to 210°F) the so-called "prevent protection system" activated to shut off the air conditioner. This system was installed by the manufacturer to avoid operation when fluid was blown out of the liquid trap (Fig. II-19), thereby allowing hot, high pressure water vapor to reach both sides of the expansion valve. When this is allowed to occur, the air conditioner functions as a heating system rather than a cooling system. The cause of this occurrence is related to the fact that at high temperatures the differential pressure across this "U-tube" system increases, eventually decreasing the liquid level sufficiently to allow gas to blow by. As the hot gas blows by the cutoff sensor, it reaches a high temperature trip point deactivating the evaporator air circulation fan and the AC/Htr loop pump. This "prevent" is activated by a "strap-on" temperature gage that was set at the manufacturer's plant before insulation was applied to the air conditioner. Apparently, the addition of extra insulation causes the line temperature to rise to higher temperatures than normal causing an early cutoff.

K. Special Air Conditioner Tests

Based on the data accumulated from test initiation to June 23, it was apparent that the air conditioner was consuming excessively high levels of energy especially at higher generator inlet temperatures. At this time it was considered imperative to lower the energy consumption. Testing was terminated and studies of the absorptive system thermodynamics indicated the air conditioner COP would increase at lower generator temperatures and flow rates. However, the key to operation of the unit at these reduced levels was to assure percolation. Within the envelope of existing hardware a lower inlet temperature and/or a bypass flow rate keyed to the inlet temperature were the only feasible options. However, since very little data were available from the manufacturer, the exact effects of changing the flow rate through the air conditioner and lowering the inlet temperature were not known. As a result a series of tests was mapped to assess these effects.

Testing was begun on June 25 and continued through July 10 with interruptions only on June 27 and the weekend days of June 22, 23, 29, and 30 and July 6 and 7. Eleven tests were run in this time period with the primary objective being to determine the energy consumption by the air conditioner at a variety of flow rates through the generator over a range of temperatures expected to be encountered. In addition, the temperature at which the air conditioner ceased functioning was another important test parameter. A compilation of these tests is given in Table III-7.

Unfortunately, the solar house facility was not designed to control inputs to the air conditioner. Consequently, test data are very scattered, which prevents exact determination of data trends and causes for variations. In addition the limited time allocated for these tests precluded acquiring a full range of data for each flow rate. In particular, the gas inlet temperature and relative humidity to the evaporator and the cooling tower temperature were not controlled variables. All tests were run with the heater and collector logic disarmed to avoid rapid variations in generator inlet temperatures. Typically the energy storage tank temperature was brought up to the desired start temperature during the night. The temperature decay rate into the generator was also uncontrolled, dictated by the rate that the air conditioner used energy and by the effect of fluid channeling in the energy tank.

The range of inlet temperatures tested was from 181° F to 223° F. The flow rate through the generator was set at 10 gpm, 7 gpm, 5 gpm, 3 gpm, and 1.7 gpm. These discrete levels were attained by controlling the total flow rate in the AC/Htr loop by using a throttling/bypass valve arrangement.

Data resulting from these tests are plotted in the form of energy absorbed by the air conditioner, $\dot{Q}_{gen}$, the apparent rate of cooling provided by the air conditioner, $\dot{Q}_{AC_{(app)}}$, and the coefficient of performance, COP, for varying inlet temperatures (i.e., T-008) and flow rates (F-003) into the air conditioner. The band formed by these data is plotted in Figures III-24 through III-36 for the limited test conditions applied. It is obvious from these plots that smooth repeatable variations did not result. However, it is conjectured that a complete set of tests over a full range of cooling tower and house temperature/humidity conditions will produce a more uniform band of data. Because of the

TABLE III-7. SPECIAL AIR CONDITIONER TEST RESULTS SUMMARY

Test No. (Date)	Hot Water Flow Rate (gpm)	Cooling Tower Water Supply Temperature (°F)	Trailer Inside Air Temperature (°F)	Hot Water Temperature[a] at Percolation Loss (°F)	Restarts
6/25	10	68.0	74.5	183.0	
7/3	10	77.9	89.8	185.8	
7/2	10	70.5	80.7		Failed without heaters at 191.1°F Successful with heaters
7/3	10	78.0	82.0		Successful without heaters at 186.7°F
7/3	10	78.7	91.6		Successful without heaters at 188.2°F
7/3	10	76.9	86.6		Failed without heaters at 184.4°F Successful with heaters
7/2	7	78.0	87.1	188.3	
6/28	7	72.6	75.3	186.5	
7/2	7	78.0	87.1		Successful without heaters at 187.3°F

TABLE III-7. (Continued)

Test No. (Date)	Hot Water Flow Rate (gpm)	Cooling Tower Water Supply Temperature (° F)	Trailer Inside Air Temperature (° F)	Hot Water Temperature[a] at Percolation Loss (° F)	Restarts
7/3	7	77.9	84.7		Successful without heaters at 191.7° F
7/4	5	78.8	81.9	188	
7/4	5	75.7	81.6		Successful without heaters at 193.9° F
7/4	5	78.4	82.2		Failed without heaters at 185.2° F
7/8	3	76.8	82.5	192.7	
7/8	3	76.8	82.5		Failed without heaters at 192.7° F
7/9	3	78.8	85.2		Successful without heaters at 203° F
7/8	3	78.0	86.7		Failed without heaters at 194.3° F Successful with heaters
7/8	3	78.1	88.8		Successful without heaters at 196.8° F

TABLE III-7. (Concluded)

Test No. (Date)	Hot Water Flow Rate (gpm)	Cooling Tower Water Supply Temperature (° F)	Trailer Inside Air Temperature (° F)	Hot Water Temperature[a] at Percolation Loss (° F)	Restarts
7/9	1.7	76.9	85.0	199.2	
7/9	1.7	80.1	85.2	202.6	
7/9	1.7	80.6	87.2		Successful without heaters at 204.5° F

a. Hot water supply temperature at which evaporator temperature started to rapidly rise.

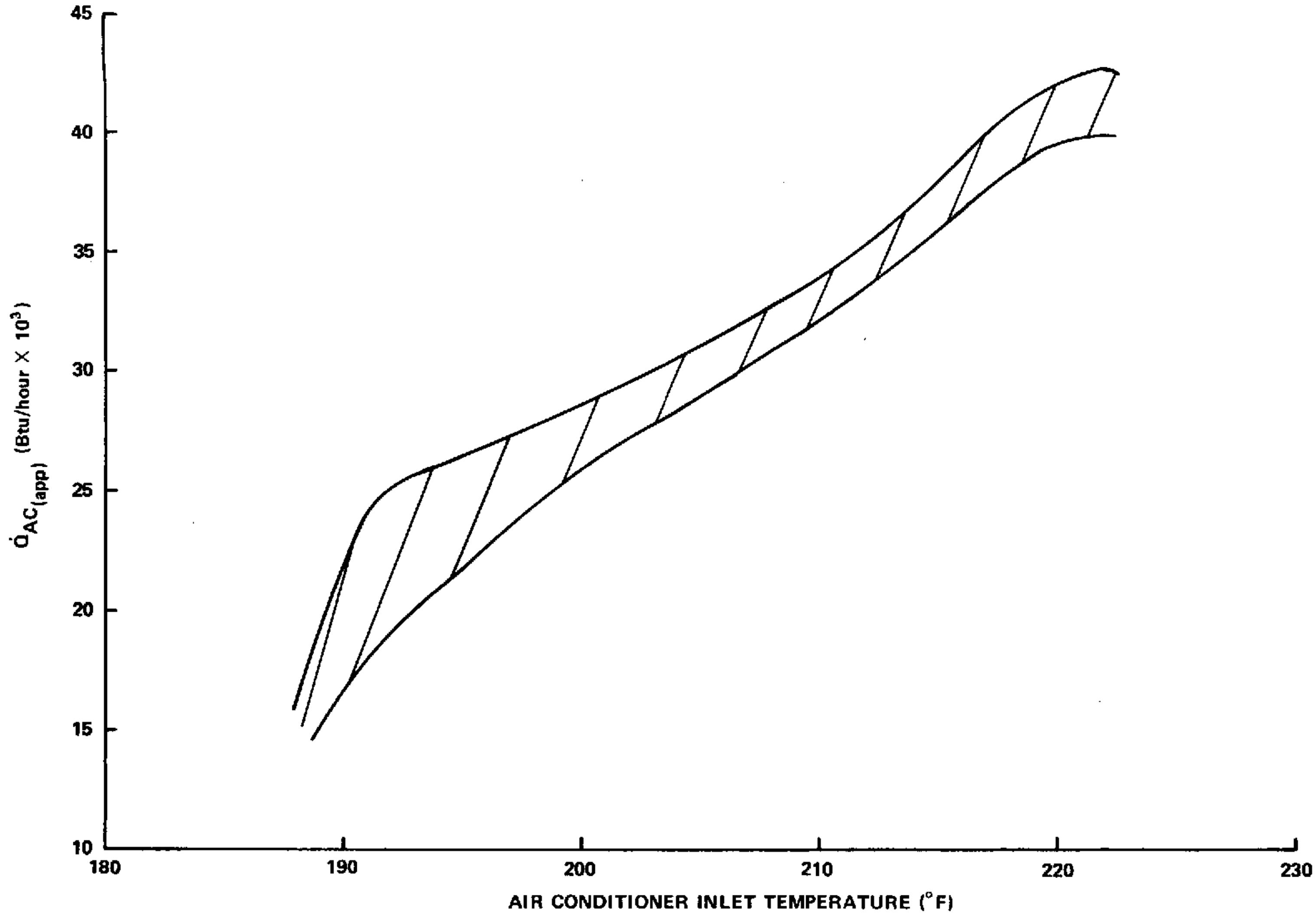


Figure III-24. $\dot{Q}_{AC_{(app)}}$ versus air conditioner inlet temperature (5.0 gpm).

90

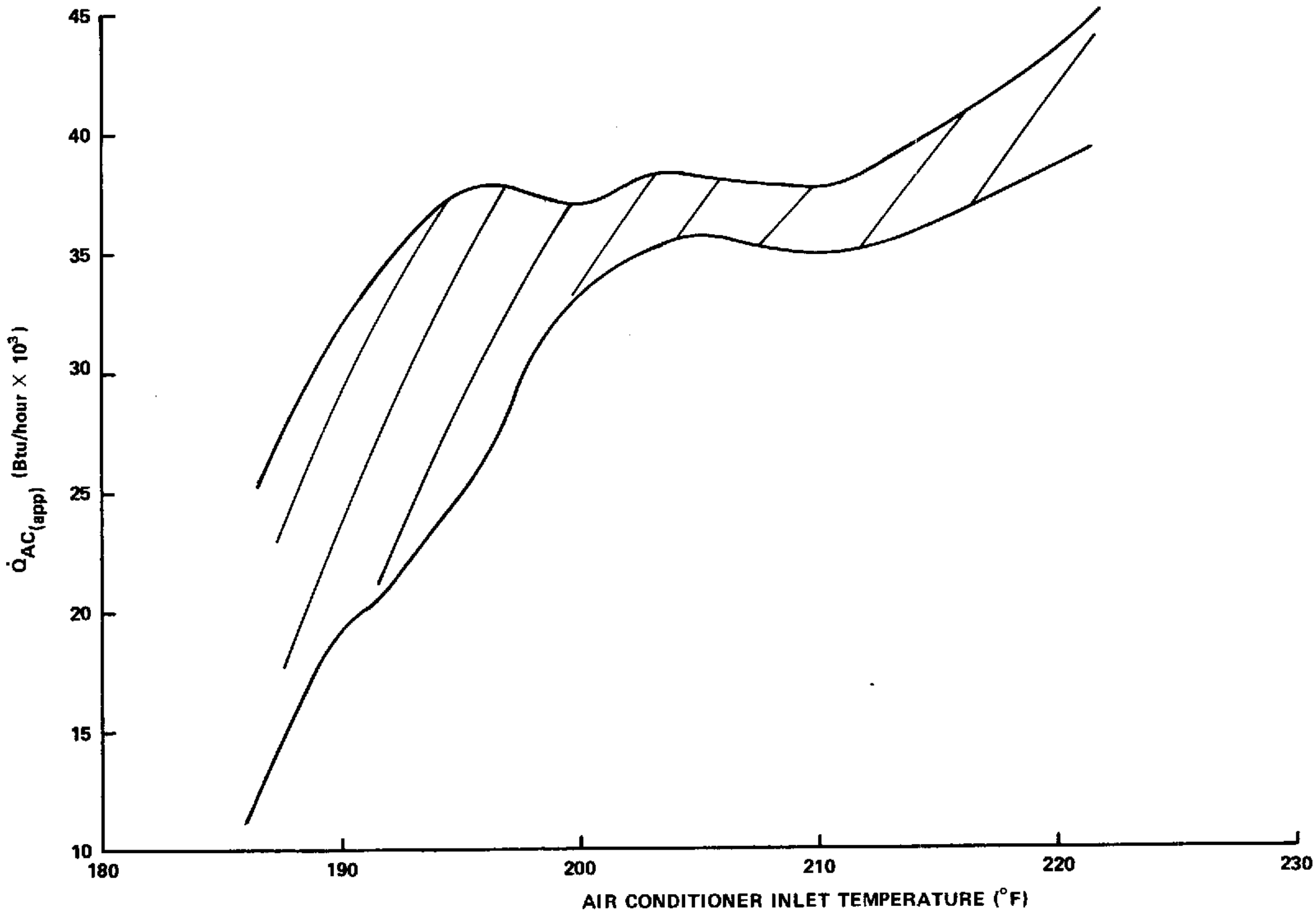


Figure III-25. $\dot{Q}_{AC_{(app)}}$ versus air conditioner inlet temperature (7.0 gpm).

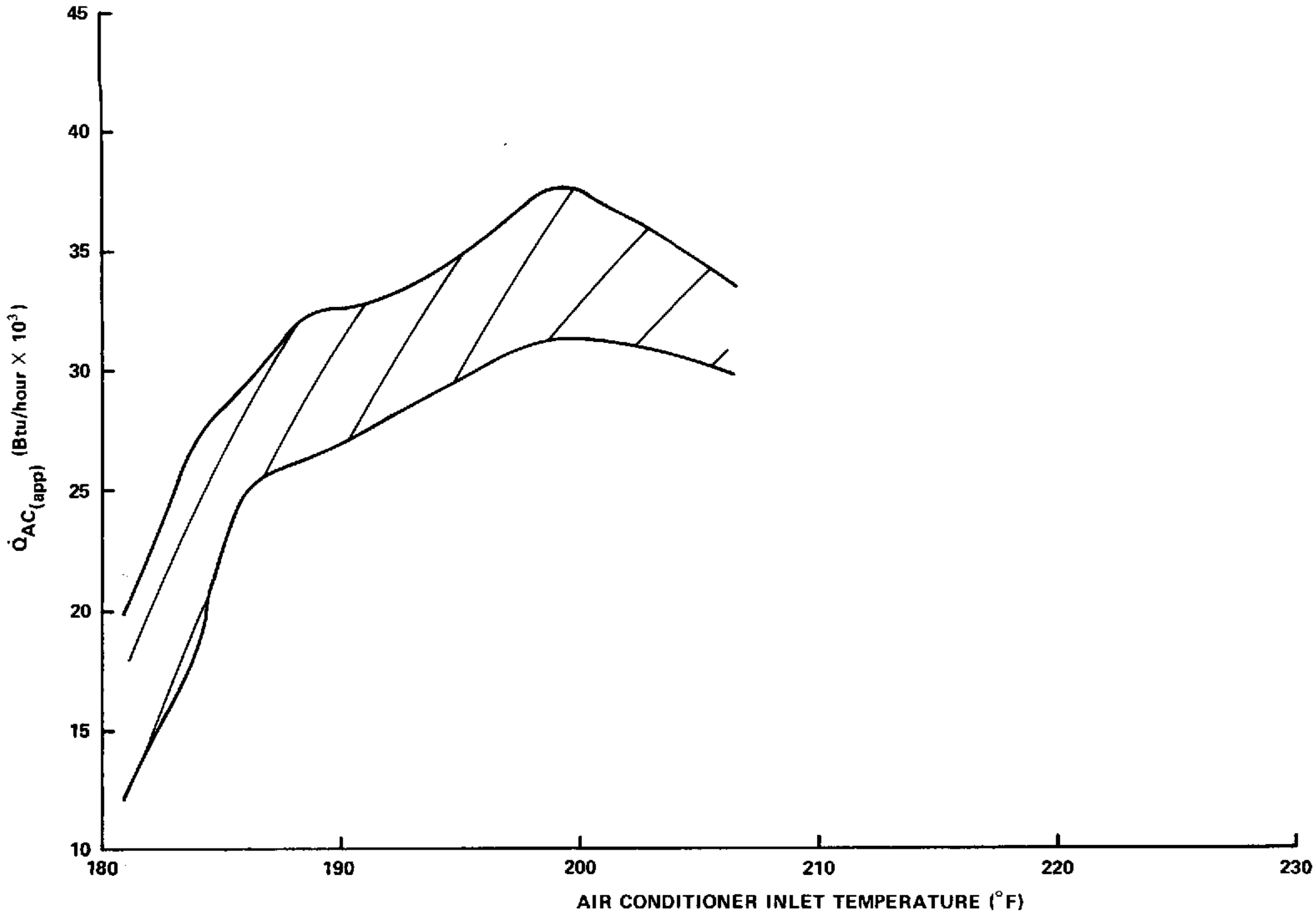


Figure III-26. $\dot{Q}_{AC_{(app)}}$ versus air conditioner inlet temperature (10.0 gpm).

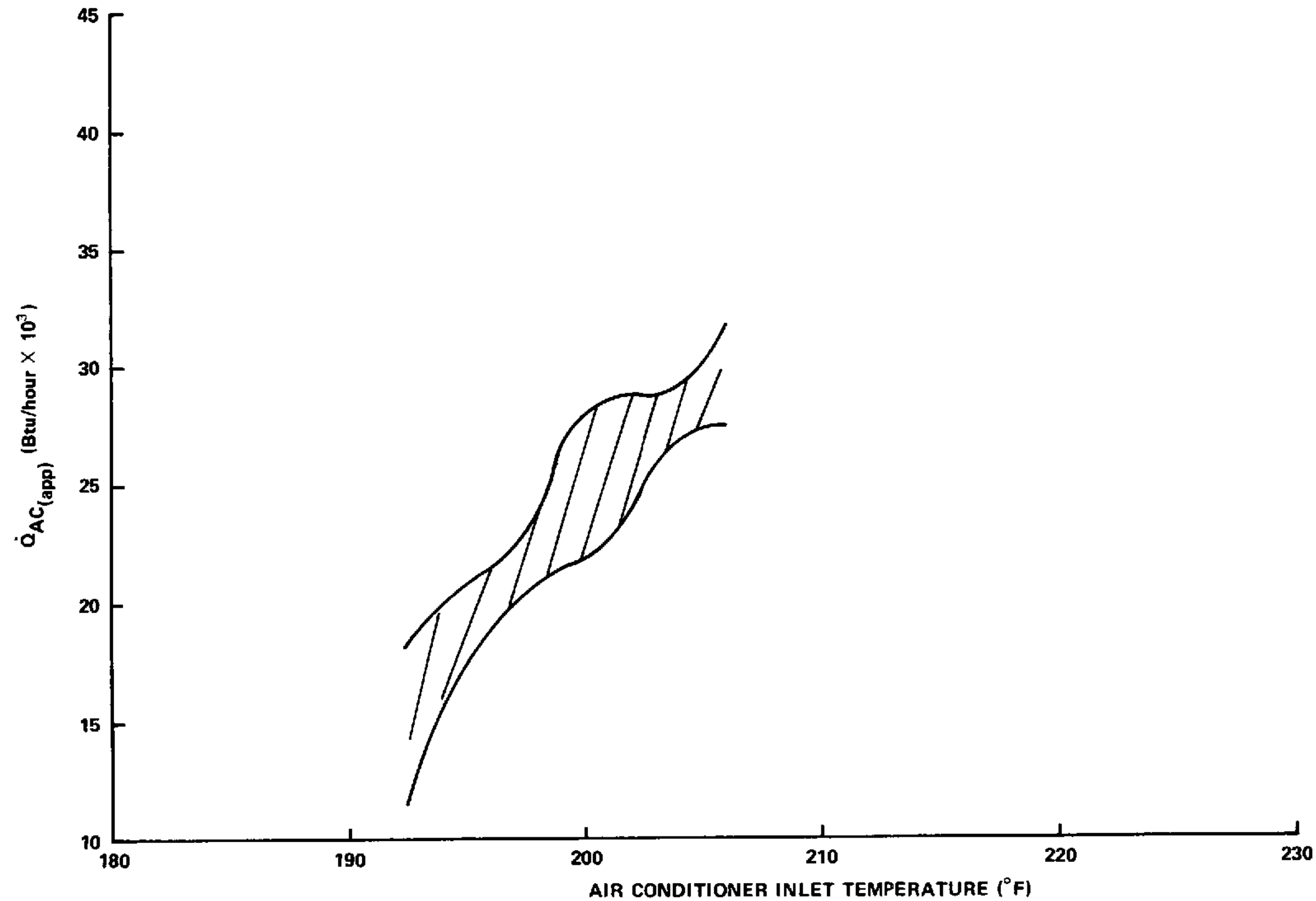


Figure III-27. $\dot{Q}_{AC_{(app)}}$ versus air conditioner inlet temperature (3.0 gpm).

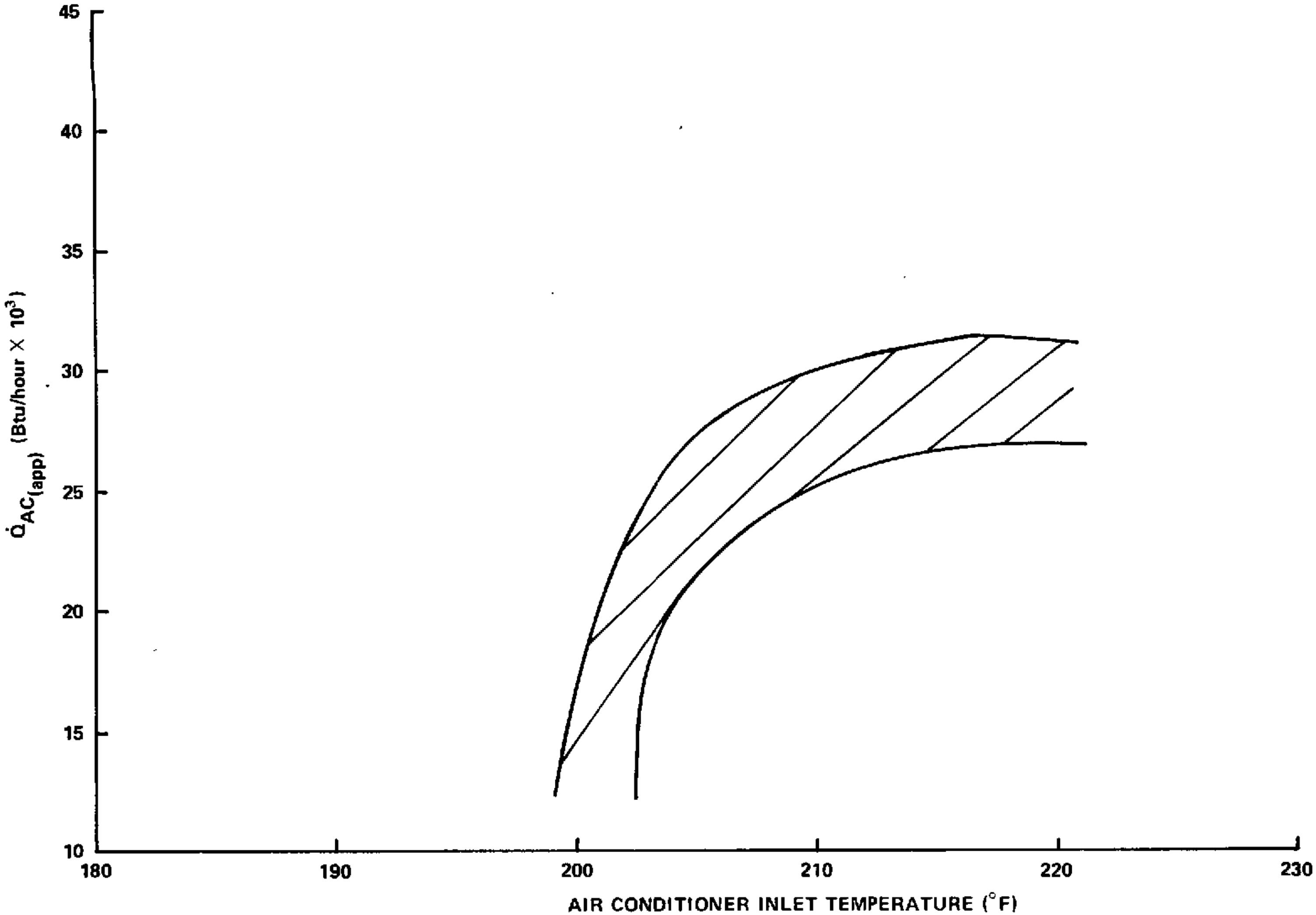


Figure III-28. $\dot{Q}_{AC_{(app)}}$ versus air conditioner inlet temperature (1.7 gpm).

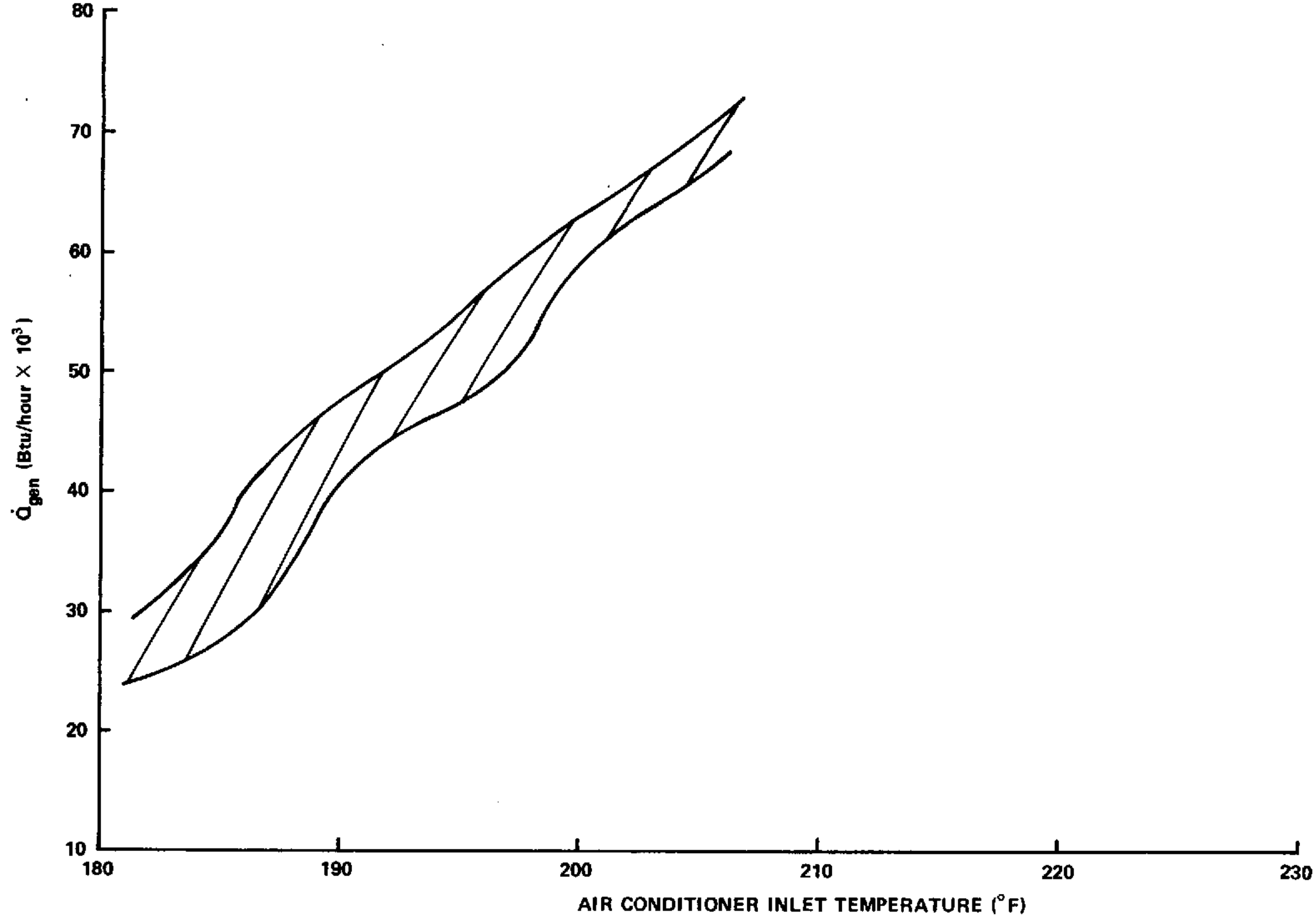


Figure III-29. $\dot{Q}_{gen}$ versus air conditioner inlet temperature (10.0 gpm).

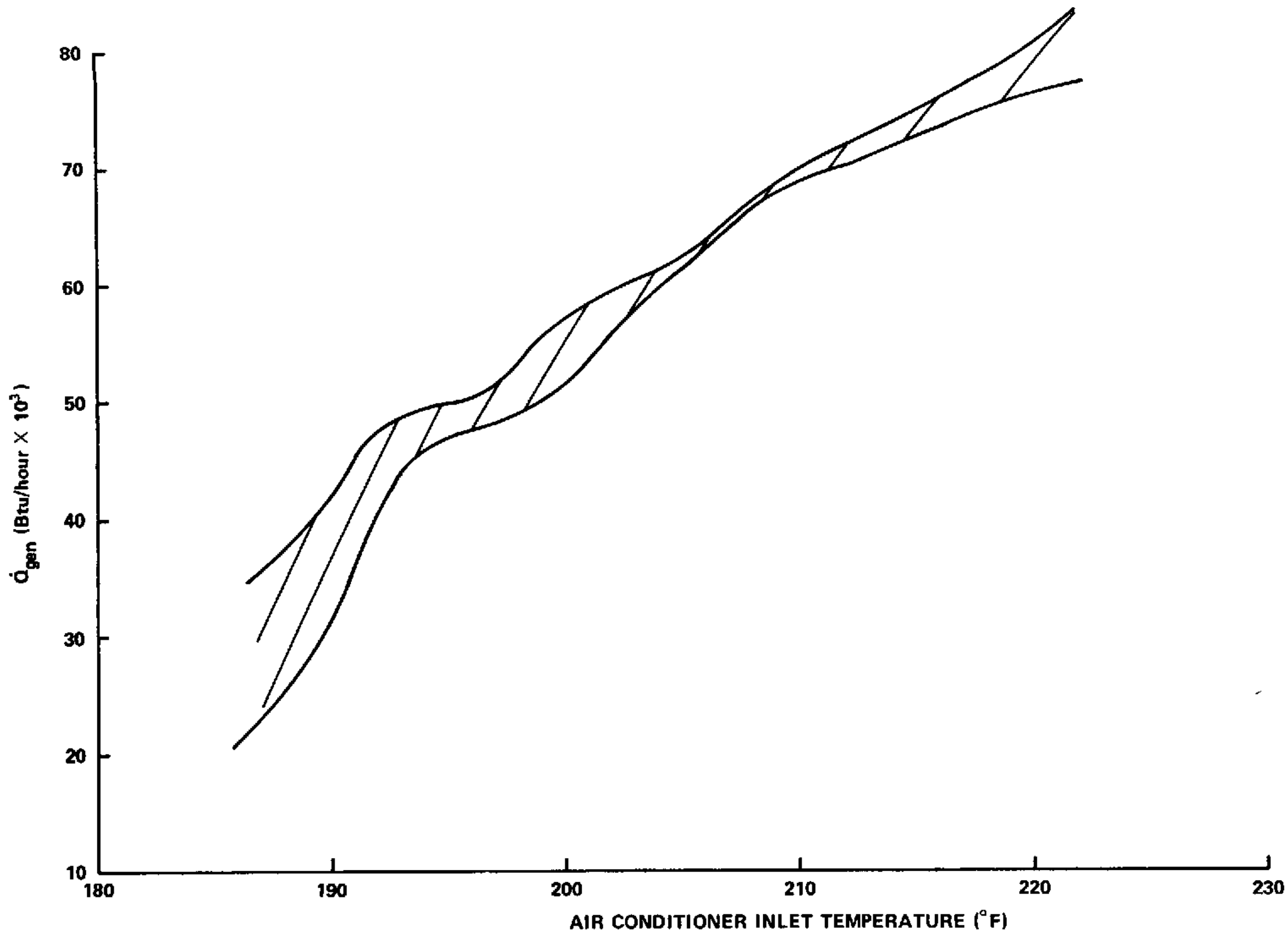


Figure III-30. $\dot{Q}_{gen}$ versus air conditioner inlet temperature (7.0 gpm).

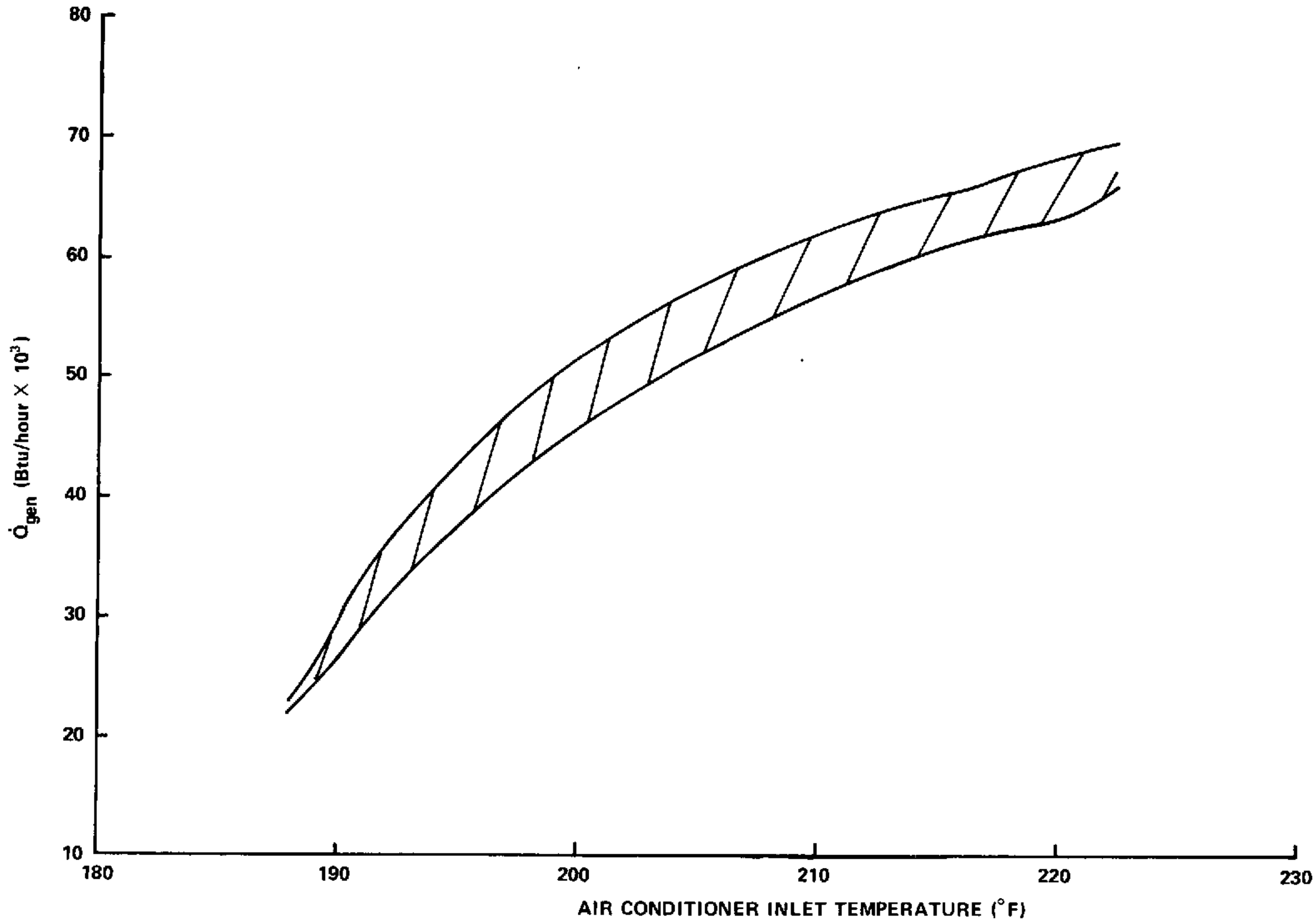


Figure III-31. $\dot{Q}_{gen}$ versus air conditioner inlet temperature (5.0 gpm).

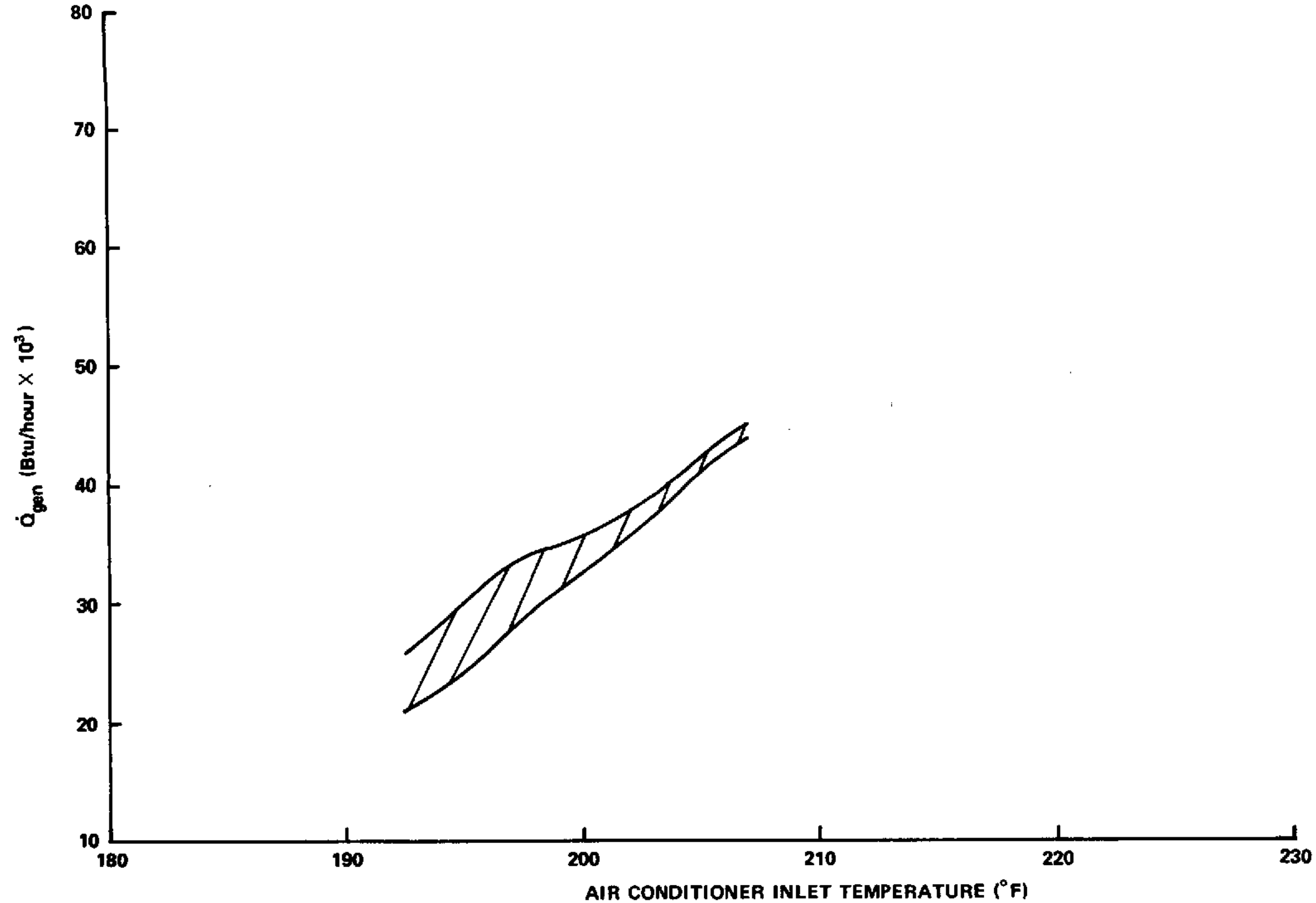


Figure III-32. $\dot{Q}_{gen}$ versus air conditioner inlet temperature (3.0 gpm).

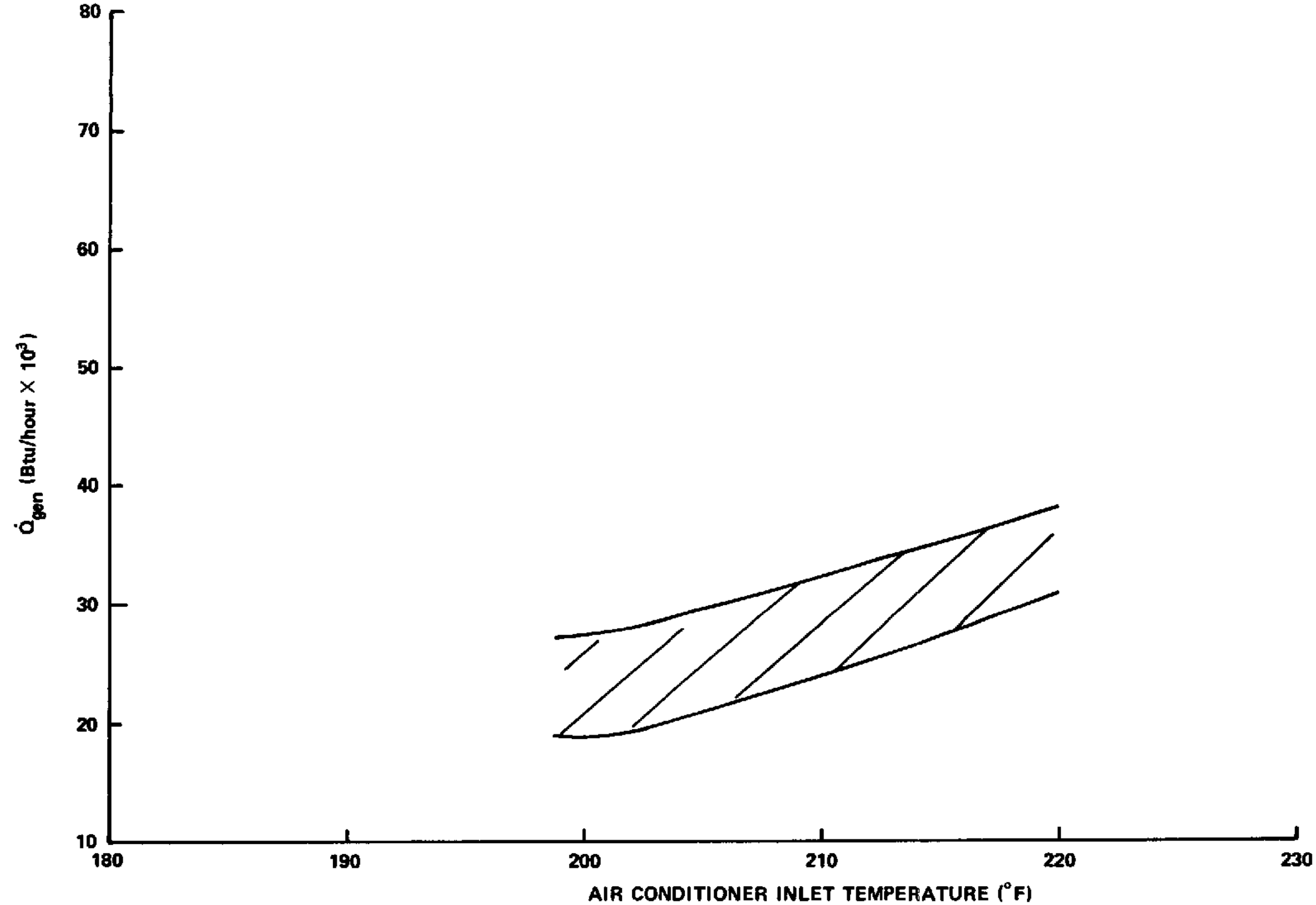


Figure III-33. $\dot{Q}_{gen}$ versus air conditioner inlet temperature (1.7 gpm).

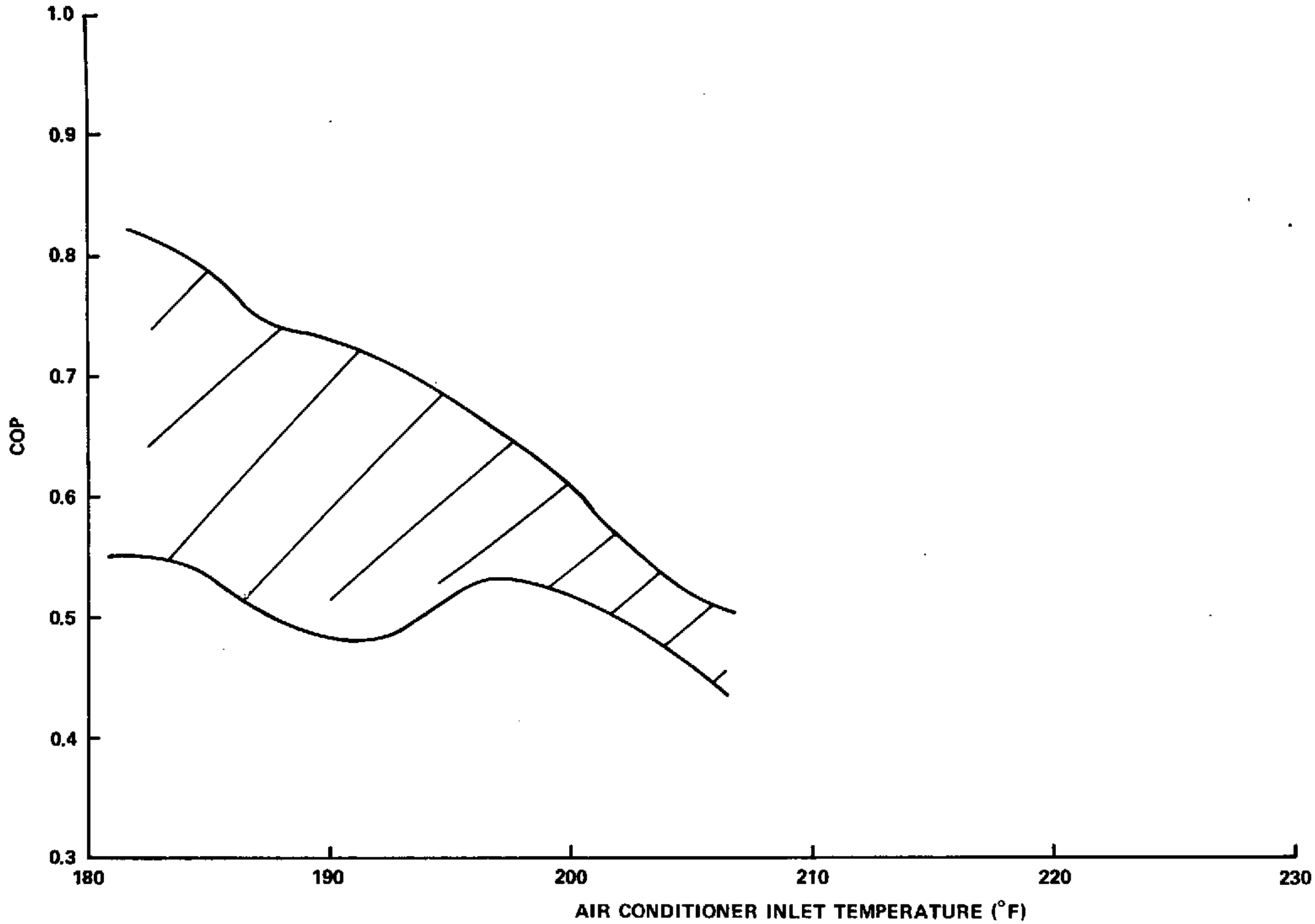


Figure III-34. COP versus air conditioner inlet temperature (10.0 gpm).

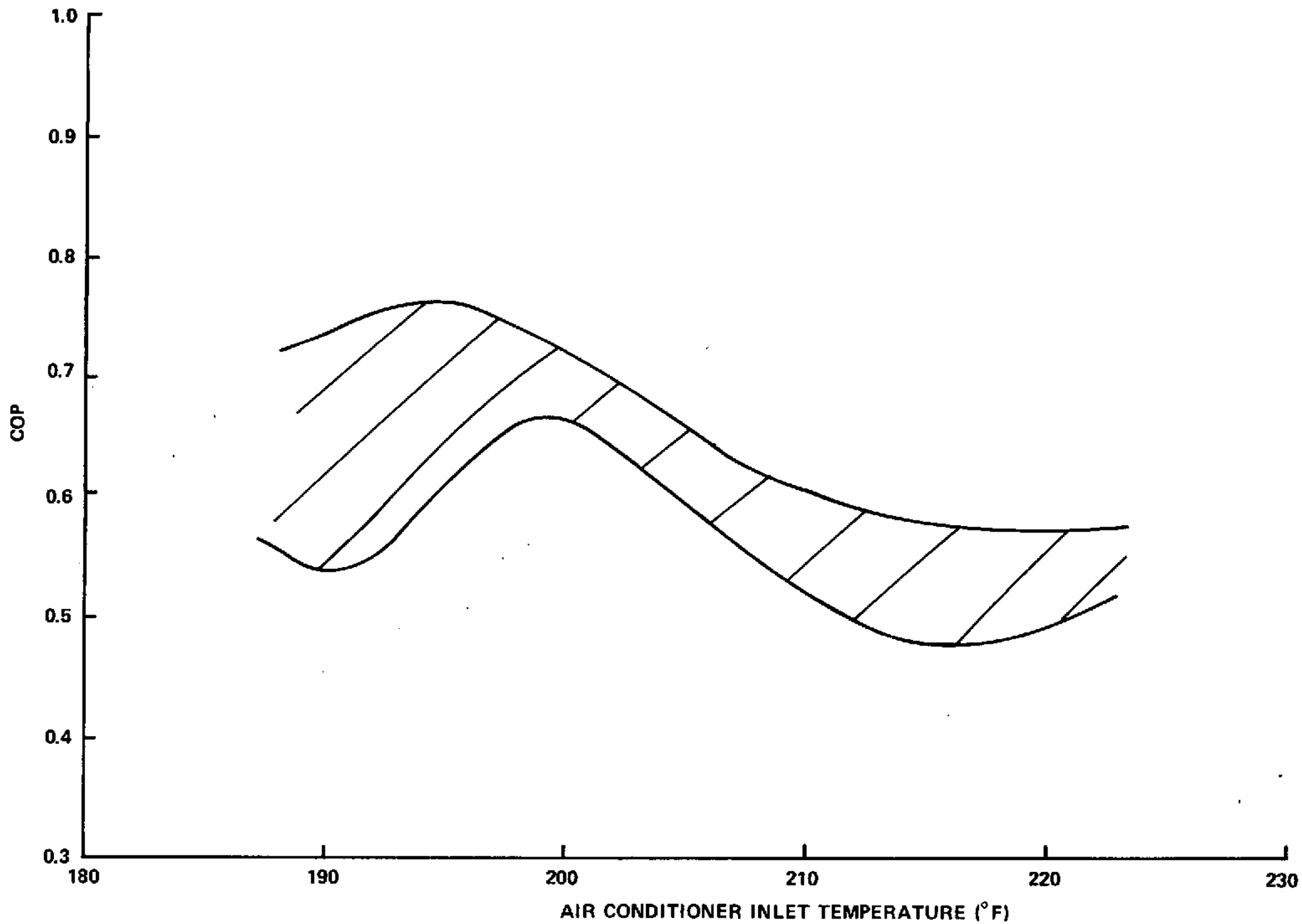


Figure III-35. COP versus air conditioner inlet temperature (7.0 gpm).

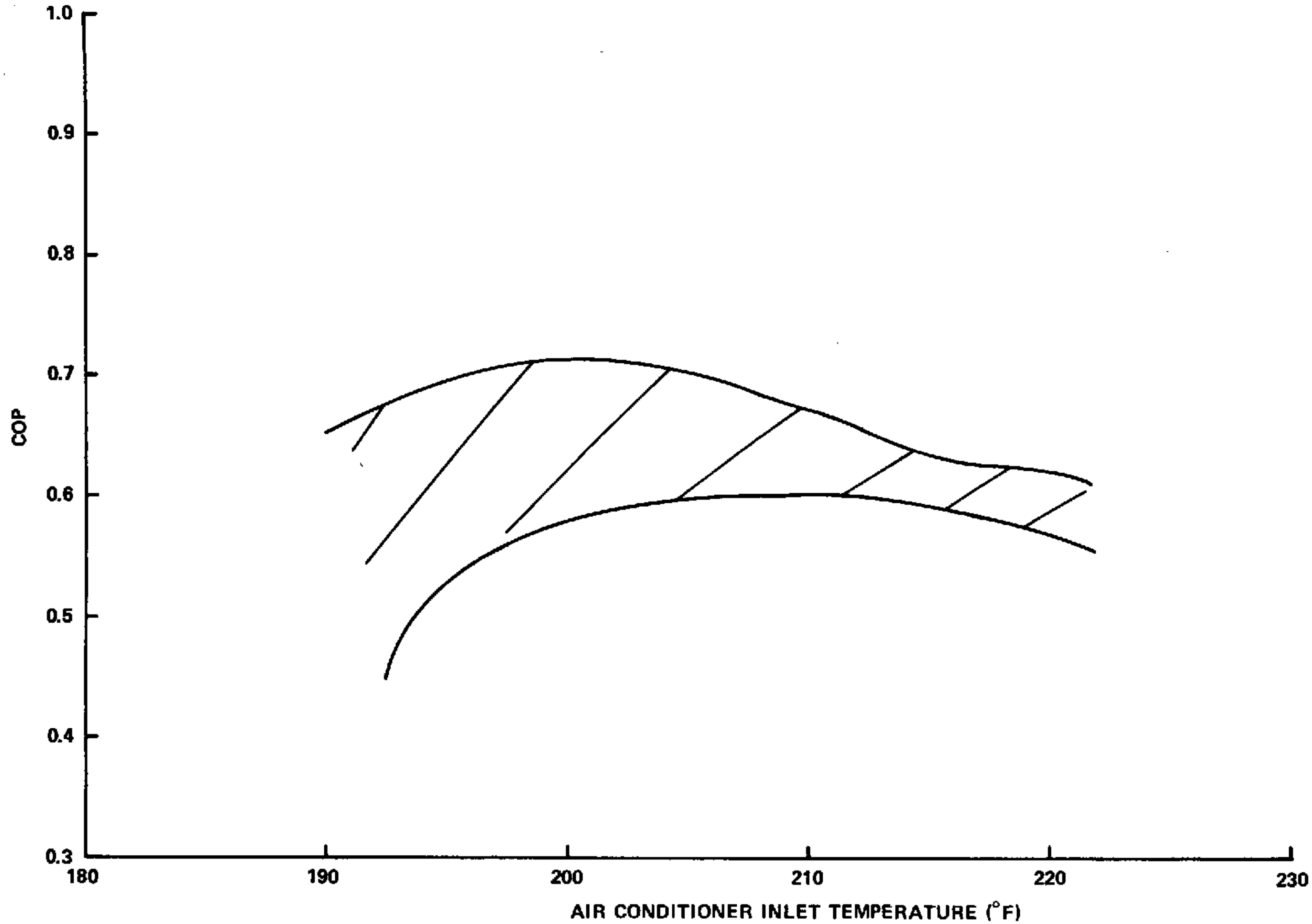


Figure III-36. COP versus air conditioner inlet temperature (5.0 gpm).

lack of control of the cooling tower temperature, the house air temperature, and relative humidity influences of these parameters are not discussed in detail.

During these tests a number of restarts were attempted to verify the capability of reestablishing percolation. These data are also given in Table III-7. The cooling tower temperature data indicated that as the cooling tower temperature was elevated, the generator energy input decreased and the minimum percolation temperature rose.

These data also typically verified the analytically justifiable trend that indicates the peak COP occurs just prior to loss of percolation. These values indicate COP values in excess of 0.8. However these high values are difficult to obtain in continuous steady-state operation because they occur precariously near the percolation loss temperature. The percolation loss temperature is affected by the cooling tower water temperature. These data also indicate that even though the COP and $\dot{Q}_{gen}$ values decrease with decreasing temperature and flow rate, the cooling capability of the unit, $\dot{Q}_{AC_{(app)}}$, also decreases with temperature. However, since $\dot{Q}_{gen}$ decreases at a more rapid rate, the overall effect is to increase the air conditioner performance (i.e., increase COP).

L. New Logic System

From the data generated during the special air conditioner tests, a new logic system was designed. This system was designed to allow a substantial margin of safety to avoid air conditioner damage while improving the ratio of energy absorbed by the collector subsystem to energy required to drive the air conditioner. Figure III-37 is a schematic representation of the new logic system. Using special air conditioner test data along with the new logic system diagram, a new system performance map was determined as shown in Figure III-38. This map depicts expected values of Q_{gen}, Q_{AC}, and COP as they vary with hot water inlet temperature.

The new logic system incorporates three separate control loops. The basis of the new system is termed the "minimum energy" loop. This uses the two existing air conditioner bypass valves, MV-4 and MV-5 (Fig. II-9), to bypass flow around the air conditioner at high temperatures. Bypassing suppresses heat transfer in the generator, thereby conserving energy. Below 193° F, the full flow rate capability is used to assure the maintenance of percolation in the air conditioner. Below 188° F, the calrod heaters are activated

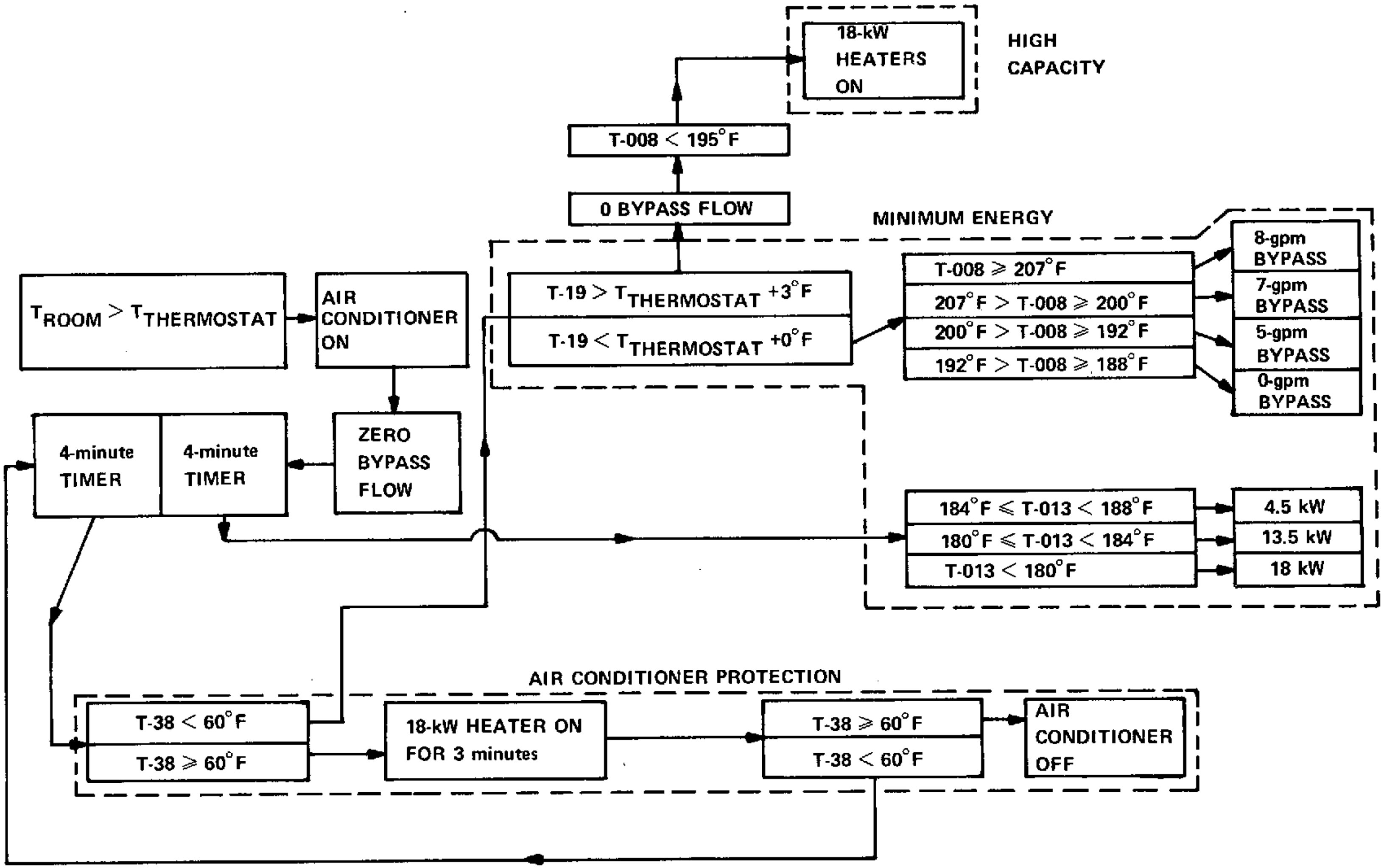


Figure III-37. New air conditioner logic system.

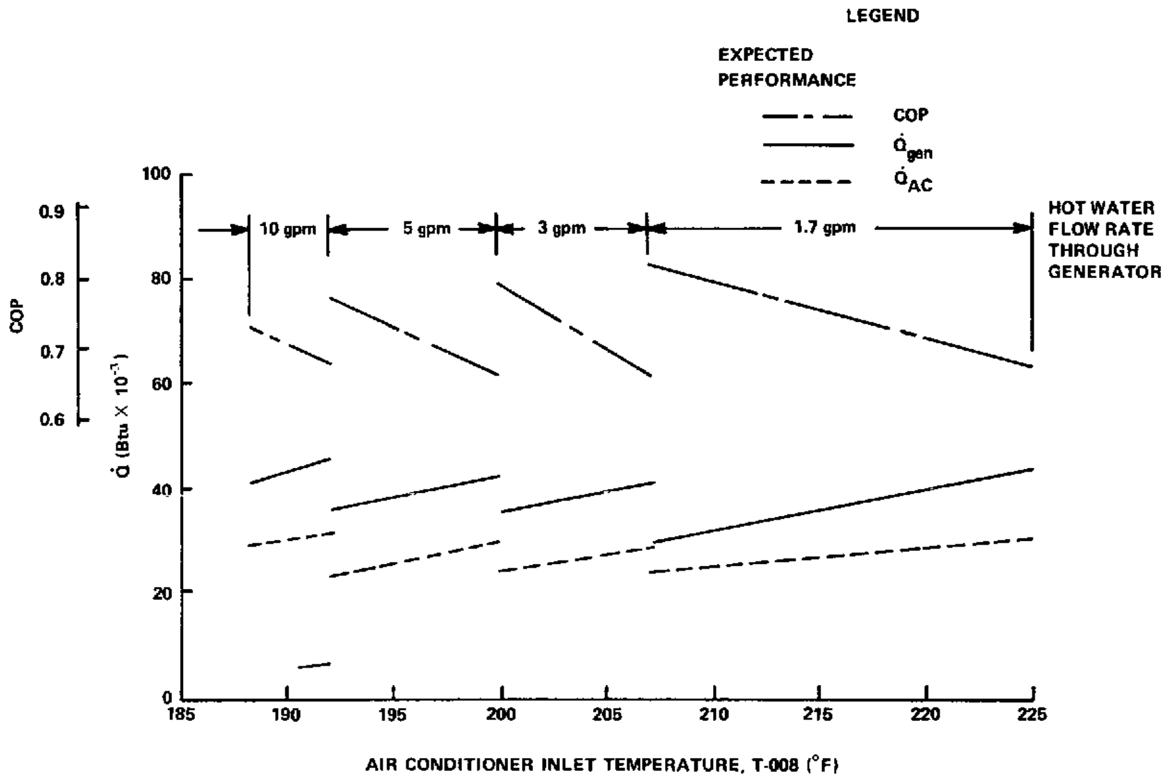


Figure III-38. New logic system performance.

in steps to maintain a minimum generator inlet temperature of 188° F. From 193° F to 203° F, the flow rate through the air conditioner generator is reduced to 5 gpm (i.e., MV-4 open and MV-5 closed). From 203° F to 207° F, the flow rate through the generator is 3 gpm (i.e., MV-5 open and MV-4 closed). Above 207° F, both MV-4 and MV-5 are open, lowering the generator flow rate to the lowest value used of 2.3 gpm. (Note: Special air conditioner tests were run at 1.7 gpm since the exact value of the flow rate with both valves open was unknown.) This scheme produced performance values as given in Figures III-39, III-40, and III-41.

The air conditioner protection loop provides protection from conditions in which sustained operation in the "minimum energy" loop is not possible or marginal. The air conditioner evaporator case temperature, T-038, being the most direct indication of proper air conditioner operation, is used to detect improper air conditioner performance. This temperature was redlined so that

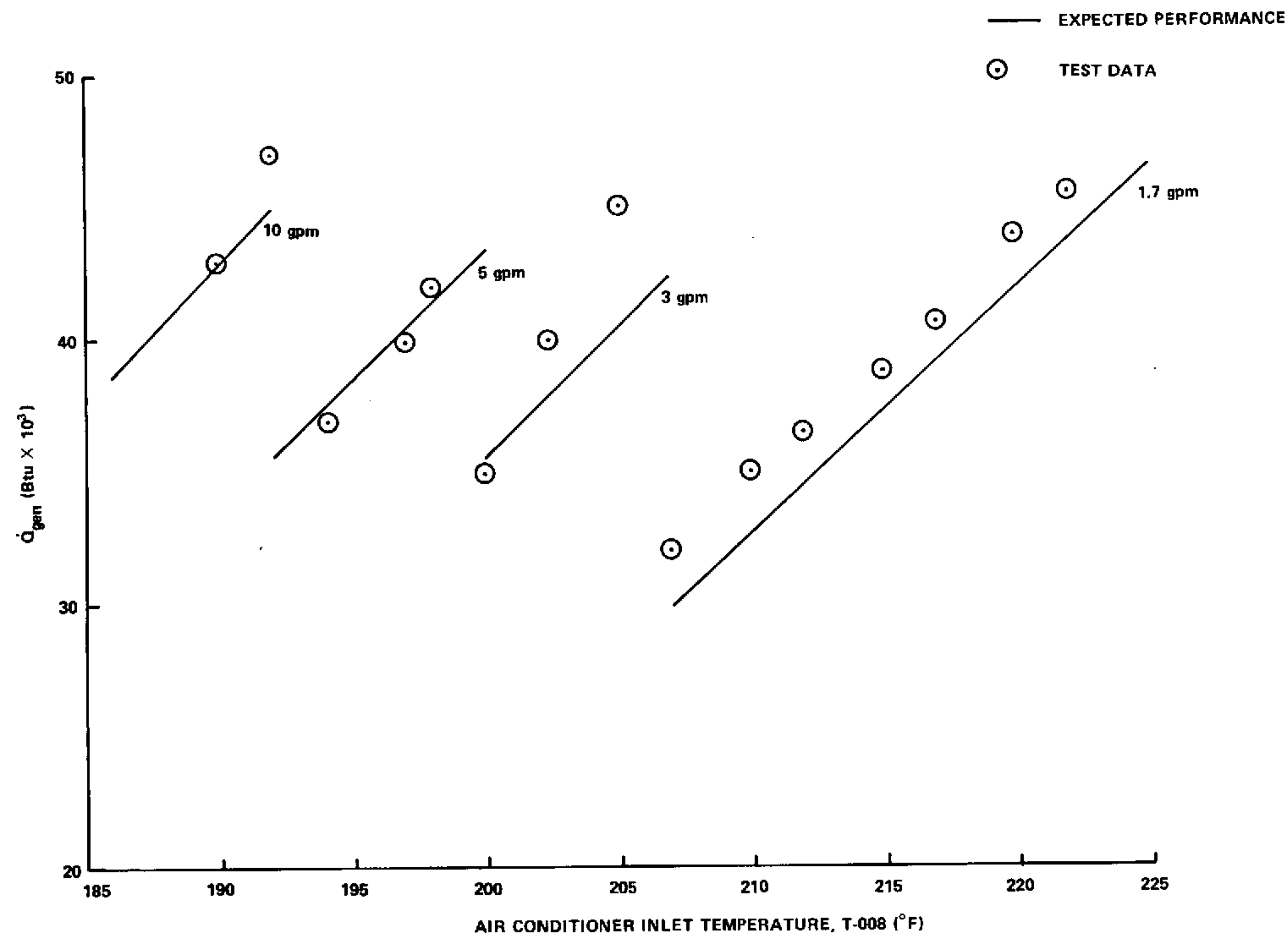


Figure III-39. New logic system $\dot{Q}_{gen}$.

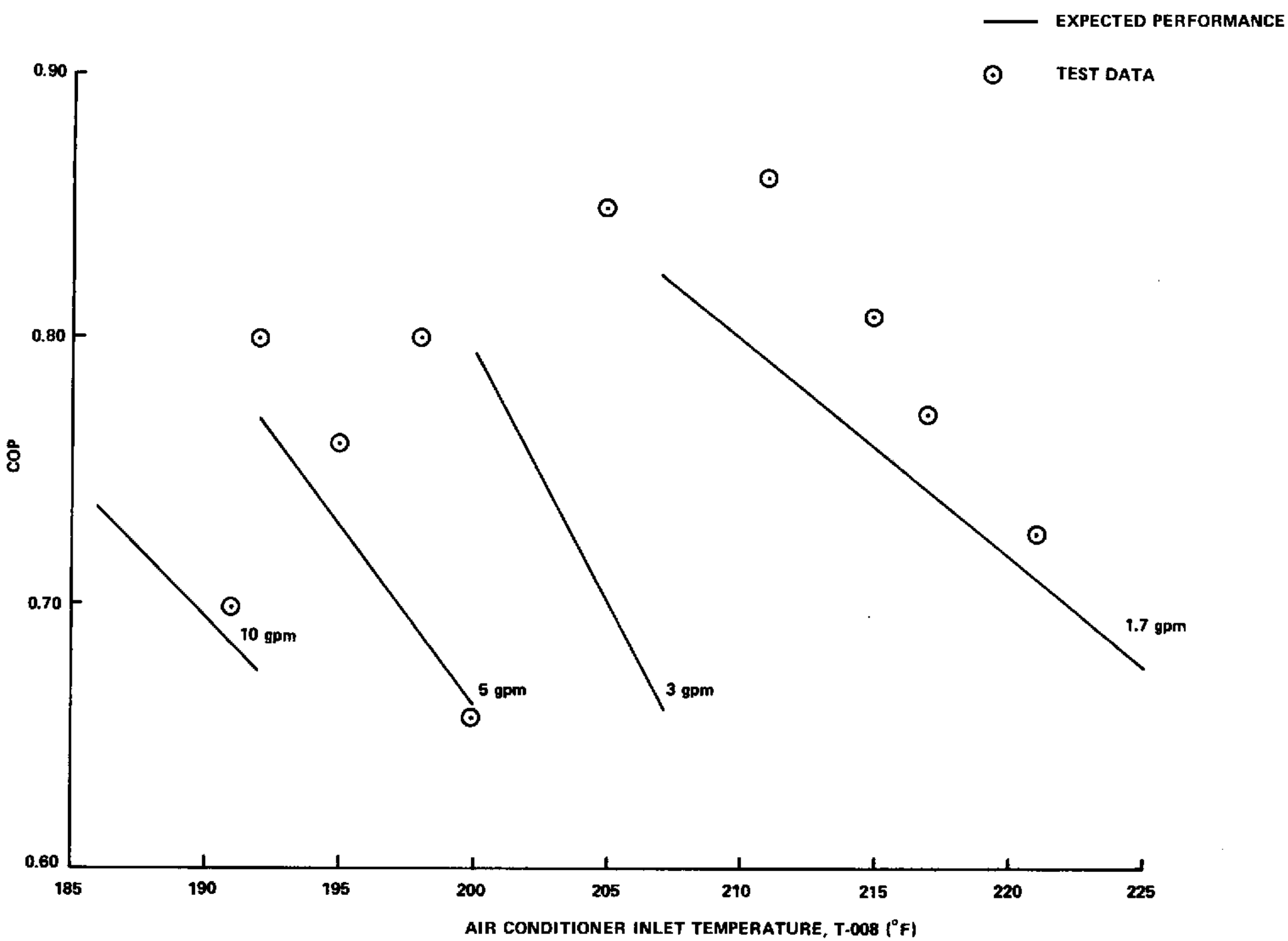


Figure III-40. New logic system COP.

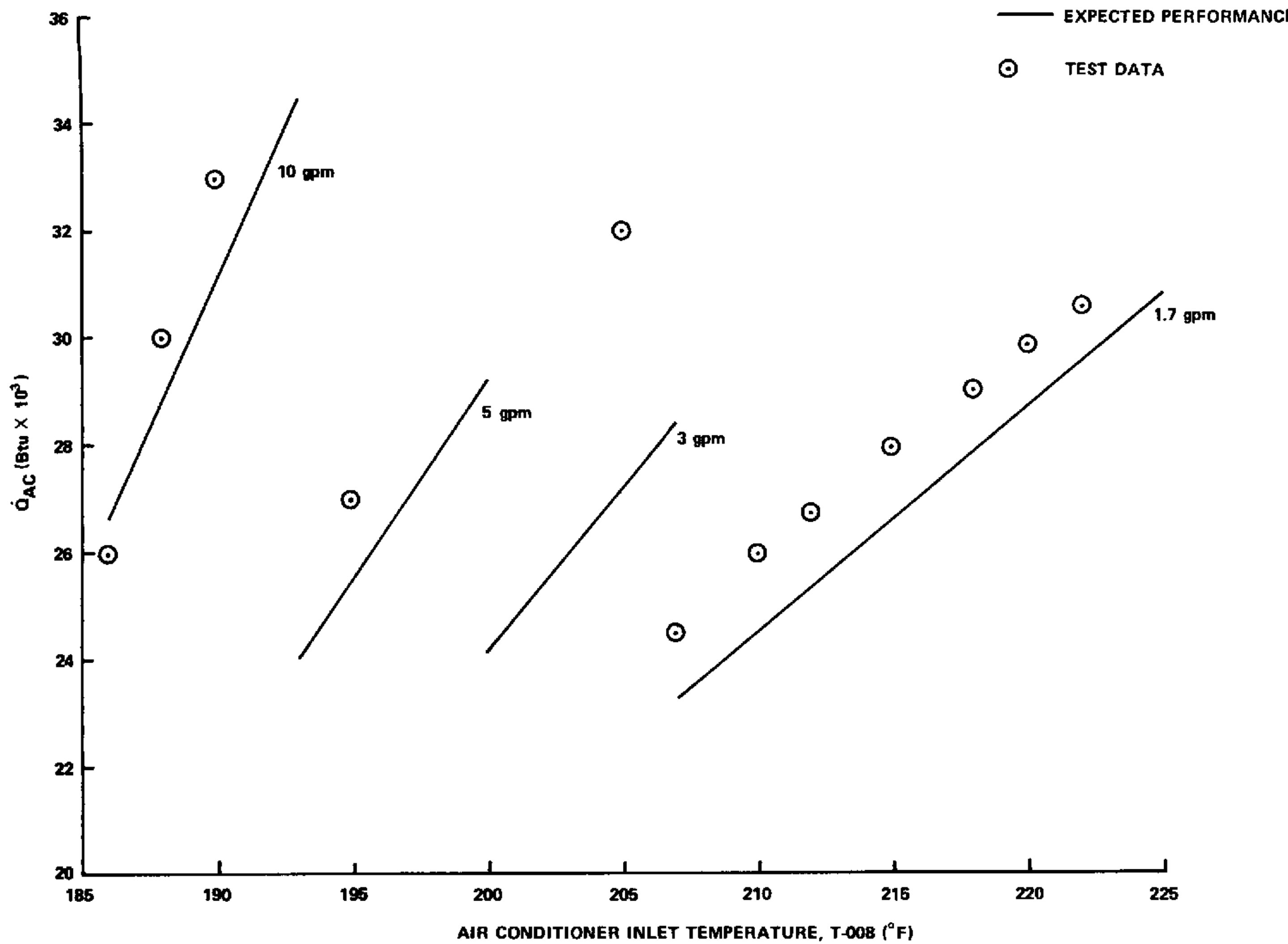


Figure III-41. New logic system $\dot{Q}_{AC}$.

at values above 60° F full auxiliary heater power is activated to allow the air conditioner the best chance to start. Should the air conditioner fail to run with full heater power, an automatic shutdown is activated. This shutdown opens all air conditioner circuit breakers until manual reset is accomplished. Should the air conditioner successfully start, then the normal "minimum energy" loop is entered. Any time the 60° F evaporator maximum temperature is violated, the "protection" loop is again engaged.

Should the room temperature at any time rise more than 3° F above the thermostat set point, the "high capacity" loop activates. This loop forces full hot water flow through the generator; if T-008 is above 195° F, this is the sole operation performed. Should T-008 be below 195° F, full heater power is applied. This loop is designed to assure that the maximum cooling capability is being obtained from the unit when the inside temperature is excessively high. When the house temperature drops back to the thermostat setting, this loop is deenergized and the normal "minimum energy" loop is reacquired.

M. Thermal Energy Balance

An examination of the test system shows that an overall thermal energy balance or imbalance is one of the most meaningful indicators of how well the system has performed. In theory, the optimum system should collect sufficient energy to operate the air conditioner and overcome all thermal losses. In reality, a cooling system of this type has not been perfected. Under the heading of energy used falls $\dot{Q}_{gen}$, the thermal energy required to operate the air conditioner, and the system heat leak, Q_{loss}. The energy supplied is that which is collected. This may either be supplied directly to the air conditioner from the collector during the day of collection, Q_{col}, or extracted from the energy storage tank, $Q_{egy\ tk}$, which stores energy previously collected. (Note: A negative sign is applied to $Q_{egy\ tk}$ when it supplies energy to the air conditioner since this results in a tank temperature drop.) When insufficient energy is available at temperatures required to drive the air conditioner, the auxiliary heaters, $Q_{aux\ htrs}$, must supply this shortfall. Considering all these energy sources and sinks, a total system energy balance may be written to determine the energy loss, Q_{loss},

$$Q_{loss} = Q_{col} - Q_{egy\ tk} + Q_{aux\ htrs} - Q_{gen} \quad .$$

Table III-8 lists these values, excluding Q_{loss}, along with high and low daily ambient temperatures and thermostat settings for a number of representative days before control system modification. A number of days are excluded because of a variety of anomalies that invalidated results for these days. Table III-9 lists similar values for representative days after the modification through August.

From these tables, it can be determined that before the modification the average energy collected was 194 000 Btu/day, including an average of 1 day per week lost to inclement weather. The average value of Q_{gen} is 479 846 Btu/day. Using these values along with the average calculated heat loss per day of 136 315 Btu/day results in an average daily energy shortfall of 422 161 Btu/day. Similarly, the average Q_{col}, Q_{gen}, and Q_{loss} for post-modification days are 395 226 Btu/day, 532 464 Btu/day, and 202 906 Btu/day. These data indicate an average daily shortfall of 340 144 Btu/day, an obvious improvement in the total energy balance. This improvement is even more dramatic when it is considered that the dates in August for which postmodification test data were accumulated were warmer than the premodification June dates; consequently, there were longer daily air conditioner run times. It also should be noted that higher thermostat settings induce lower cooling requirements. As can be noted by comparing Tables III-8 and III-9, the premodification thermostat settings were significantly higher on the average than the post-modification settings. A more meaningful comparison of energy used by the air conditioner is given by $\dot{Q}_{gen}$. The premodification average value of $\dot{Q}_{gen}$ was 55 926 Btu/hour versus 34 621 Btu/hour for the postmodification days. However, considering only this parameter is misleading since the cooling rate provided by the modified system is, in general, less than that of the older system. Air conditioner performance is best assessed using average values of COP determined from daily values. Before the major logic system modification these values averaged 0.59 and after modification they averaged 0.70. These comparisons indicate significant improvement; however, the difference in these two values would have been even greater had the hot water temperature at which the system operated been higher (the case when a more favorable energy balance is achieved). Also the postmodification COP includes all test data. For a number of these days the heater control was inadvertently set higher than the logic required causing unnecessarily low COP values, thereby lowering the total average. All of this serves to show the advantage of operating at minimum temperatures and flow rates into the air conditioner.

TABLE III-8. DAILY ENERGY SUMMARY — BEFORE CONTROL SYSTEM MODIFICATION

Date	Air Conditioner Run Time (hours)	Q_{col} (Btu)	$Q_{egy\ tk}$ (Btu)[a]	Q_{gen} (Btu)	$Q_{aux\ htrs}$ (Btu)	Ambient Temperature T_{high} (°F)	T_{low} (°F)	Thermostat Setting (°F)
6-3	7.22	260 200	-334 080	529 644	0	82	57	78
6-4	9.37	143 710	-492 480	698 470	202 673	81	64	78
6-6	6.18	69 421	- 90 890	353 932	320 694	82	64	79
6-7	8.89	149 616	- 29 800	504 123	468 914	83	63	79
6-8	7.43	0	-135 590	384 949	386 374	79	68	79
6-9	8.61	268 797	125 166	473 620	456 846	84	68	78
6-10	5.35	397 532	137 080	291 084	169 490	81	61	78
6-11	10.35	388 800	- 52 150	541 235	256 657	81	56	78/76
6-14	8.68	246 350	25 560	511 600	353 200	85	59	76/78
6-15	8.35	66 300	- 89 460	495 500	457 000	85	63	78
6-17	4.37	333 665	- 29 820	256 750	763	79	50	78
6-18	5.67	0	-266 960	396 390	316 432	77	57	78/76
6-19	11.58	352 875	- 28 400	607 677	381 260	87	59	76
6-20	15.0	225 415	- 14 200	558 715	482 335	91	64	76
6-22	11.63	3 412	- 73 840	593 995	691 584	88	67	76

a. Negative values indicate energy loss from tank.

TABLE III-9. DAILY ENERGY SUMMARY — AFTER CONTROL SYSTEM MODIFICATION

Date	Air Conditioner Run Time (hours)	Q_{col} (Btu)	$Q_{egy\ tk}$ (Btu)	Q_{gen} (Btu)	$Q_{aux\ htrs}$ (Btu)	Ambient Temperature		Thermostat Setting (°F)
						T_{high} (°F)	T_{low} (°F)	
7-29	5.42	268 934	-727 040		5 000	91	74	76
7-30	5.97	565 710	-298 200	548 848	0	87	60	76
7-31	6.25	648 487	-213 000		328 846	87	60	76
8-1	6.94	604 066	48 280	488 153	88 667	90	61	76
8-2	1.94	114 680	- 90 880	510 609	343 572	84.5	69	76
8-3	3.13	172 715	- 28 400	528 788	468 327	82.5	69	76
8-4	4.9	395 314	170 400	432 181	334 007	84	55	76
8-5	6.25	612 712	416 060	483 200	465 301	86	59	76/72
8-6	6.25	533 239	298 200	547 000	530 400	88	59	72
8-7	0	0	-416 060	657 737	194 779	74	68	72/76
8-8	0	0	-248 600	292 928	155 800	82	68	76
8-9	2.43	270 036	11 610	549 131	472 752	85	66	76
8-10	5.90	645 774	71 000	630 332	228 434	88	67	76
8-11	3.89	406 463				86	67	76

TABLE III-9. (Continued)

Date	Air Conditioner Run Time (hours)	Q_{col} (Btu)	$Q_{egy\ tk}$ (Btu)	Q_{gen} (Btu)	$Q_{aux\ htrs}$ (Btu)	Ambient Temperature		Thermostat Setting (°F)
						T_{high} (°F)	T_{low} (°F)	
8-12						89	68	76
8-13	6.39	507 160	- 9 940	558 854	221 456	87	66	76
8-14	0.07	4 010	- 90 800	562 180	391 471	82	67	74
8-15	5.21	363 319	68 160	514 754	439 880	87	66	76
8-16	4.86	676 538	79 520	567 877	306 196	86	67	76
8-17	18.5	596 213	- 4 260	557 631	157 419	88	67	76
8-18	14.8	216 295	- 99 400	446 006	325 026	87	68	76
8-19	9.72	707 982	61 340	506 199	158 123	89	64	76
8-20	15.3	748 527	61 920	565 646	194 458	90	65	76
8-21	15.97	467 000	- 96 480	607 000	206 620	86	67	76/72 at 3 p.m.
8-22	17.64	298 771	432 200	655 947	571 308	85	66	72
8-23	12.92	678 243	214 560	547 294	420 762	89	63	72/76 at 8 a.m.
8-24	15.97	409 795	- 84 960	545 358	419 455	90	64	76

TABLE III-9. (Concluded)

Date	Air Conditioner Run Time (hours)	Q_{col} (Btu)	$Q_{egy\ tk}$ (Btu)	Q_{gen} (Btu)	$Q_{aux\ htrs}$ (Btu)	Ambient Temperature T_{high} (°F)	T_{low} (°F)	Thermostat Setting (°F)
8-25	16.6	628 299	86 400	560 383	344 222	91	67	76
8-26	20	433 321	-207 360	579 590	132 043	92	72	76
8-27	18.9	445 923	- 24 480	564 811	243 009	93	71	76
8-28	19.2	310 060	- 33 120	612 625	496 019	88	71	76
8-29		226 000				88	68	76
8-30	13.33	6 661	-163 300	421 416	454 469	81	68	76
8-31	12.12	75 228	- 14 200	459 000	327 448	81	64	76

As shown in Tables III-8 and III-9, Q_{col} also increases when comparing early data averages with data generated later. Although lowering of the air conditioner operating temperature and, consequently, the collector inlet temperature does improve collection, this is not the principle cause of this increased collection. Other possible causes for this improvement are discussed in subsection III. C.

IV. CONCLUSIONS

The successful operation of the MSFC house has demonstrated the technical feasibility of using solar energy for cooling. Although hardware problems were experienced, indications are that solutions to all these problems lie within existing technological capabilities.

Two of the most significant hardware problems were related to corrosion in the AC/Htr fluid loop and the control system set-point drift. Neither of these problems is necessarily pertinent because types of hardware used in both these systems will probably be vastly different from hardware selected for residential use. Both the valving that corroded and the controller that drifted are "off-the-shelf" NASA test equipment designed for other uses and were used in numerous test setups during the past 8 to 10 years. As a result, notation of these failures is of value only as it applies to new hardware design.

The corrosion inhibitor, sodium chromate, has been labeled an unacceptable material for large scale use. This is a result of the environmental effect of dumping large quantities of chromate in streams and rivers. For this reason, studies are proceeding to find a substitute for this substance in the event an aluminum collector is used. Use of a copper or steel system may preclude the necessity for corrosion inhibition.

Thermal system performance was detailed in Section III. Daily thermal performance can be assessed by examining the percent of total air conditioner thermal energy supplied by the solar collector subsystem (Fig. IV-1). For the reader's convenience Table IV-1 summarizes the average values of the most pertinent parameters for initial and final system configurations. A comparison of these values serves to dramatize the improvements realized from the discussed modifications.

Table IV-1 gives the current average energy split as 38 percent solar and 62 percent auxiliary power. These averages are somewhat misleading in that they represent actual measured values for the environment to which the MSFC house was subjected from late July to the first of September. In this period, for the test days considered, approximately 40 percent were significantly overcast (i.e., approximately 3 days per week overcast, little or no sun). This compares with only 25 percent of the days being overcast (approximately 1.5 days per week) from late May to late June. With more favorable climatic conditions the percent of solar energy improves. For instance, the

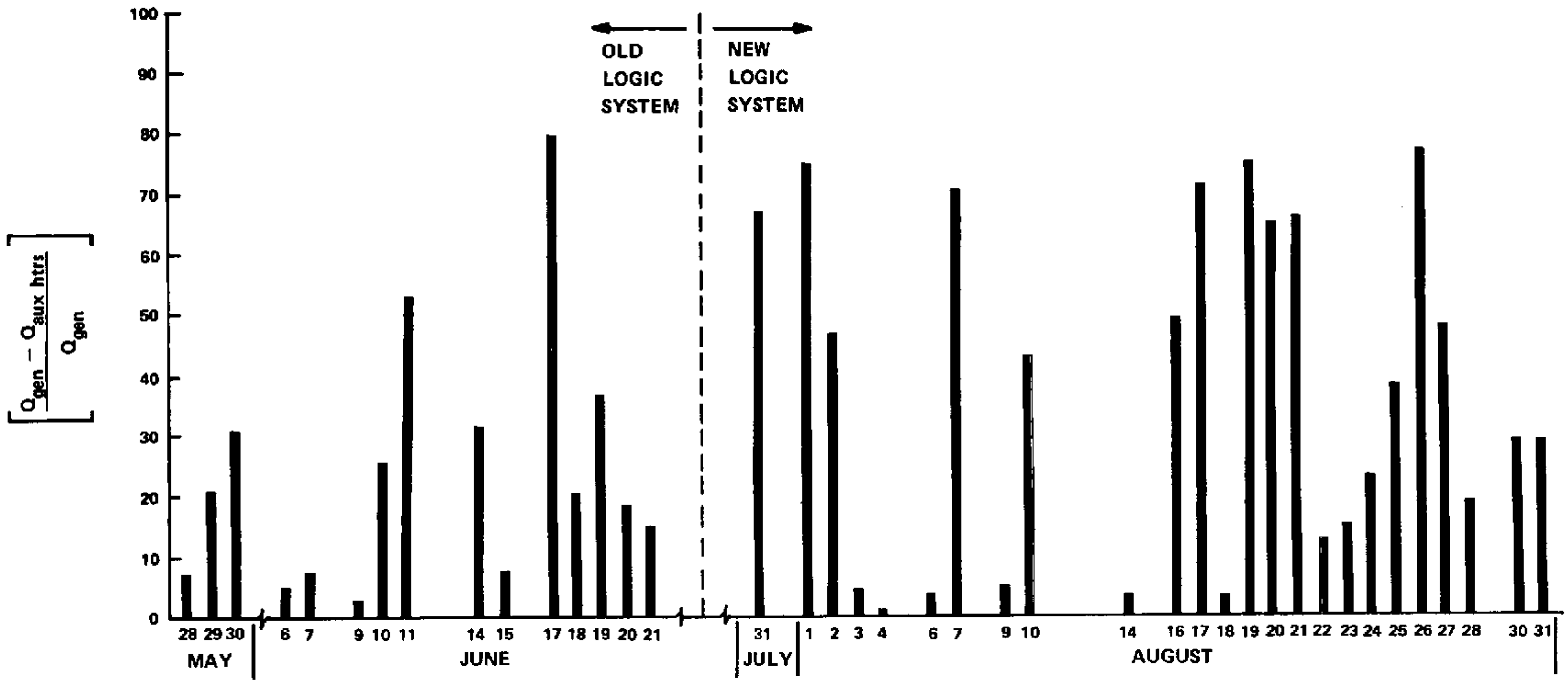


Figure IV-1. Percent of total air conditioner thermal energy supplied by the solar system.

TABLE IV-1. AVERAGE SYSTEM PERFORMANCE

	Air Conditioner Energy Consumption Rate (Btu/hour)	Air Conditioner Cooling Rate (Btu/hour)	Coefficient of Performance	Collection Rate (Btu/day)	Percent Solar
Initial System (May 28 to May 31)	71 453	30 010	0.42	231 953	15
New Logic System (July 29 to September 1)	34 621	22 956	0.70	371 192	38

average energy collected on a clear day (using data for August) was 554 424 Btu/day with an average Q_{gen} for these days of 568 600 Btu/day. Using this and assuming 6 clear days and 1 overcast day (zero collection), with an average daily loss of 180 000 Btu, the split would have been 56 percent. For 5 clear days in a 7-day period it would have been 47 percent. Since 3 cloudy days out of 7 represents an unusually high frequency of overcast days for this locale, it is conjectured that a 50-percent split would be a more representative performance estimate for the current system given typical weather conditions for summer operation.

Preliminary evaluation of data accumulated during winter operation indicates that the system provides nearly 100 percent of the thermal energy required for heating; however, evaluation of the heating data has not been completed.

Studies being run outside the scope of these tests indicate the possibility of decreasing absorptive refrigerator operating temperatures. There are several advantages of operating at lower temperatures. The lower operating temperatures would allow more energy storage per day by increasing the collector efficiency and total collector on-time. A significant decrease in the operating temperature may also allow use of an unpressurized water storage container with a corresponding reduction in system costs. Other advantages lie in a reduction in heat leak and the less severe environment (i.e., lower temperatures and pressures) to which the equipment would be subjected.

Although a significant improvement in energy utilized by the air conditioner was realized with the discrete bypass control system, further improvement may be realized using a variable flow rate or constant temperature system. This control can be accomplished via a variable speed pump, a throttling valve, a pump bypass valve, a temperature mixing valve, or a combination of these devices. Either of these devices can use the hot water supply temperature to the air conditioner to control flow rate or temperature to the air conditioner. When executed properly, a variable flow rate or constant temperature device can provide a relatively constant energy input to the air conditioner. Another technique for improving air conditioner performance might be the use of ground or well water for cooling the air conditioner condenser, since in this region this water will typically be cooler than water cooled by the evaporative cooling tower technique.

The currently demonstrated average energy split can be improved. Several methods that may be used to accomplish improvements are:

- Increasing the collector surface area.
- Modifying the collector design to improve collection efficiency.
- Advance the absorption air conditioner state-of-the-art by lowering the generator hot water operating temperature.
- Lowering the average air conditioner energy consumption rate by improving the control system.
- Eliminating "flow channeling" in the energy storage tank.
- Modifying the energy storage tank and line routing to decrease the collector fluid supply temperature.

Operation of the MSFC facility to provide an engineering evaluation of solar heating and cooling systems for buildings is continuing. Some of the potential improvements listed above will be evaluated in the existing facility.

APPROVAL

SOLAR POWERED RESIDENTIAL HEATING AND COOLING SYSTEM DEVELOPMENT TEST PROGRAM

By William R. Humphries and Darrell E. Melton

The information in this report has been reviewed for security classification. Review of any information concerning Department of Defense or Atomic Energy Commission programs has been made by the MSFC Security Classification Officer. This report, in its entirety, has been determined to be unclassified.

This document has also been reviewed and approved for technical accuracy.

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120

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