DOE/CS/31591--T6

Phase II
Technical Progress Report

DEVELOPMENT OF A SOLAR DESICCANT DEHUMIDIFIER

81-17773

March 27, 1981

Prepared by
J. Rousseau

AC03-77CS31591

Prepared for
U.S. Department of Energy
Contract EG-77-C-03-1591

AIRESEARCH MANUFACTURING COMPANY

DISCLAIMER

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GARRETT

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FOREWORD

This technical progress report presents test data obtained as part of the development program conducted to fully characterize a desiccant dehumidifier system using low-temperature solar thermal energy for regeneration. The work described herein constitutes the first part of the Phase II effort and is concerned with off-design testing in the recirculated mode configuration.

This program is funded by the Department of Energy, San Francisco Operations Office, under contract EG-77-C-03-1591. The period covered by this technical report is from September 1, 1980 through February 28, 1981. Mr. R. S. Todaro is the DOE contracting officer, and Mr. C. J. Dankowski is the program coordinator. Technical monitoring is the responsibility of Mr. K. Collier of the Solar Energy Research Institute (SERI).

Mr. J. Rousseau is the program manager and principal investigator for AiResearch, and Mr. J. D. McPherson is the contract administrator. The experimental work on the solar desiccant air conditioner (SODAC) is conducted in the controlled atmosphere chambers of the Dunham-Bush facility in Harrisonburg. Mr. P. Rublee is responsible for the Dunham-Bush portion of this effort.

To provide continuity and to facilitate comprehension without having to refer back to other documents previously published under this contract, some data are repeated in this report; specifically, the system description and some of the off-design test data obtained in Phase I are presented.

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SECTION 1

INTRODUCTION AND SUMMARY

BACKGROUND

This program was initiated in September 1977. The first phase covered a three-year period and was concerned with the design, fabrication, and development testing of the 1.5-ton solar desiccant air conditioner (SODAC). The Phase I test program was also concerned with (1) configuration development, (2) design point optimization, and (3) off-design performance characterization. The second phase of the program started on September 1, 1980, and is scheduled for completion in August 1981. This is the first technical report published in Phase II; it summarizes the first six-month effort on the program.

The Phase II effort is a continuation of the development testing and is concerned with determination of the SODAC performance in the recirculated and ventilated mode configuration (see Figure 1-1). As originally conceived, the SODAC features two-speed indoor and outdoor fans to permit more efficient operation at reduced capacity. The test program covers testing over the entire range of interfacing parameters at both flow conditions.

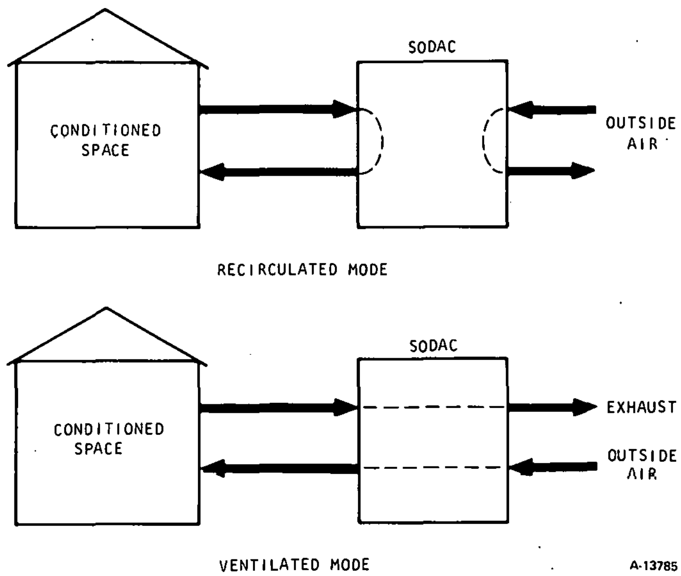


Figure 1-1. Recirculated and Ventilated Mode Flow Paths

This report presents the test data in the recirculated mode. Some of data obtained in Phase I are repeated in this report for completeness. In both full-flow and half-flow cases, the experimental data are compared to computer predictions.

To facilitate comprehension, the description of the system and its operation is presented in Section 2. This description is taken from the Phase I final summary report. Section 3, which contains the test data, also repeats the description of the system test facility and procedures.

Previous documentation under the SODAC program is listed under references at the end of this section. These references are included for details on the design, economic evaluation, and construction of the machine.

CONCEPT

The baseline SODAC schematic is shown in Figure 1-2. The arrangement of the rotary desiccant bed and regenerator is depicted in Figure 1-3. For comparison with competing desiccant system approaches, the system incorporates the following important features:

(a) Granular silica gel is used as the desiccant.

(b) Outside air precools the bed before it enters the adsorbing zone; heat removed from this portion of the bed is used to preheat the bed as it enters the desorbing zone. Thus, the recovered thermal energy results in a 25-percent increase in coefficient of performance (COP).

(c) The silica gel bed and regenerator are packaged in thin cylindrical drums; this arrangement provides large flow areas and minimum bed depth, resulting in a small pressure drop.

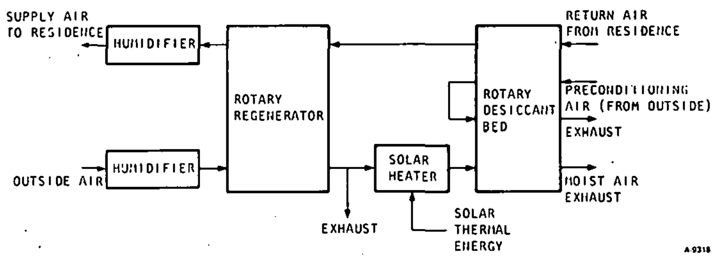


Figure 1-2. System Schematic

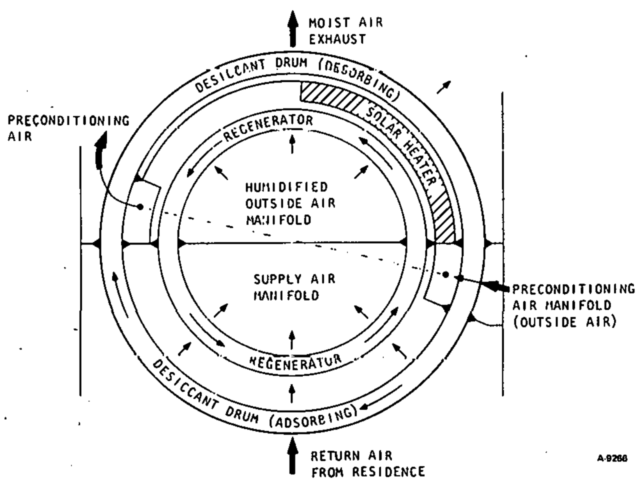


Figure 1-3. Dehumidifier Cross Section

PHASE II OBJECTIVE

The objective of Phase II is the experimental evaluation of the SODAC over a range of interfacing parameters defined by the following:

(a) Indoor wet and dry bulb temperatures

(b) Outdoor wet and dry bulb temperatures

(c) Hot-water temperature representing the solar heat input

The test program also includes the following: (1) two system configurations--recirculated mode and ventilated mode, and (2) the design point air flow rates and half-flow rates corresponding to operation at high loads and reduced loads.

TECHNICAL ACCOMPLISHMENT

SODAC testing in the recirculated mode was initiated in Phase I. In Phase II, recirculated mode testing at full flow and half flow was completed. All test data obtained in Phase I and Phase II are included in this report for completeness. Design point performance defined by the ARI standard conditions and a 200°F solar energy (water) temperature level are listed in Table 1-1 for full-flow and half-flow conditions.

TABLE 1-1

DESIGN POINT PERFORMANCE
RECIRCULATED MODE

Indoor temperatures: 80°F dry bulb, 67°F wet bulb
Outdoor temperatures: 95°F dry bulb, 75°F wet bulb
Water source temperature: 200°F

EXPERIMENTAL DATA	Full Flow	Half Flow
Capacity, Btu/hr	16,200	7,400
Coefficient of performance	0.59	0.56
Conditioned space air flow, scfm	760	380
Outdoor air flow, scfm	925	490
Water usage, gal/hr	3.9	2.0
Parasitic power (fans, pumps and and drive), kw	0.8	0.24
PREDICTED DATA		
Capacity, Btu/hr	16,200	8,000
Coefficient of performance	0.52	0.46
Parasitic power, kw	0.75	

The experimental data obtained agrees very well with computer predictions, with the actual COP being somewhat higher (10 to 15 percent) than predicted, and a capacity within 10 percent of prediction at half flow.

Off-design performance in terms of capacity and COP are plotted in Figures 1-4, 1-5, and 1-6. In these plots, all data were reduced to ARI standard conditions and a 200°F water (solar energy) temperature, except for the parameters in abscissa. Climate simulation test runs conducted were reduced to ARI standard conditions using the scope of the curves in Figures 1-4 through 1-6; the test runs were shown to correlate fairly accurately with design point data obtained on the test. This is significant because it confirms the method used for year-round performance prediction in Phase I of the program. The series of tests performed completes the characterization of the SODAC in the recirculated mode.

FUTURE WORK

The system configuration will be changed to the ventilated mode, and performance data will be generated at full-flow and half-flow over a scope of interfacing parameters.

REFERENCES

1. Gunderson, M.E., K.C. Hwang, and S.M. Railing, Development of a Solar Desiccant Dehumidifier, First Technical Progress Report, U.S. Department of Energy Report SAN-1591-1, March 1978.

2. Gunderson, M.E., K.C. Hwang, S.M. Railing, and J. Rousseau, Development of a Solar Desiccant Dehumidifier, Second Technical Progress Report, U.S. Department of Energy Report SAN-1591-2, November 1978 (in publication).

3. Gunderson, M.E., Development of a Solar Desiccant Dehumidifier, Third Technical Progress Report, U.S. Department of Energy Report SAN-1591-3, June 1979.

4. Rousseau, J., Development of a Solar Desiccant Dehumidifier, Phase I Final Summary Report, AiResearch Report 80-17481, September 1980 (to be published by DOE).

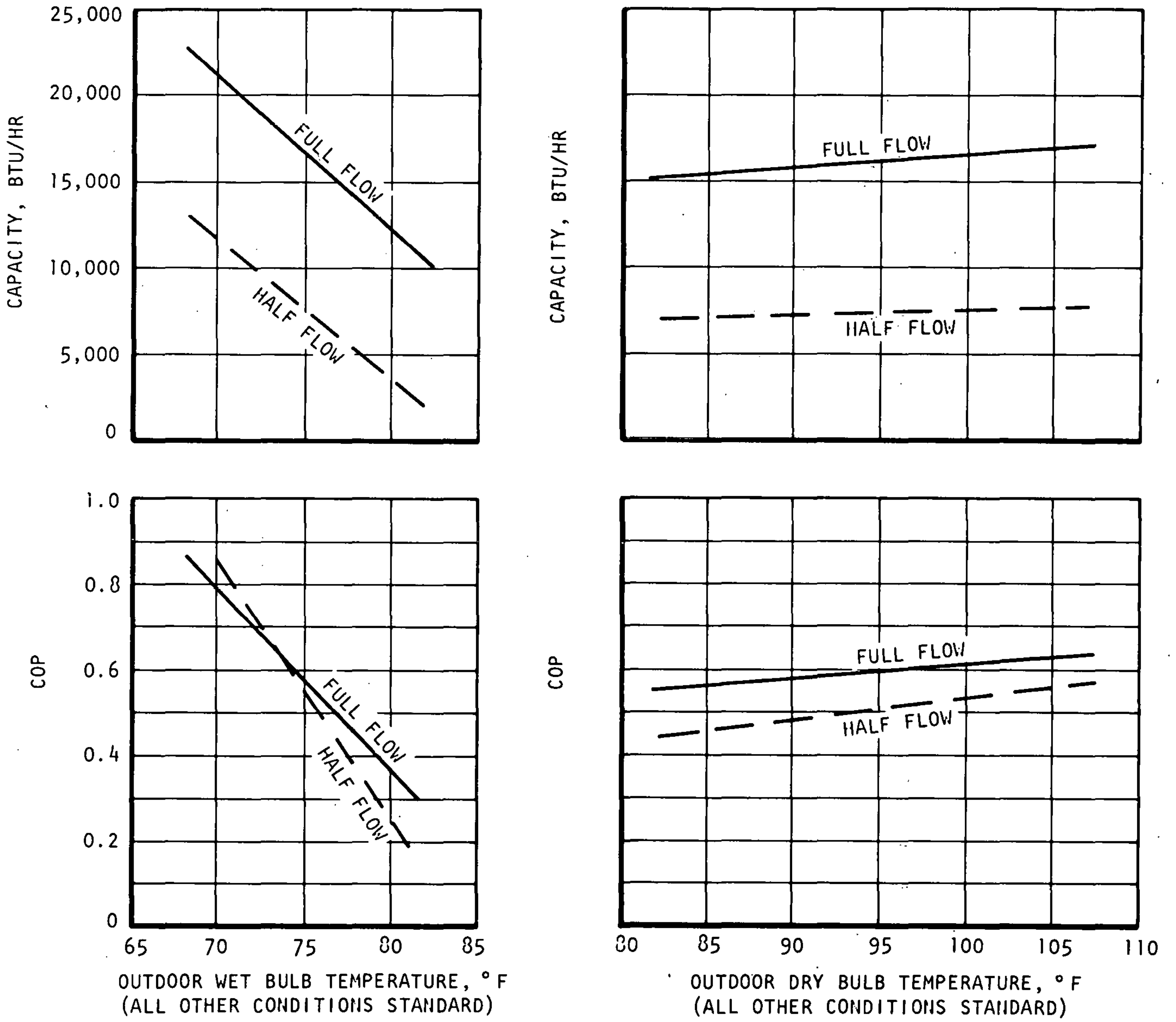


Figure 1-4. SODAC Performance--Effect of Outdoor Temperature

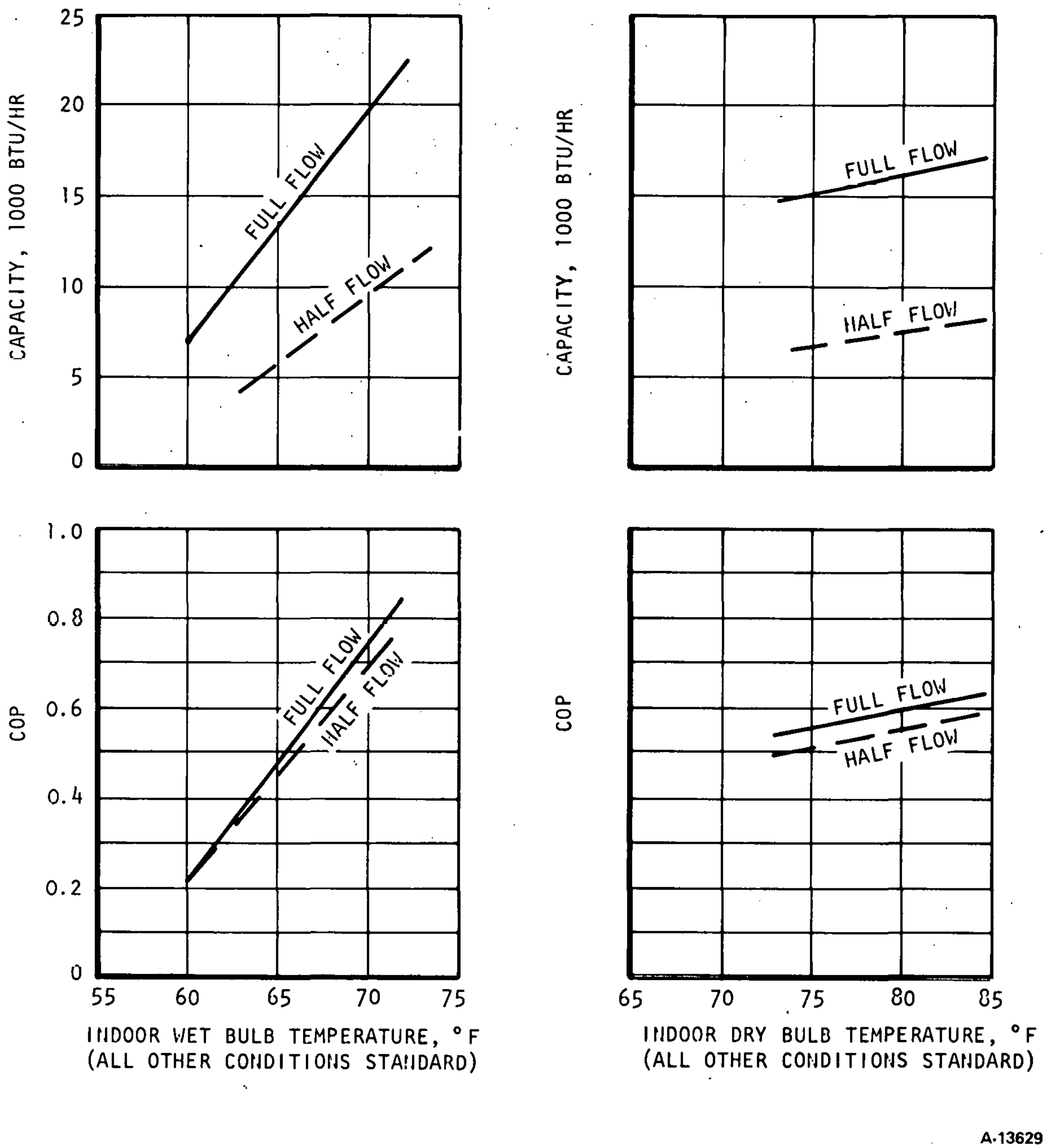


Figure 1-5. SODAC Performance--Effect of Indoor Temperatures

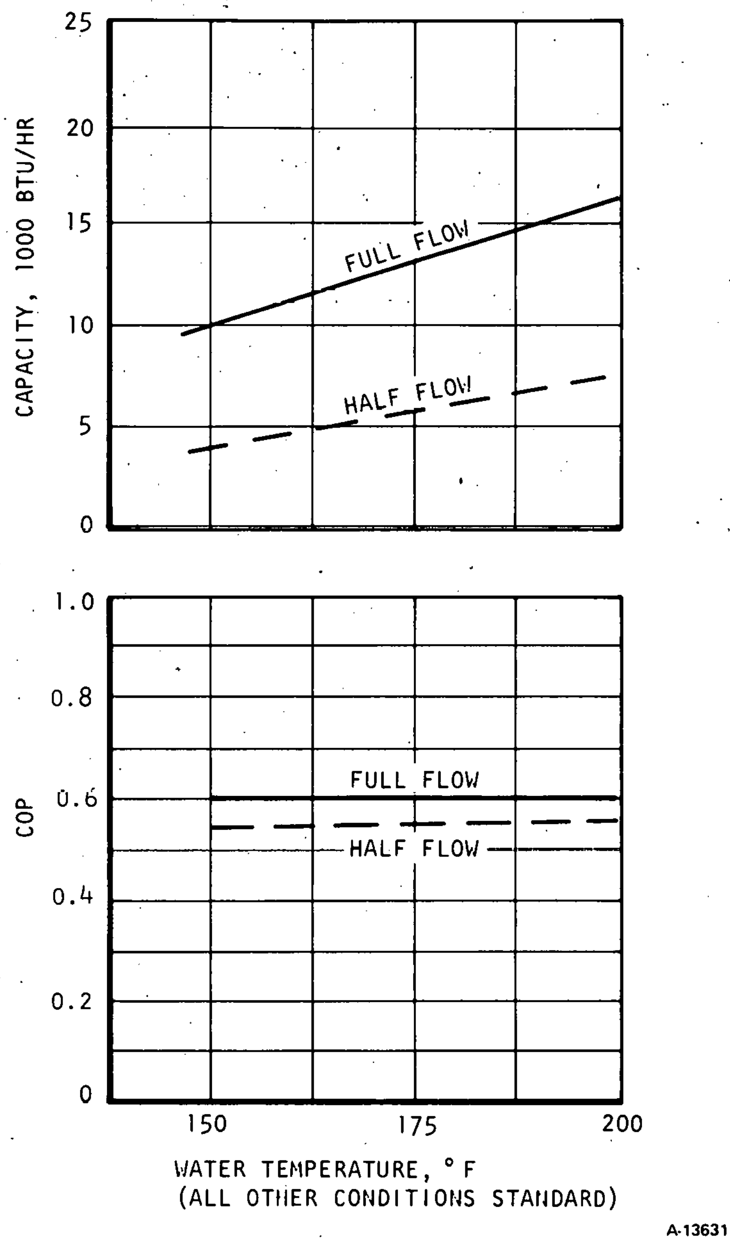


Figure 1-6. SODAC Performance--Effect of Water Temperature

SECTION 2

SYSTEM DESCRIPTION

GENERAL

This section includes a description of the SODAC system and its operation; the characteristics of the major components; the performance at design conditions; and the control schemes for optimum operation in various climates.

FUNCTIONAL DESCRIPTION

A schematic of the system is given in Figure 2-1; the flow path shown is the recirculated mode operation. The desiccant bed and the regenerator are two thin cylinders rotating around parallel axes. The desiccant is granular silica gel, and the regenerator matrix is a fine screen of galvanized steel. A top view of the package along the axes of rotation is shown in Figure 2-2 (the humidifiers are not shown). The solar heater is located between the dryer and the regenerator.

The operation of the SODAC system is as follows. Warm humid air from the residence is directed to the adsorbing side of the rotary dryer. Water is adsorbed from the air stream, which is heated in the process. The air is then cooled in the rotary regenerator. The specific humidity of this air stream is sufficiently low so that its dry bulb temperature can be lowered by adiabatic humidification to levels adequate for sensible cooling, while still retaining reasonable latent cooling capacity. Ambient outside air is used to regenerate the sorbent bed and to cool the rotary regenerator. This stream is humidified adiabatically and recirculated through the rotary regenerator. About one-half of this air is then exhausted from the package without flowing through the dryer. The remainder is heated in a fixed-boundary heat exchanger by solar thermal energy and used to desorb the desiccant.

A minor amount of air directly from outside is circulated through the hot portion of the sorbent bed as it rotates from the desorbing zone to the adsorbing one. In this manner, the bed is cooled to a temperature level where it can adsorb moisture when exposed to the return air from the residence. This preconditioning airflow is then used to preheat the bed prior to desorption, thus reducing the solar thermal energy necessary for this process.

The processes occurring within the three airstreams are illustrated in the psychrometric plot of Figure 2-3. The state points shown correspond to those of Figures 2-1 and 2-2. This plot represents only an approximation since the temperatures and humidity content of the air vary along the periphery of the regenerator and desiccant beds as they rotate in opposite directions. Also, the drying and desorbing processes are not strictly adiabatic because of the heat capacity of the sorbent itself.

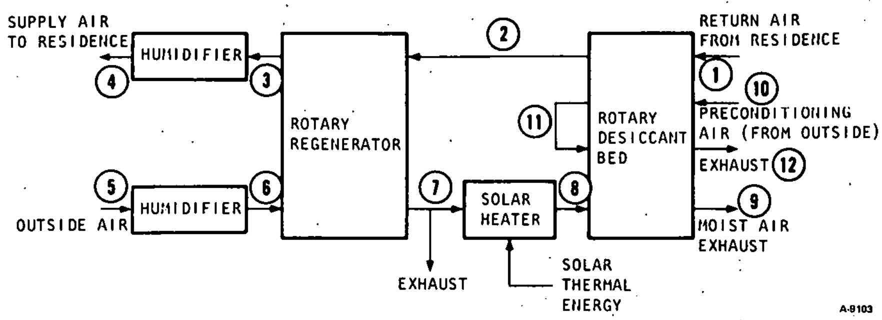


Figure 2-1. Air Conditioner Schematic

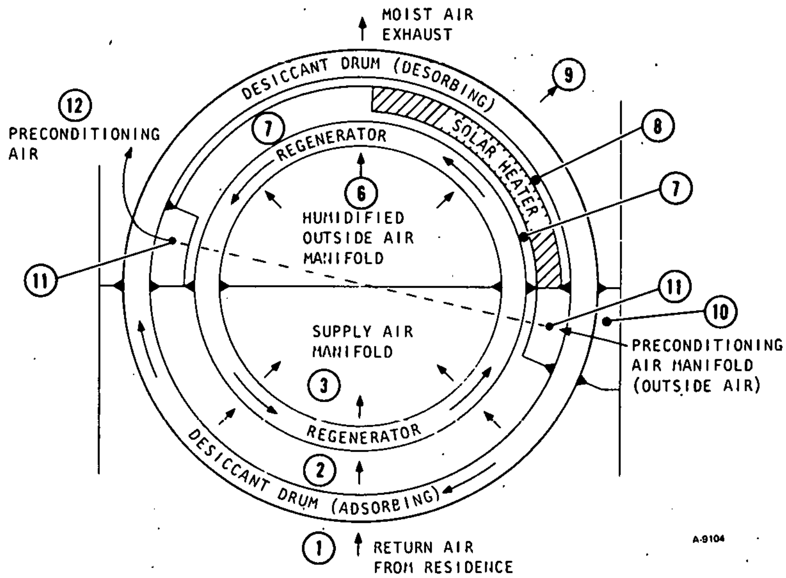


Figure 2-2. Dehumidifier Arrangement

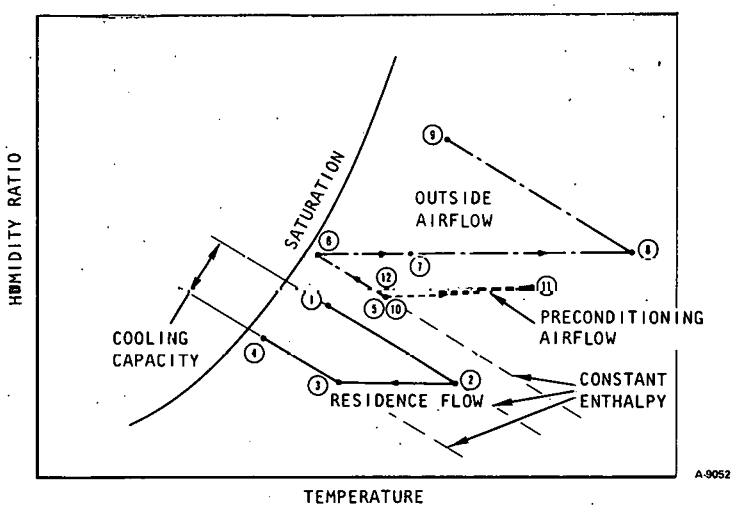


Figure 2-3. Psychrometric Process

BASELINE SYSTEM CHARACTERISTICS

Physical Characteristics

The SODAC system was designed to provide 1.5 tons (nominal) of air conditioning under the following conditions:

Conditioned space temperatures	80°F dry bulb 67°F wet bulb
Outside air temperatures	95°F dry bulb 75°F wet bulb
Solar energy source temperature	200°F

Pertinent characteristics of the major system components were determined from computer studies and are listed in Table 2-1.

TABLE 2-1

BASELINE COMPONENT CHARACTERISTICS SUMMARY

Cooling capacity: 1.5 tons (18,000 Btu/hr)

Desiccant Bed	Regenerator
8 to 10 mesh silica gel	24 x 24 x 0.014 in. stainless steel screen
Bed inside diameter: 31.3 in.	Matrix inside diameter: 19.0 in.
Bed active height: 34.7 in.	Matrix active height: 34.5 in.
Bed thickness: 1.25 in.	Matrix thickness: 1.13 in.
Bed weight (dry): 110 lb	Matrix weight: 165 lb
Rotating speed: 5 rph	Rotating speed: 20 rpm
Working capacity: 3.1 percent	Effectiveness: 90 percent
Pressure drop: 0.63 in. H20	Pressure drop: 0.19 in. H_2O
Airflow Rates	**Solar Heater**
Residence airstream: 850 scfm	Effectiveness: 85 percent
Preconditioning airstream: 120 scfm	Arc: 86.6 deg
Outside airstream (without preconditioning air): 830 scfm	Heating rate: 35,000 Btu/hr
	Water flow rate: 3600 lb/hr
Solar heater airstream: 455 scfm	Pressure drop (air side): 0.04 in. H20
Preconditioning Air	**Mechanical Drive**
Manifold arc: 22.5 deg	Power requirement: 0.1 kw (max.)

Note: Design point calculations made at following standard conditions:

Conditioned space: 80°F dry bulb, 67°F wet bulb
Outside: 95°F dry bulb, 75°F wet bulb
Hot water supply temperature: 200°F
Barometric pressure: 14.7 psia

Predicted Design Point Performance

The performance of the system at the design conditions listed above is defined in Figure 2-4, and the pertinent data on the system performance are listed below.

- The thermal COP of the system (cooling capacity/solar thermal energy input) is 0.52.
- The pressure drop through the system is 1.3 in. H_2O.
- The temperature gradient across the regenerator is about 27°F from the outer diameter to the inner diameter.
- The average temperature of the supply air at regenerator outlet is 81°F.
- The desorbing air temperature at the solar heater outlet varies from 176° to 186°F in the clockwise direction.

About 30 percent of the total capacity of the system at design point is for latent heat removal. The ratio of latent to sensible heat capacity increases rapidly as outside air wet bulb temperature drops. The latent capacity of the supply airstream could be increased by bypassing the humidifiers or by using a less effective humidifier.

In its final configuration, the desiccant drum could not be loaded with the nominal 110 lb of silica gel granules, as listed in Table 2-1. The introduction of vertical partitions around the periphery of the drum resulted in a loss of about 10 lb of silica gel. Since the capacity of the machine is directly propotional to the quality of silica gel contained in the sorbent bed, the actual capacity of the machine was estimated at 1.35 tons. The process air flow rates were reduced by the same ratio.

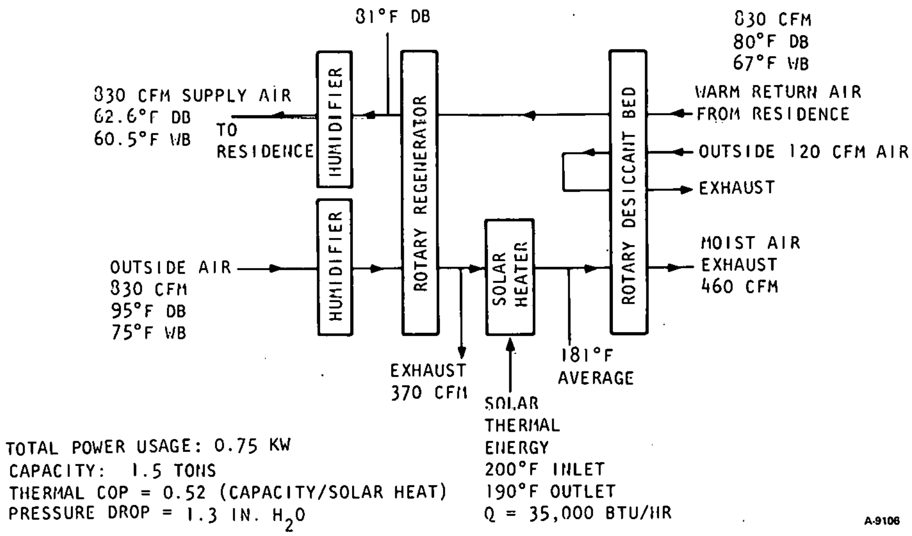


Figure 2-4. Baseline SODAC Performance

SYSTEM CONTROL

Comfort is generally given as function of dry bulb temperature and one other variable that indicates humidity, such as wet bulb temperature or relative humidity. Both wet bulb temperature and relative humidity are difficult to sense accurately and repeatedly with simple instruments. It is therefore not practical to use humidity as a control parameter for a residential air conditioner.

Figure 2-5 illustrates the SODAC control scheme on four set temperatures--T_{AUX}, T_{MAX}, T_{MIN}, and T_{OFF}. The system is controlled based on the conditioned space dry bulb temperature. This control scheme has been selected for two primary reasons. First, experimentation has shown that good comfort levels can be maintained without sensing controlled space humidity. Depending upon the location and the system configuration, a set of four control temperatures can be selected that will match closely the latent and sensible cooling capacity to the latent and sensible loads.

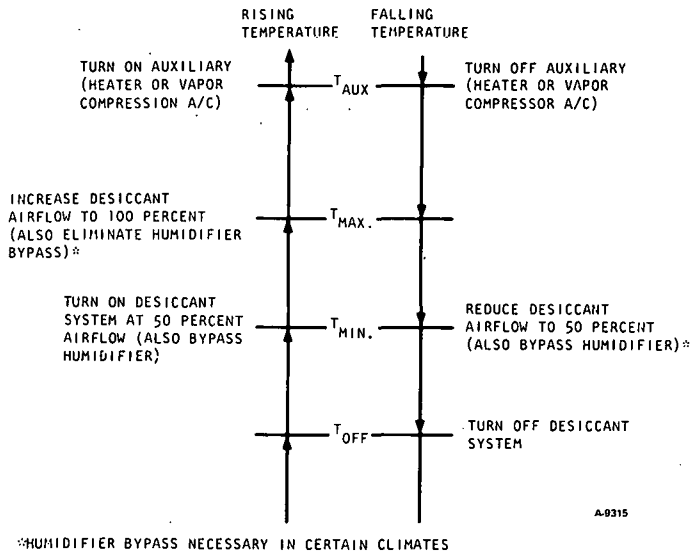
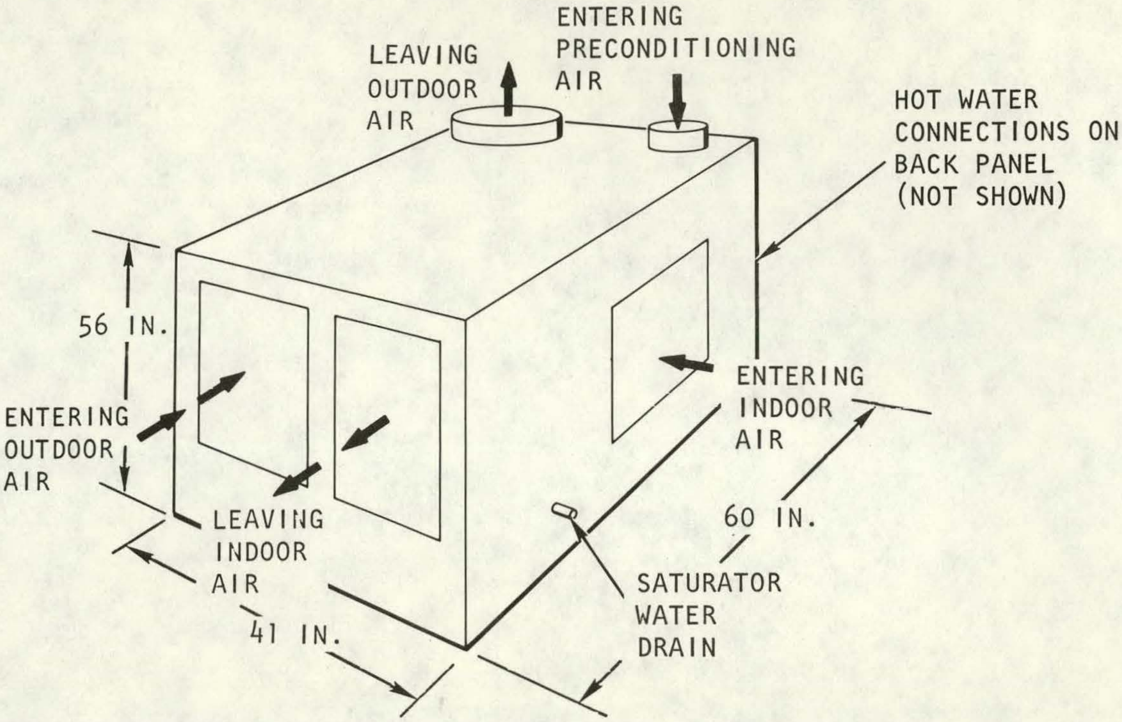


Figure 2-5. Control Scheme

Upon initial startup, several rotational cycles of the desiccant bed are required before steady-state operation is achieved. Steady-state operation is achieved in approximately 0.5 hr. It does not appear practical to cycle the system on and off to match average capacity to the load as is the practice with conventional vapor compression air conditioners. This problem is overcome by reducing the speed of the fans, and thus the airflow and capacity, when the temperature of the controlled space drops below a value that would not require full capacity. With this method, the desiccant bed is kept in operation and the transient startup problem is avoided.

SODAC PACKAGE

The SODAC package incorporates all equipment depicted on the schematic shown in Figure 2-1, including the drive for the drums and the water system associated with operation of the humidifiers. The fans required to circulate the air through the system are not included. A pictorial view of the package showing major interfaces, and a photograph of the unit with a side panel removed are shown in Figure 2-6. The overall dimension of the package, excluding duct attachment points, are 60-in. long by 41-in. wide by 56-in. high.



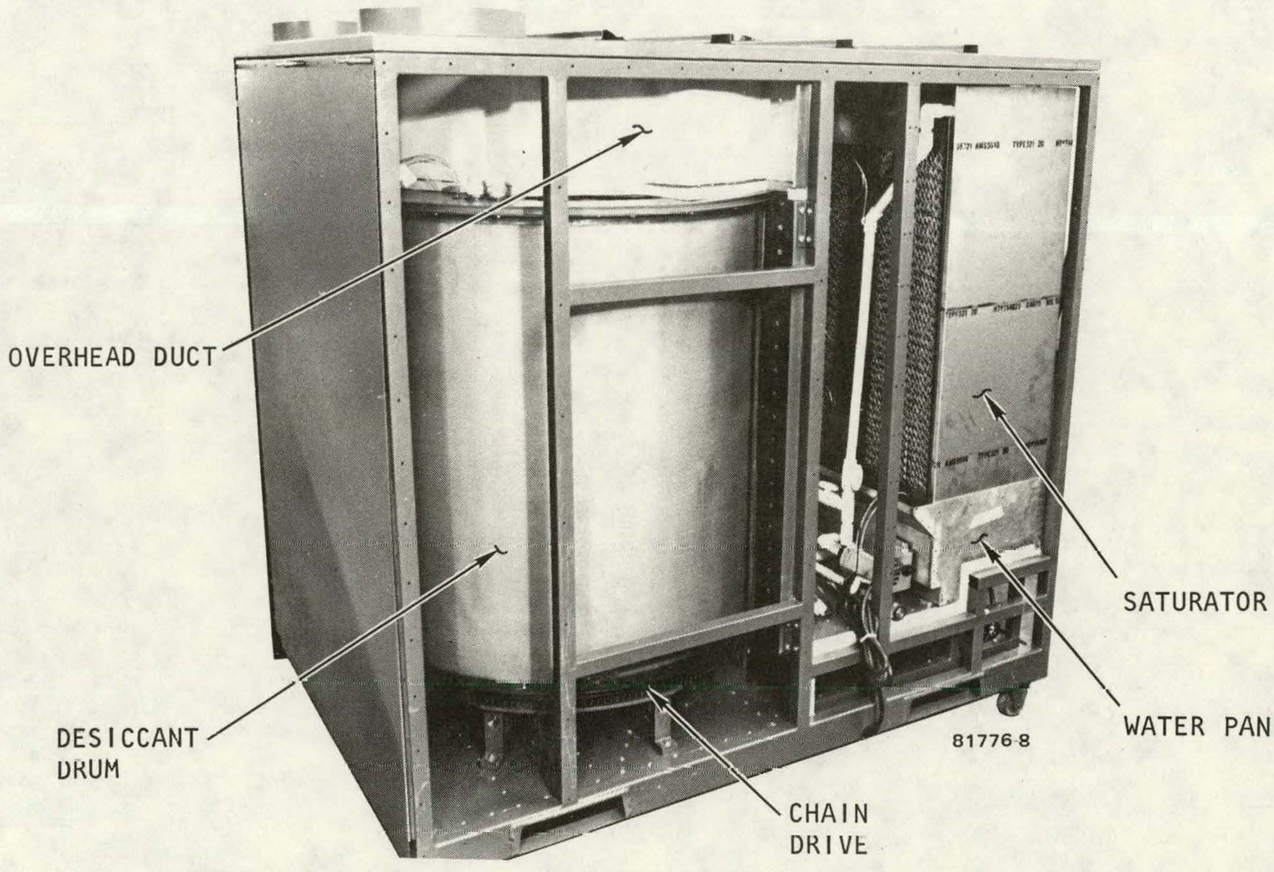


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Figure 2-6. SODAC Package

SECTION 3

SYSTEM TEST

GENERAL

The test program was conducted at the Dunham-Bush facility in Harrisonburg, Virginia. This facility was selected because of the availability of fully instrumented controlled atmosphere test chambers of a suitable size and capacity.

TEST FACILITIES AND EQUIPMENT

A schematic of the test stand is shown in Figure 3-1. The airflow measuring apparatus, room conditioning apparatus, temperature measurement instruments, and manometers are all existing equipment at Dunham-Bush, and all conform to the requirements of ASHRAE 37-69. The hot water supply is also existing equipment that has been used to test other solar equipment.

The airflow and room conditioning apparatus for the outdoor side test room has a capacity of 5000 cfm; the indoor side equipment has a capacity of 2500 cfm. Hot water can be delivered at 25 gpm. Adjacent to the test rooms is a control room with sufficient equipment to support the instrumentation, as defined in the following text.

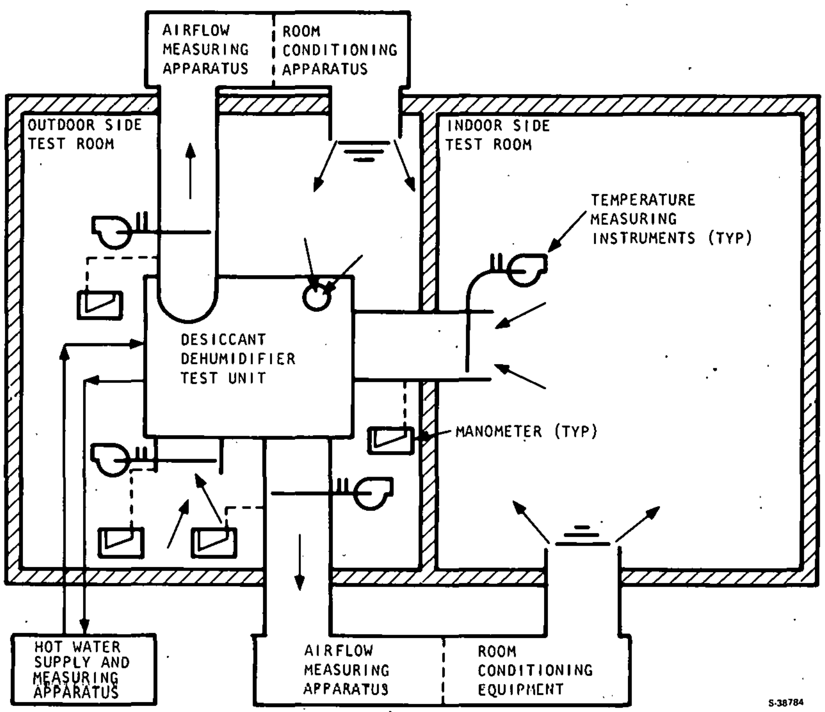


Figure 3-1. Test Stand Schematic

INSTRUMENTATION

The location of some instruments is illustrated in Figure 3-2. All the parameters measured during the preliminary testing and performance characterization work are listed in Table 3-1.

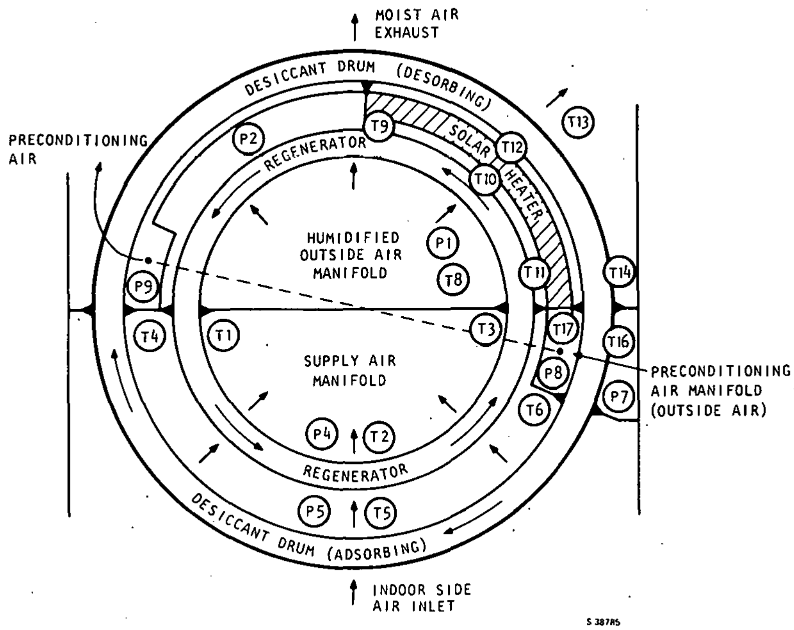


Figure 3-2. Location of Internal Pressure Taps and Temperature Probes

TEST PROCEDURE

Multipoint recorders connected to a few strategic thermocouples were used to monitor chamber and desiccant system status. Thermocouples throughout the system can also be used to determine stability of operation of the chambers and system. Data were taken no sooner than 25 min after equilibrium was reached. Since the desiccant drum rotational cycle is 12 min, this 25-min interval represents three successive cycles at the stable condition.

TEST PROGRAM

The Phase II test program completes the recirculation mode testing initiated in Phase I and includes the following:

(a) Full-flow off-design tests simulating hot-dry and hot-humid climates

(b) Half-flow design and off-design tests, where one interfacing parameter (i.e., indoor and outdoor wet and dry bulb temperatures, and water temperature) is varied from standard conditions

(c) Half-flow dry and humid climate simulation

A total of 40 test runs were conducted, including tests repeated due to obvious errors, and for system calibration.

TABLE 3-1

TEST INSTRUMENTS

Tag No.	Data	Range	Units	Accuracy
T1	Regenerator outlet temperature	70 to 110	°F	±1°F
T2	Regenerator outlet temperature	70 to 110	°F	±1°F
T3	Regenerator outlet temperature	70 to 110	°F	±1°F
T4	Desiccant drum outlet temperature	75 to 200	°F	±1°F
T5	Desiccant drum outlet temperature	75 to 200	°F	±1°F
6	Desiccant drum outlet temperature	75 to 200	°F	±1°F
T7	Indoor side entering air temperature	50 to 100	°F	±0.1°F
T8	Outside air manifold temperature	50 to 100	°F	±1°F
T9	Solar heater air inlet temperature	50 to 100	°F	±1°F
T10	Solar heater air inlet temperature	50 to 100	°F	±1°F
T11	Solar heater air inlet temperature	50 to 100	°F	±1°F
T12	Solar heater air outlet temperature	100 to 225	°F	±1°F
T13	Desorbing air outlet temperature	100 to 225	°F	±1°F
T14	Desorbing air outlet temperature	100 to 225	°F	±1°F
T15	Outdoor side leaving air temperature	70 to 200	°F	±0.1°F
T16	Preconditioning air inlet temperature	70 to 100	°F	±1°F
T17	Preconditioning air crossover temperature	70 to 200	°F	±1°F
T18	Water inlet temperature	100 to 220	°F	±0.1°F
T19	Water outlet temperature	100 to 200	°F	±0.1°F
T20	Indoor side saturator water temperature	40 to 80	°F	±1°F
T21	Outdoor side saturator water temperature	50 to 100	°F	±1°F
T22	Indoor side leaving air	40 to 80	°F	±0.1°F
T23	Outdoor side entering air temperature	70 to 100	°F	±0.1°F
T24	Indoor side nozzle temperature	40 to 80	°F	±0.1°F
T25	Outdoor side nozzle temperature	70 to 100	°F	±0.1°F
T26	Outdoor side leaving wet bulb	50 to 100	°F	±0.1°F
T27	Outdoor side entering wet bulb	50 to 100	°F	±0.1°F
T28	Indoor side leaving wet bulb	40 to 80	°F	±0.1°F
T29	Indoor side entering wet bulb	40 to 80	°F	±0.1°F
T30	Outdoor side nozzle wet bulb	50 to 100	°F	±0.1°F
T31	Indoor side nozzle wet bulb	40 to 80	°F	±0.1°F
P1	Outside air manifold pressure	0 to -2	in. H_2O	±0.01 in. H_2O
P2	Exhaust plenum pressure	0 to -2	in. H_2O	±0.01 in. H_2O
P3	Outdoor side leaving pressure	0 to -3	in. H_2O	±0.01 in. H_2O
P4	Supply air manifold pressure	0 to -3	in. H_2O	±0.01 in. H_2O
P5	Desiccant drum outlet pressure	0 to -3	in. H_2O	±0.01 in. H_2O
P6	Indoor side leaving air pressure	0 to -2	in. H_2O	±0.01 in. H_2O
P7	Preconditioning air inlet pressure	0 to -1	in. H_2O	±0.01 in. H_2O
P8	Preconditioning air crossover pressure	0 to -2	in. H_2O	±0.01 in. H_2O
P9	Preconditioning air outlet pressure	0 to -3	in. H_2O	±0.1 in. H_2O
P10	Indoor side leaving supply air pressure	0 to -3	in. H_2O	±0.01 in. H_2O
P11	Indoor side saturator inlet pressure	0 to -3	in. H_2O	±0.01 in. H_2O
P12	Outdoor side saturator outlet pressure	0 to -1	in. H_2O	±0.01 in. H_2O
P13	Water inlet pressure	0 to 25	psig	±1 psi
P14	Water outlet pressure	0 to 25	psig	±1 psi
P15	Outdoor side nozzle ΔP	0 to 12	in. H_2O	±0.01 in. H_2O
P16	Indoor side nozzle ΔP	0 to 12	in. H_2O	±0.01 in. H_2O
F1	Water flow rate	0 to 75	lbm/min	±0.1 lbm/min
S1	Drive shaft speed	0 to 200	rpm	±1 rpm
W1	Electric power input	0 to 500	w	--

FULL-FLOW TEST DATA

Initial testing indicated that the parameters specified as a result of the computer analyses represented near optimum operating conditions. Design point SODAC internal conditions corrected to the 1.35-ton capacity were defined by the following parameters.

Indoor flow rate	760 scfm
Outdoor flow rate	930 scfm (total)
Preconditioning airflow	190 scfm
Heater bypass orifice area	24 sq in.
Desiccant bed rotational speed	5 rph

Figure 3-3 shows typical data at design point. The sensitivity of the system to the interface parameters is shown in Figures 3-4 through 3-6. These data were presented in the Phase I summary report and are included in this report for completeness.

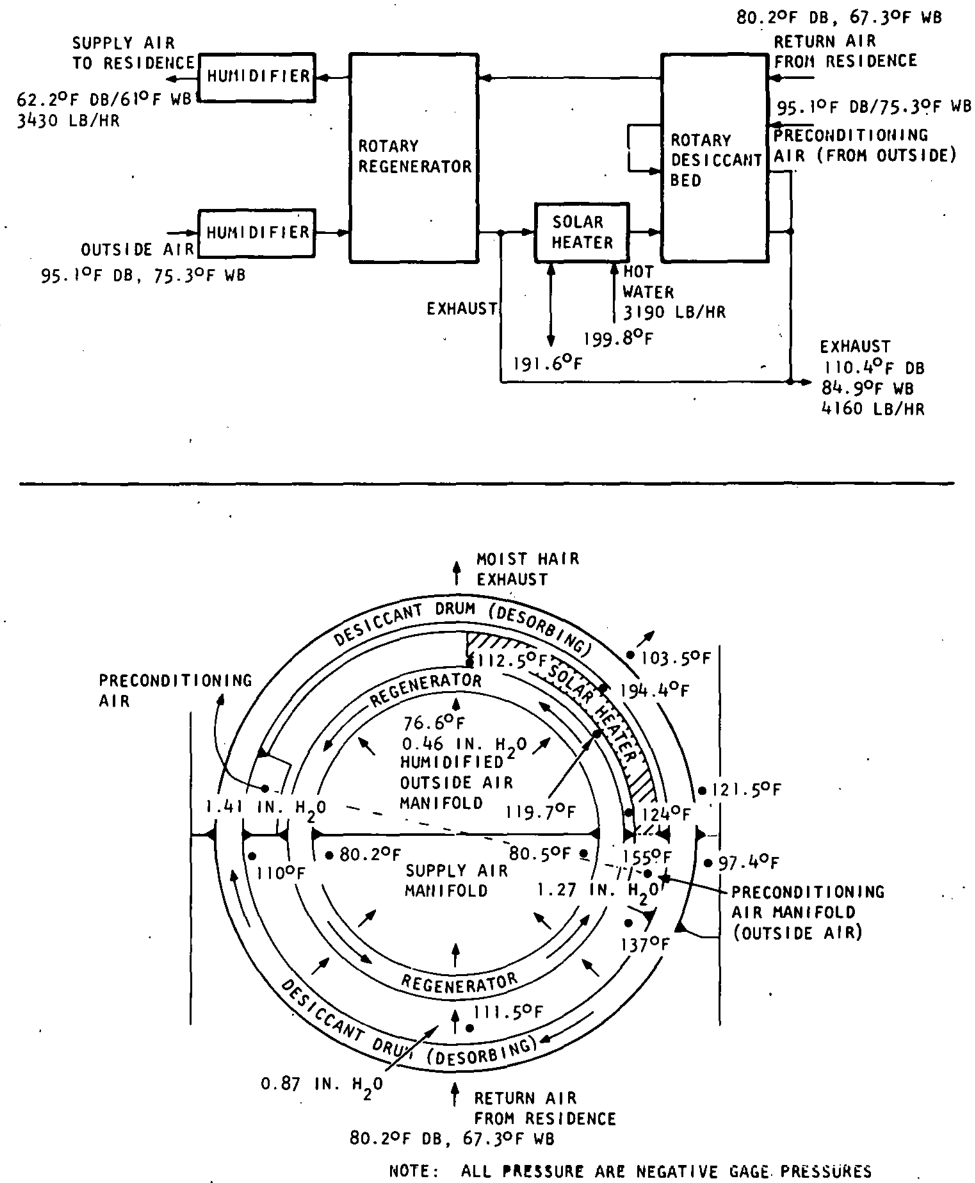
Table 3-2 summarizes SODAC performance at interfacing parameters simulating hot-dry and hot-humid climates. These data were corrected using the slopes of computer prediction curves in Figures 3-4 through 3-6. The off-design data, reduced in this manner to standard ARI conditions, are listed in Table 3-3 together with design point test data. The capacities and COP's obtained for the hot-humid climate simulations (Runs 11/5, 11/6, and 11/7) are somewhat low. However, considering the size of the correction factors, the correlations used appear to be fairly accurate.

HALF-FLOW TEST DATA

Calibration Data

Initial testing at half-flow indicated that the solar heater bypass area was slightly too large, and as a result of reduced heater air flow, capacities and COP's were lower than anticipated. This flow distribution problem was corrected by reducing the heater bypass area from 24 to 21 sq in. Performance at full flow and ARI conditions was used to determine the effect at full flow.

Table 3-4 shows that changing this area from 24 to 21 sq in. has only a negligible effect on the performance of the system at full flow; however, the effect at half flow is much more pronounced (see Table 3-5). All half-flow data presented are for the 21-sq in. bypass area. This change will not affect the full flow performance significantly.



Q SOLAR = 26,980 BTU/HR
CAPACITY AT TEST CONDITIONS: 16,400 BTU/HR
COP AT TEST CONDITIONS: 0.161
CAPACITY AT STANDARD CONDITIONS: 16,300 BTU/HR
COP AT STANDARD CONDITIONS: 0.60

A-9102

Figure 3-3. Typical Design Point Test Data

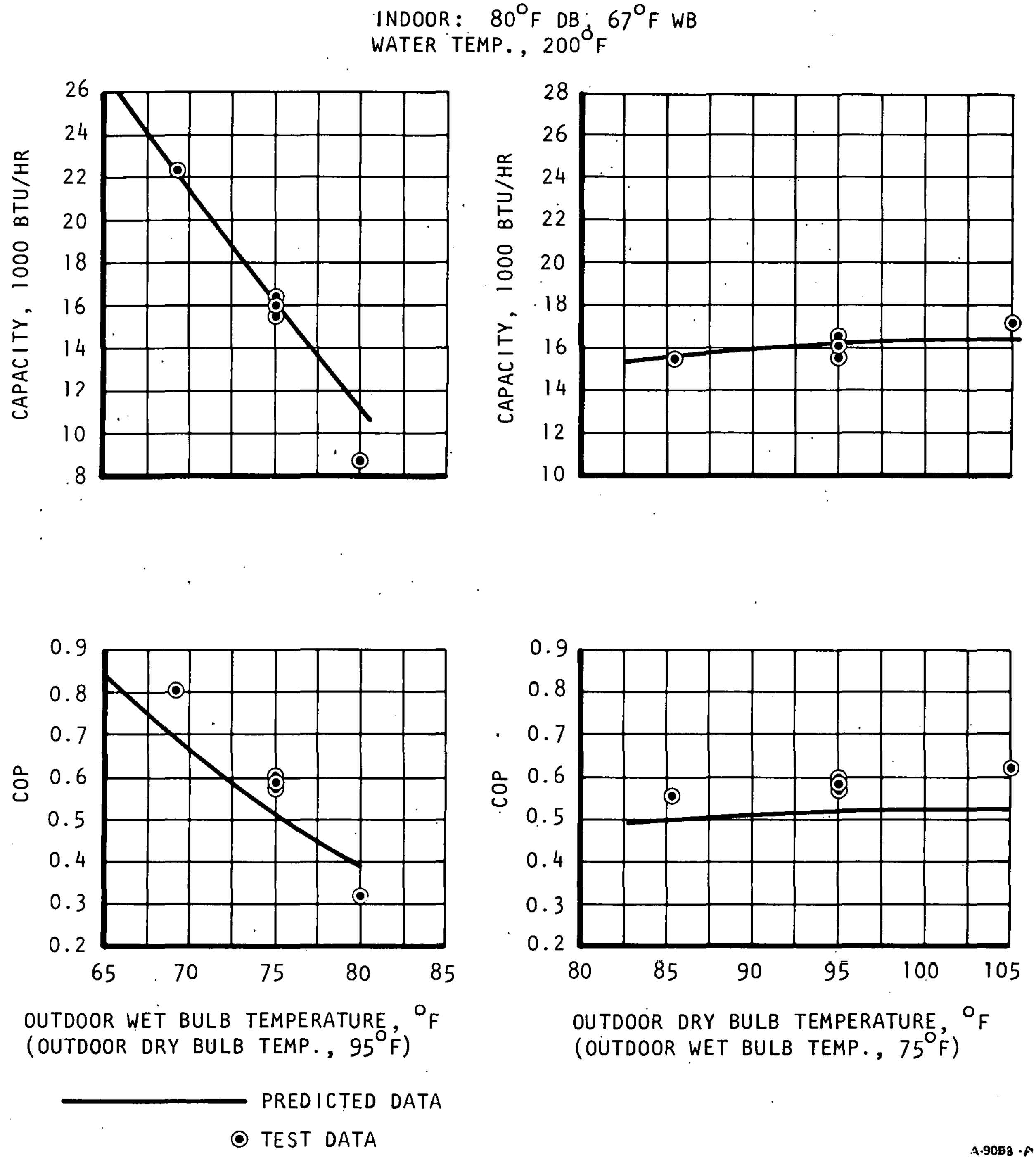


Figure 3-4. Design Flow Test Performance--Effect of Outdoor Temperatures

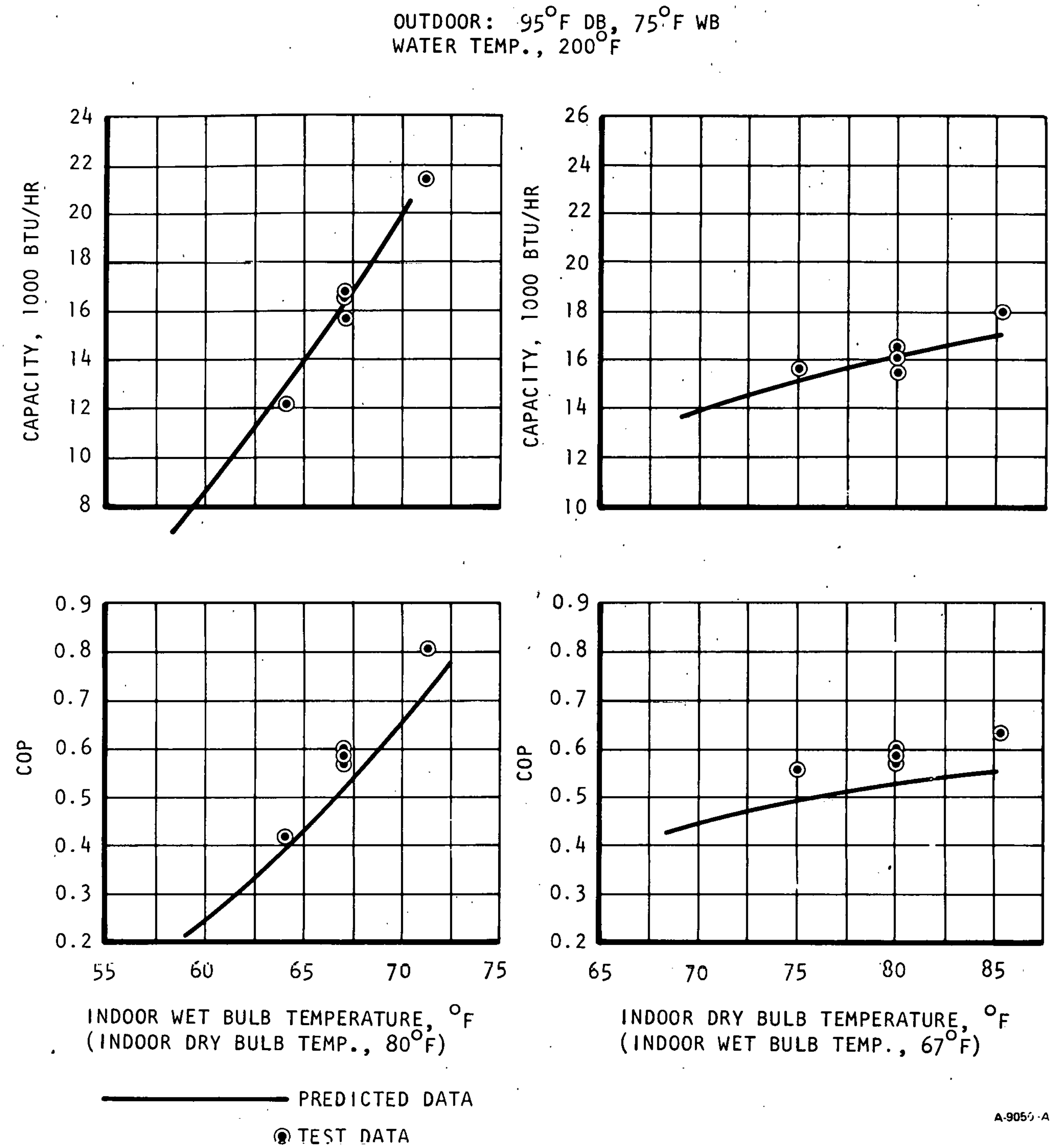


Figure 3-5. Design Flow Test Performance--Effect of Indoor Temperatures

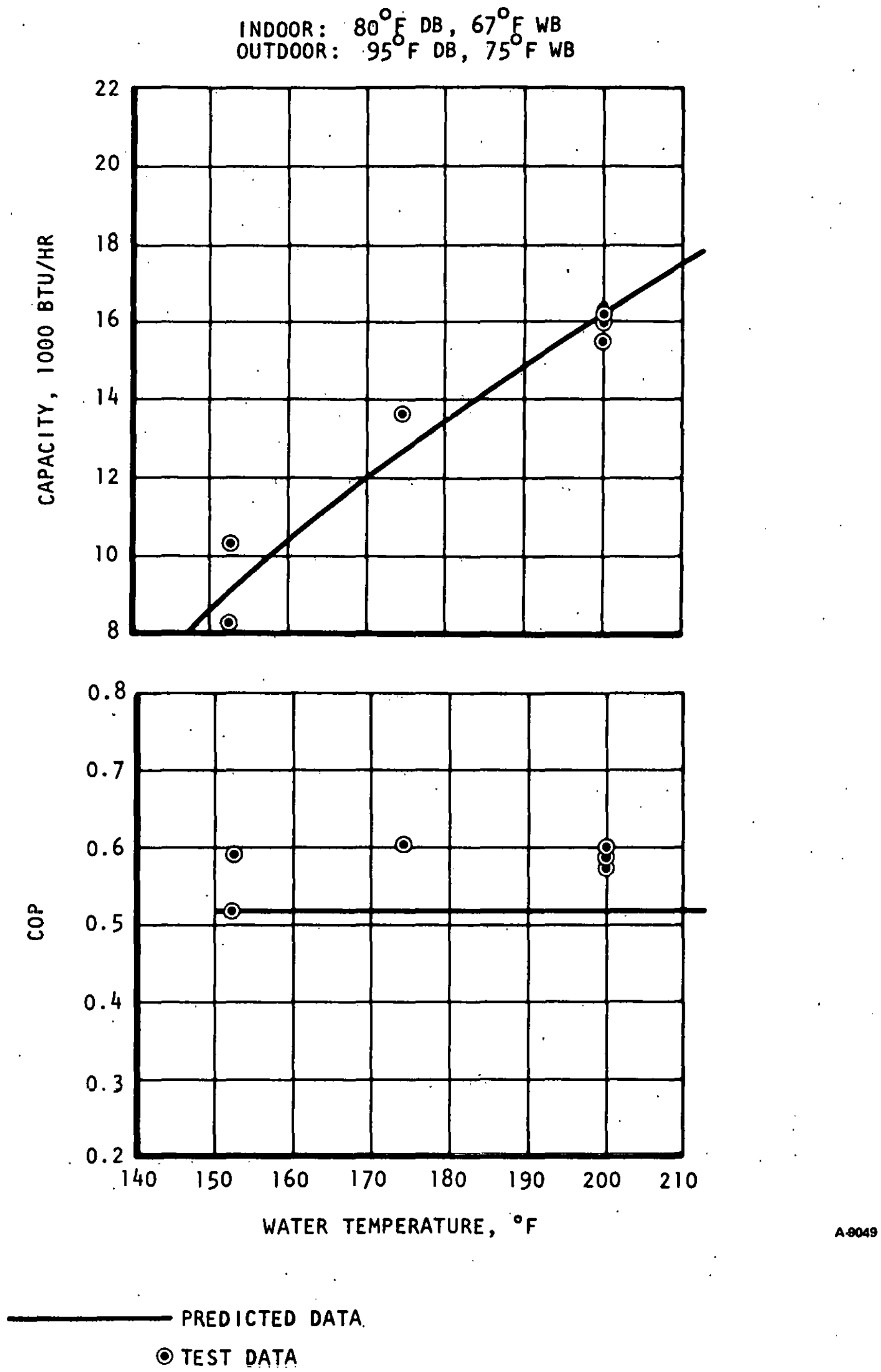


Figure 3-6. Design Flow Test Performance--Effect of Water Temperature

TABLE 3-2

FULL-FLOW CLIMATE SIMULATION DATA
RECIRCULATED MODE

		Test Conditions								Test Results				
		Outdoor			Indoor			Water		Indoor Outlet				
Run No.	Purpose	Inlet Dry Bulb, °F	Inlet Wet Bulb, °F	Flow, scfm	Inlet Dry Bulb, °F	Inlet Wet Bulb, °F	Flow, scfm	Inlet Temp, °F	Flow, lb/hr	Dry Bulb, °F	Wet Bulb, °F	Solar Q, Btu/hr	Capacity, Btu/hr	COP
11/2	Hot-dry climate simulation	106.9	75.3	908	80.2	67.6	761	198	3290	61.9	61.1	26,600	17,100	0.64
11/3	Hot-dry climate simulation	107.2	75.1	1001	80.2	66.9	766	173.8	3290	62.0	61.2	23,000	14,900	0.65
11/4	Hot-dry climate simulation	106.8	74.5	918	75.6	63.6	770	198.9	3290	59.2	58.2	29,300	13,200	0.45
11/5	Humid climate simulation	90.1	80.6	921	75.3	63.6	760	199.1	3290	62.6	62.0	29,900	3,800	0.13
11/6	Humid climate simulation	90.3	80.8	918	80.5	66.9	756	199.5	3290	64.6	63.8	29,300	8,066	0.28
11/7	Humid climate simulation	90.5	80.4	931	80.7	67.0	758	174	3290	65.3	64.7	23,700	5,958	0.25
11/8	Full-flow calibration	95.1	75.1	910	80.6	67.5	759	198.8	3290	61.5	60.6	29,300	17,900	0.61

TABLE 3-3

COMPARISON OF FULL-FLOW DESIGN AND
OFF-DESIGN DATA CORRECTED TO ARI CONDITIONS

	Run No.	Capacity, Btu/hr	COP
Design point test runs	5/7	16,300	0.60
	5/8	15,600	0.57
	7/5	16,400	0.59
	9/1	16,100	0.60
	10/10	16,500	0.58
Simulated climate tests	11/2	16,000	0.59
	11/3	17,200	0.62
	11/4	16,900	0.60
	11/5	14,900	0.55
	11/6	14,800	0.51
	11/7	15,200	0.45

TABLE 3-4

FULL-FLOW CALIBRATION RUNS

Run No.	Heater Bypass Capacity		
	Flow Area, sq in.	Flow Area, Btu/hr	COP
5/7	24	16,300	0.60
5/8	24	15,600	0.57
7/5	24	16,400	0.59
9/1	24	16,100	0.60
10/10	24	16,500	0.58
11/8	21	17,700	0.59
12/1	21	16,300	0.61
14/2	21	16,800	0.61

TABLE 3-5

HALF-FLOW CALIBRATION RUNS

Run No.	Heater Bypass Flow Area, sq in.	Capacity Btu/hr	COP
9/2	24	5300	0.44
10/1	24	5700	0.43
11/1	24	6400	0.51
12/2	21	7400	0.56
12/3	21	7800	0.50

Half-Flow Performance at ARI Condition

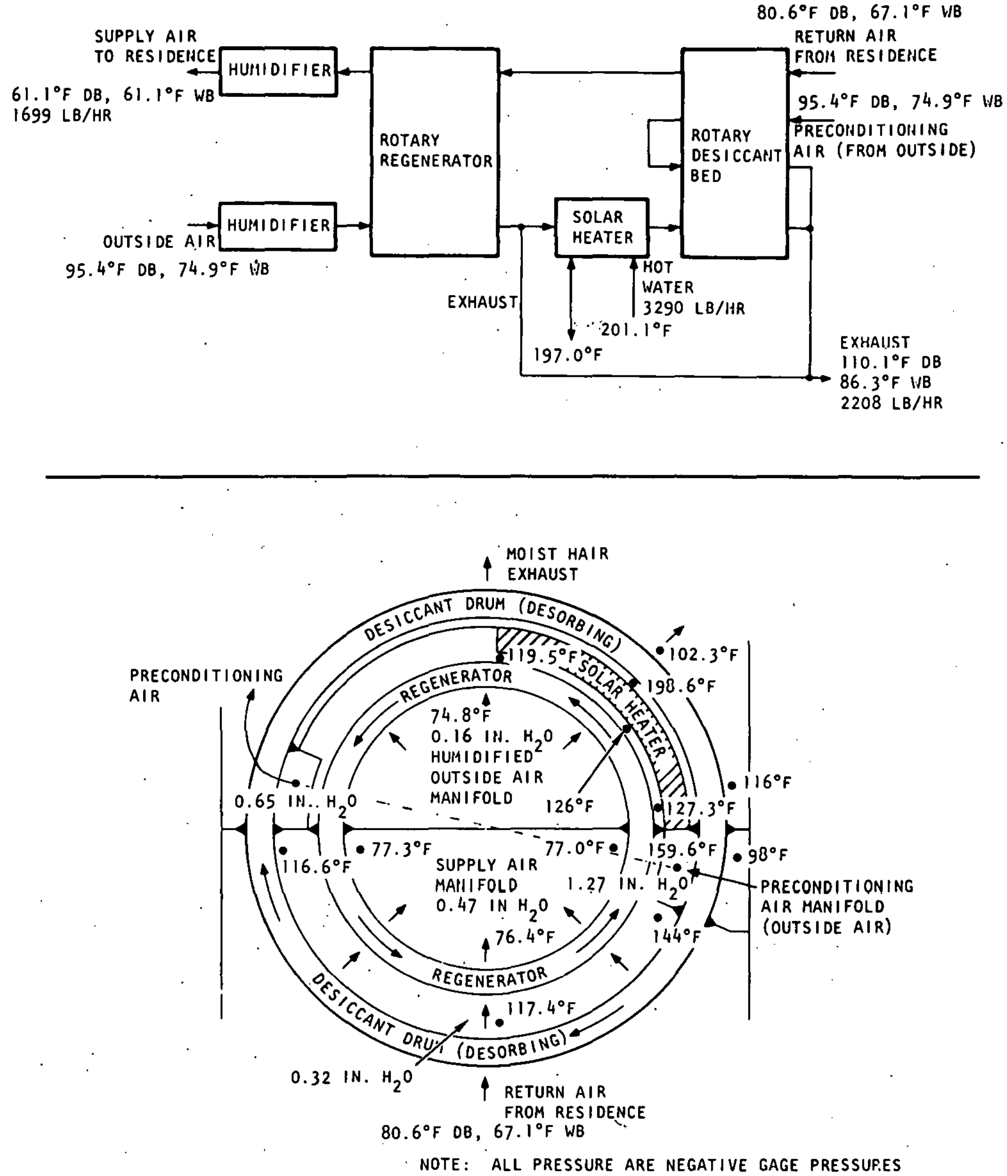
Typical half-flow performance data are shown in Figure 3-7 for interfacing conditions near ARI standard temperatures, and for a heat source temperature of about 200°F. Table 3-6 summarizes the half-flow test data. Detailed test data were supplied to SERI informally.

Figures 3-8 through 3-10 show the effect of the interfacing parameters on system capacity and COP. For these plots, all parameters were corrected to the exact value of the ARI standards with the exception of the parameter under investigation. Also shown on the plots are the computer prediction at half-flow.

The COP obtained on test is higher than predicted; this is also true for the full-flow tests. The capacity is generally lower than predicted. This could be due to heat and air leakage in the recirculated process airstream. Heat leaks at reduced flow rates would have a more pronounced effect than at full flow.

The scope of test data plots is about the same as predicted, with the exception of the effect of outdoor wet bulb temperature (see Figure 3-8), which is more pronounced than anticipated. This behavior is indicative of possible leakage from the outdoor process airstream into the recirculated airstream. Redesign of the drums and seals would be necessary to reduce leakage further than that achieved by incorporation of the partitions in the drums and by modification of the original seals (see Reference 4).

Generally, the data obtained are believed to be in good agreement with the predicted data, with the reservations noted above.



Q SOLAR = 13,490 BTU/HR
CAPACITY AT TEST CONDITIONS: 7,640 BTU/HR
COP AT TEST CONDITIONS: 0.57
CAPACITY AT STANDARD CONDITIONS: 7,390 BTU/HR
COP AT STANDARD CONDITIONS: 0.56

A-13784

Figure 3-7. Typical Half-Flow Performance at ARI Conditions

TABLE 3-6

HALF-FLOW PERFORMANCE SUMMARY
RECIRCULATED MODE

Run No.	Purpose	Test Conditions								Test Results						
		Outdoor			Indoor			Water		Indoor Outlet				Corrected Data		
		Inlet Dry Bulb, °F	Inlet Wet Bulb, °F	Flow scfm Total	Inlet Dry Bulb, °F	Inlet Wet Bulb, °F	Flow scfm	Inlet Temp, °F	Flow lb/hr	Dry Bulb, °F	Wet Bulb, °F	Solar Q, Btu/hr	Capacity, Btu/hr	COP	Capacity Btu/hr	COP
12/1	Calibration full flow	95.2	75.0	917	80.3	66.9	761	199.2	3290	61.8	60.8	26,978	16,216	0.601	16,334	0.605
12/2	Half-flow standard conditions	95.4	74.9	491	80.6	67.1	378	201.1	3290	61.1	61.1	13,489	7641	0.566	7388	0.556
12/3	Effect of water temperature	95.5	74.8	489	79.9	67.0	373	174	3290	62.8	62.8	11,844	5410	0.457	5317	0.452
12/4	Effect of water temperature	95.5	74.5	489	80.1	67.2	373	151	3290	63.9	63.9	7567	4271	0.564	3863	0.540
12/5	Effect of outdoor wet bulb	95.3	71.5	485	80.0	67.5	376	199.8	3290	59.5	59.5	13,160	10,233	0.778	9874	0.754
12/6	Effect of outdoor wet bulb	95.0	79.4	496	79.8	67.0	379	199.5	3290	64.3	64.3	13,489	3444	0.255	3498	0.256
12/7	Effect of outdoor dry bulb	85.5	75.8	500	79.6	66.9	383	199.6	3290	62.3	62.3	13,160	5910	0.449	6295	0.470
12/8	Effect of outdoor dry bulb	105.1	75.9	492	80.5	67.0	382	200.6	3290	61.4	61.4	13,490	7099	0.526	7467	0.550
12/9	Effect of indoor wet bulb	95.5	75.2	489	80.8	63.8	376	200.6	3290	59.6	59.6	13,818	4947	0.358	4944	0.360
12/10	Effect of indoor wet bulb	95.0	75.5	490	80.4	71.4	377	201	3290	64.8	64.8	12,502	9237	0.739	9380	0.751
13/1	Effect of outdoor wet bulb (repeat 12/6)	95.0	80.3	489	80.0	67.0	374	200.4	3290	65.1	65.1	13,160	2389	0.182	2357	0.181
13/2	Effect of outdoor dry bulb (repeat 12/7)	85.3	74.9	496	80.2	67.0	379	201.8	3290	61.5	61.5	13,818	7041	0.510	6834	0.505
13/3	Half-flow standard conditions	94.9	74.7	489	80.6	67.2	380	201.0	3290	60.8	60.8	13,160	8218	0.624	7797	0.603
13/4	Effect of indoor dry bulb	95.1	75.1	487	73.5	67.4	379	201.2	3290	62.3	62.3	12,831	6725	0.524	6379	0.502
13/5	Effect of indoor dry bulb	95.1	75.1	482	85.5	67.8	380	200.8	3290	61.0	61.0	12,502	8811	0.705	8197	0.67
13/6	Dry-hot climate simulation	106.6	73.7	478	80.4	66.8	373	175	3290	60.8	60.8	9870	7636	0.774	-	-
13/7	Dry-hot climate simulation	107.7	74.3	472	80.0	66.6	376	199.9	3290	60.0	60.0	12,173	8400	0.690	-	-
13/8	Dry-hot climate simulation	107.2	74.0	478	75.4	63.5	377	200.4	3290	58.6	58.6	12,831	5923	0.462	-	-
13/9	Humid climate simulation	89.9	79.5	480	79.8	67.5	371	201.1	3290	64.7	64.7	12,831	3652	0.285	-	-
13/10	Humid climate simulation	90.2	80.3	491	80.2	67.5	371	174.6	3290	66.4	66.4	10,528	1352	0.128	-	-
14/1	Humid climate simulation	90.4	80.2	490	75.3	63.2	377	200.2	3290	63.6	63.6	13,489	665	0.049	-	-
14/2	Calibration full flow	95.2	75.7	913	80.5	66.8	758	198.6	3290	61.3	60.7	26,978	15,697	0.581	16,784	0.606
14/3	Effect of water - temperature (Repeat 12/4)	95.7	74.5	496	80.1	67.2	373	152.3	3290	64.6	64.6	8,225	3348.	0.407	-	-
14/4	Dry-hot climate simulation (repeat 13/6)	107.9	73.3	490	80.3	67.1	377	174.2	3290	62.1	62.1	10,199	6478	0.635	-	-
14/5	Dry-hot climate simulation (repeat 13/7)	107.5	71.7	483	80.3	67.3	377	200.8	3290	60.1	60.1	13,160	9263	0.704	-	-
14/6	Dry-hot climate simulation (repeat 13/8)	107.0	71.2	484	75.8	63.0	375	200.5	3290	57.9	57.9	13,160	6095	0.463	-	-

*Repeated runs
**Data corrected to ARI conditions except for the parameter investigated

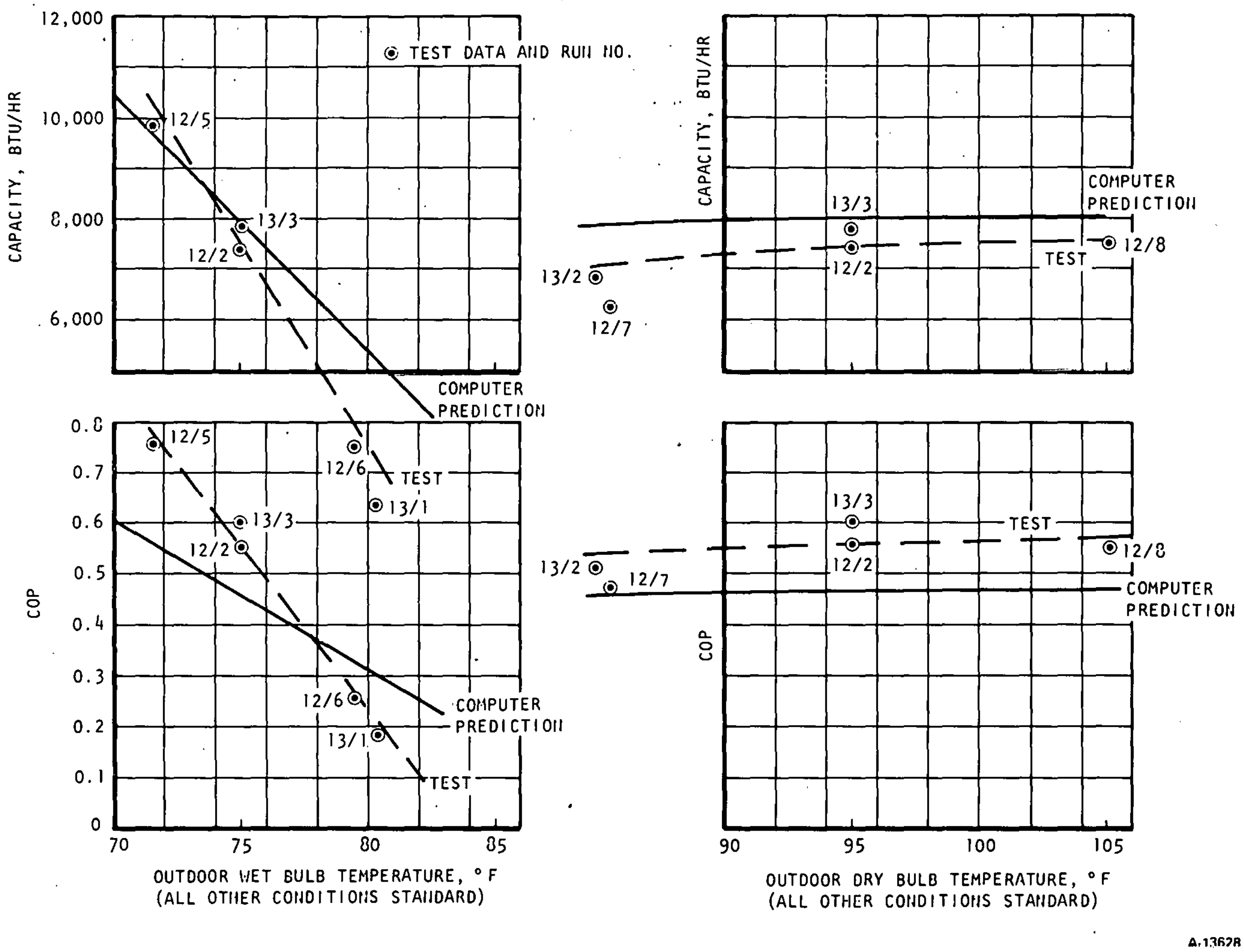


Figure 3-8. Half-Flow Test Performance--Effect of Outdoor Temperatures

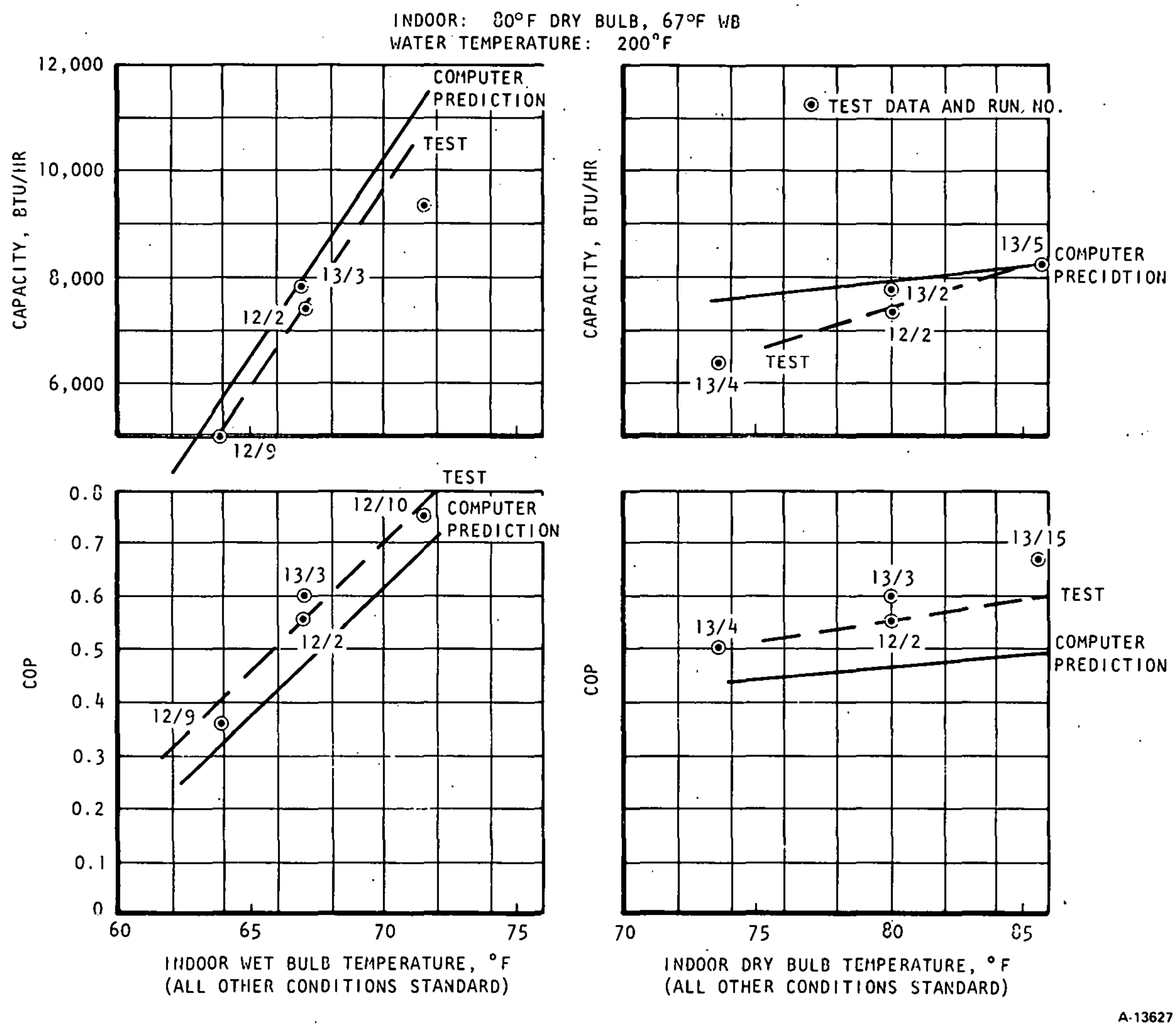


Figure 3-9. Half-Flow Test Performance--Effect of Indoor Temperatures

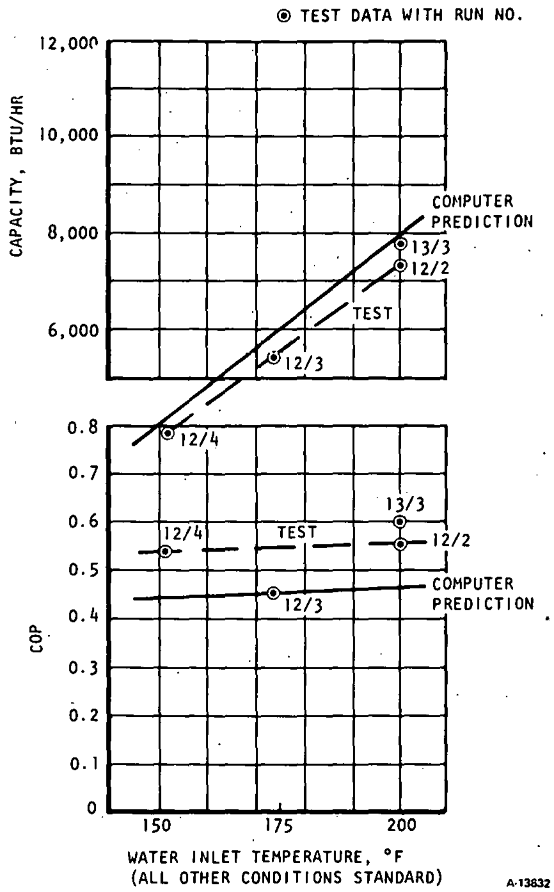


Figure 3-10. Half-Flow Test Performance--Effect of Water Temperature