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Phase II
Second Technical Progress Report

DEVELOPMENT OF A SOLAR-DESICCANT DEHUMIDIFIER

81-18436

October 16, 1981

AC03-77CS31591

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

GARRETT

AIRESEARCH MANUFACTURING COMPANY

DISCLAIMER

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Prepared by

J. Rousseau

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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 is part of the Phase II effort and is concerned with off-design testing in the ventilated 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 March 1, 1981 through October 1, 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. Dennis Schlepp 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 presented in the first Phase II technical progress report.

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 concerned with (1) configuration development, (2) design point optimization, and (3) off-design performance characterization.

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) over the entire range of interfacing parameters.

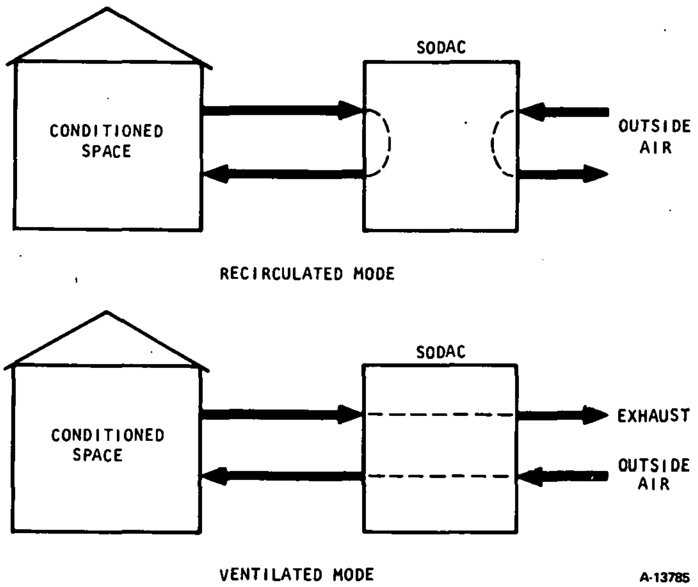

Figure 1-1. Recirculated and Ventilated Mode Flow Paths

This report presents the test data in the ventilated mode at design flow rates. Half-flow testing in this configuration was not performed because of the low system performance at full-flow.

To facilitate comprehension, the description of the system and its operation is presented in Section 2. This description has already been included in previous reports; readers familiar with the SODAC are directed to Section 3 for detailed test data.

CONCEPT

The SODAC schematic is shown in Figure 1-2 in the ventilated mode. 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.

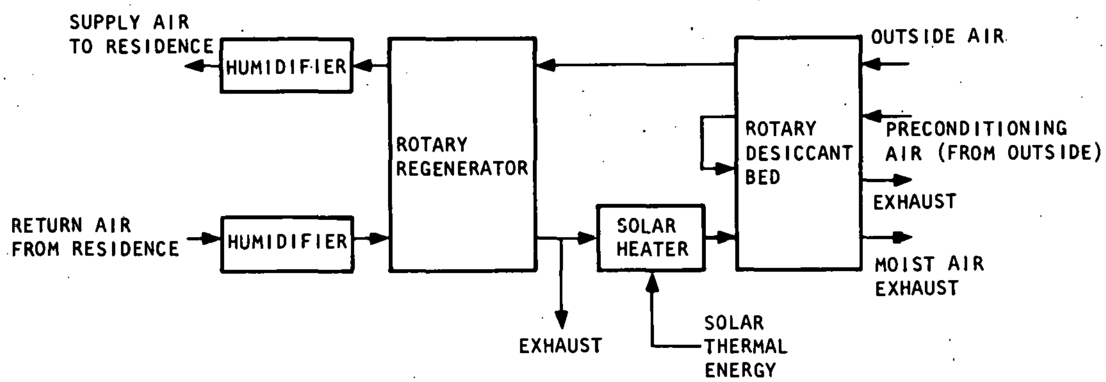

Figure 1-2. System Schematic

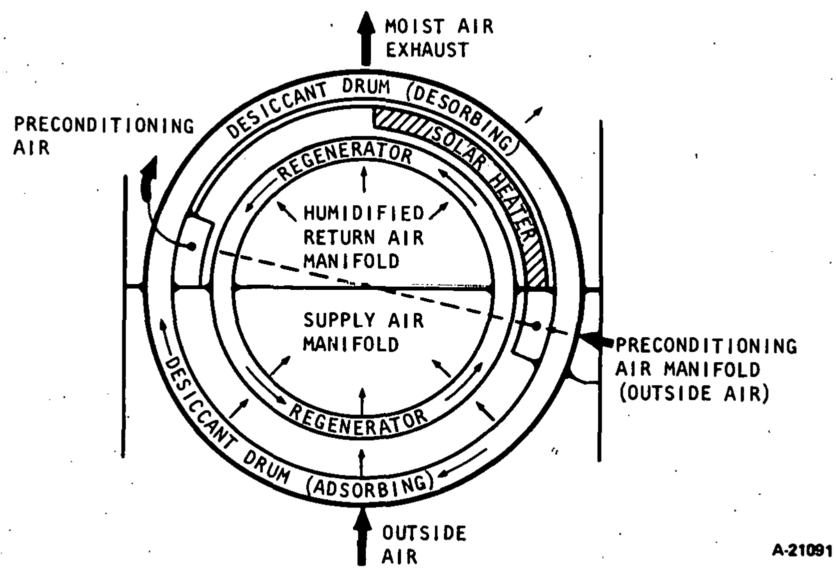

Figure 1-3. Dehumidifier Cross Section

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 Phase II test program 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. All test data obtained in Phase I and Phase II are included in the first Phase II technical progress report, AiResearch Report 81-17773. In Phase II, recirculated mode testing at full flow and half flow was completed. This second technical progress report contains test data in the ventilated mode. Design point performance defined by the ARI standard conditions and a 200°F solar energy (water) temperature level are listed in Table 1-1.

TABLE 1-1

DESIGN POINT PERFORMANCE
VENTILATED 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
Capacity, Btu/hr	11,500
Coefficient of performance	0.47
Conditioned space air flow, scfm	760
Outdoor air flow, scfm	925
Water usage, gal/hr	3.9
Parasitic power (fans, pumps, and and drive), kw	0.8
PREDICTED DATA	
Capacity, Btu/hr	13,500
Coefficient of performance	0.47
Parasitic power, kw	0.75

The experimental data obtained is consistent with computer predictions; the actual COP is about the same as predicted, and the capacity is within 15 percent of prediction.

Off-design performance in terms of capacity and COP is 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. A comparison of the experimental data in the ventilated and recirculated modes confirms the conclusions reached as a result of computer analyses--the SODAC performance in the ventilated configuration is considerably lower than in the recirculated mode. Consequently, testing of the SODAC at half-flow in the ventilated mode was abandoned after discussions with the contract technical monitor.

FUTURE WORK

A computer analysis of the SODAC indicates that significant performance advantages could be obtained by extending the silica gel adsorption zone. This can be done by repositioning the seals identified as "A" in Figure 1-7. Limitations imposed by the actual hardware will permit a maximum increase of 22 percent (from 158 deg arc to 193 deg arc).

The estimated SODAC capacity and COP in terms of the new seal location are plotted in Figure 1-8. At the 35-deg limit, the anticipated increase in capacity and COP are 31 and 12 percent, respectively.

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).

5. Rousseau, J., Development of a Solar Desiccant Dehumidifier, Phase II Technical Progress Report, AiResearch Report 81-17773, March 27, 1981 (to be published by DOE).

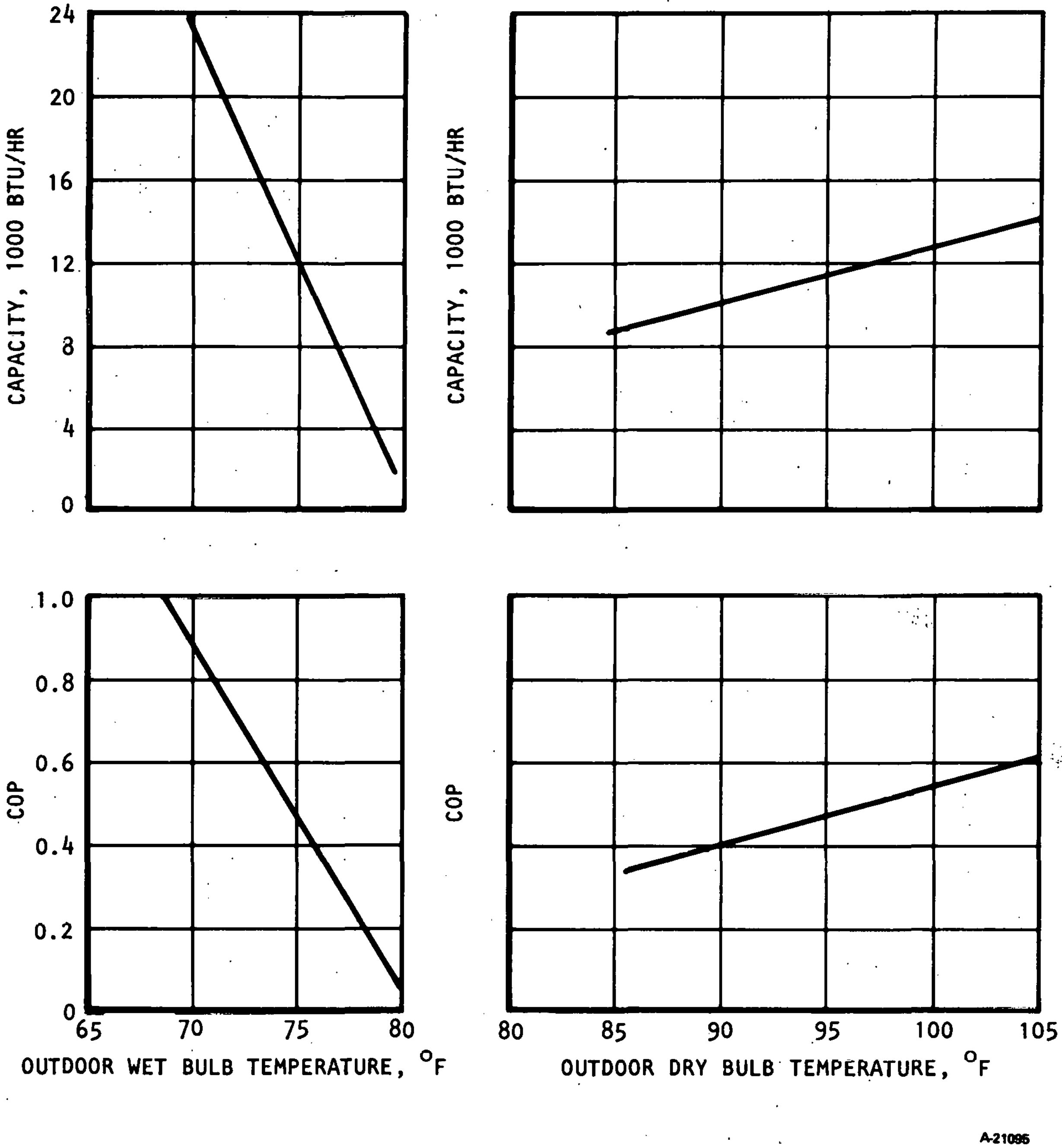

Figure 1-4. SODAC Ventilated Mode Performance--
Effect of Outdoor Temperatures

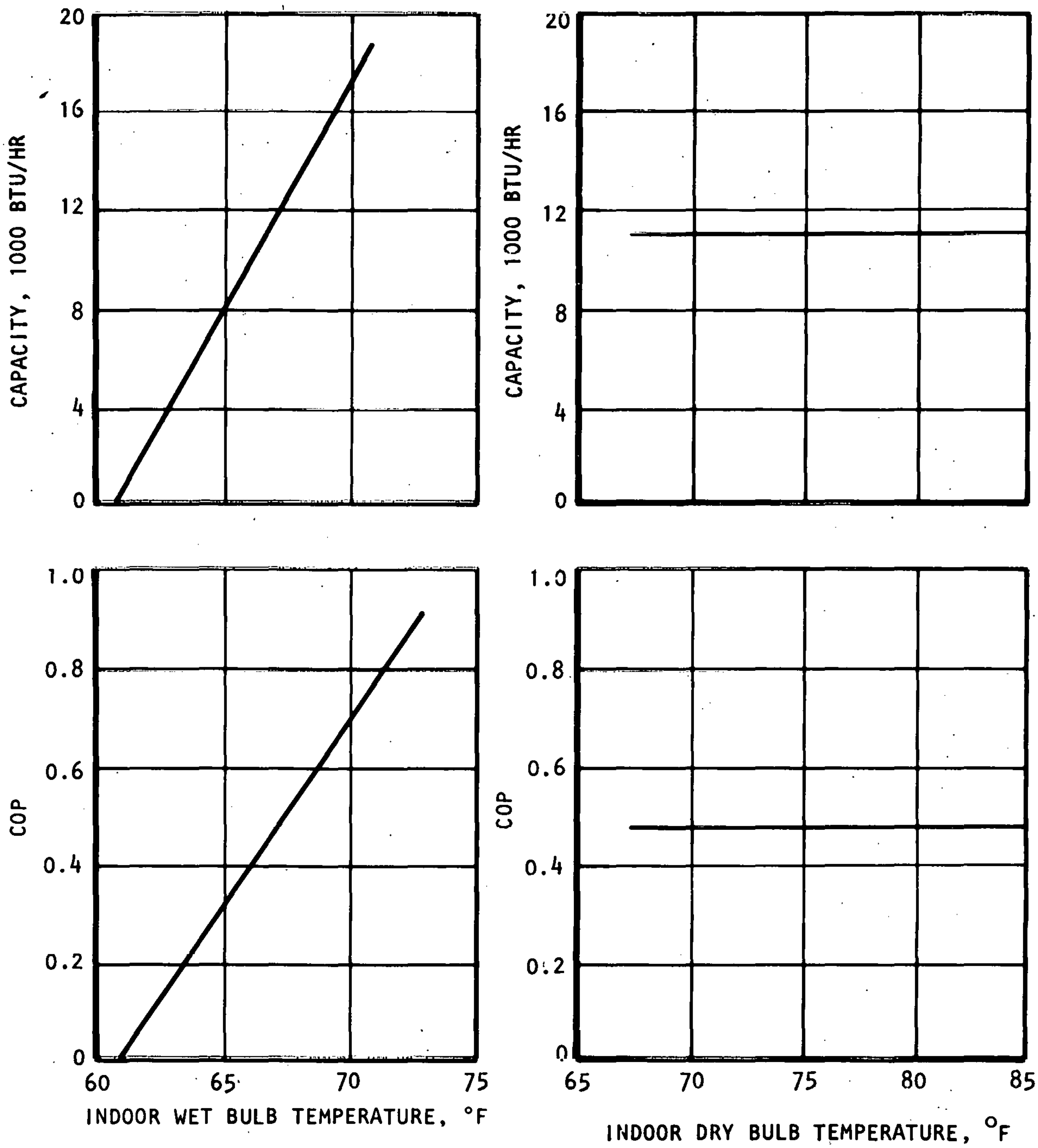

Figure 1-5. SODAC Ventilated Mode Performance--
Effect of Indoor Temperatures

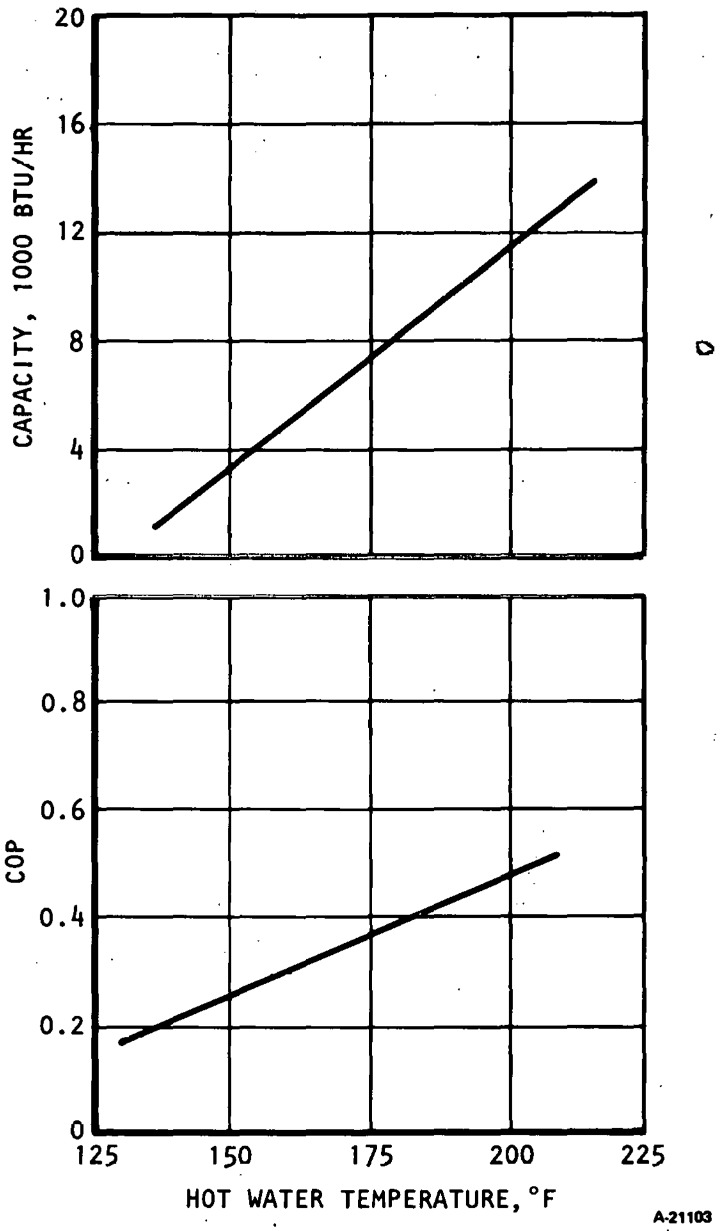

Figure 1-6. SODAC Ventilated Mode Performance--
Effect of Water Temperatures

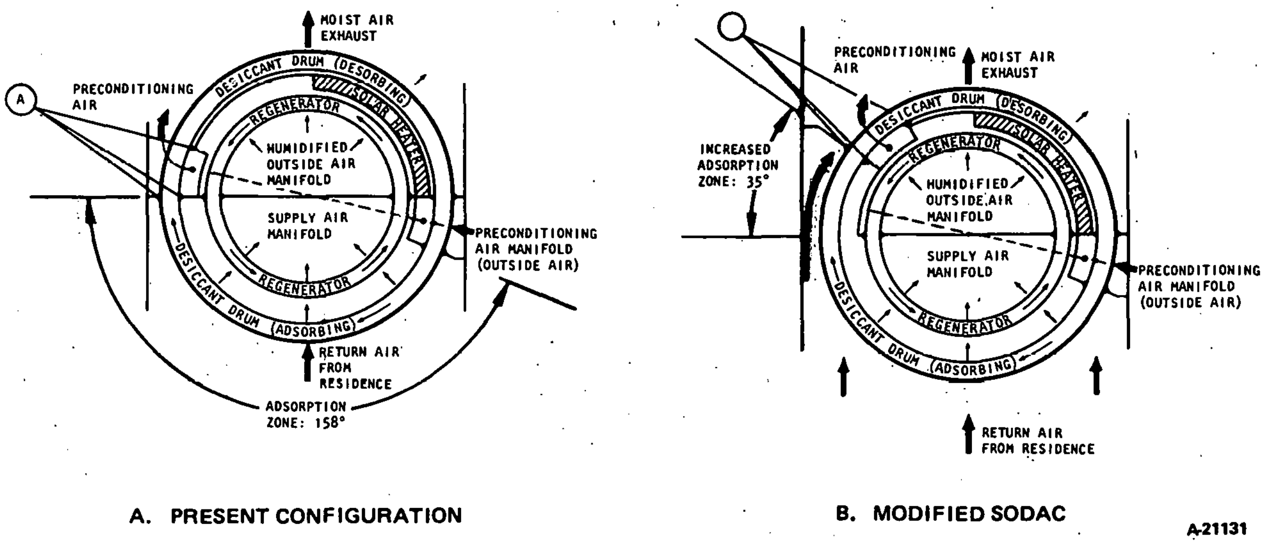

Figure 1-7. Seal Relocation

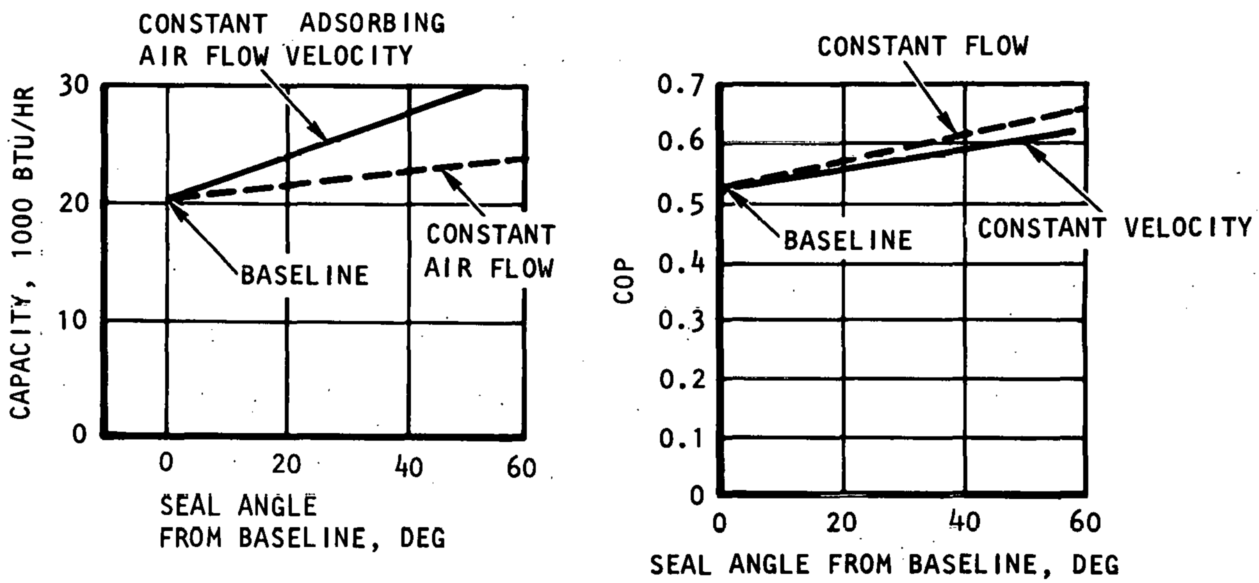

Figure 1-8. Estimated Performance Improvement

SECTION 2

BASELINE 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. All data are for the baseline recirculated mode. Data for the ventilated mode configuration are at the end of this section.

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.

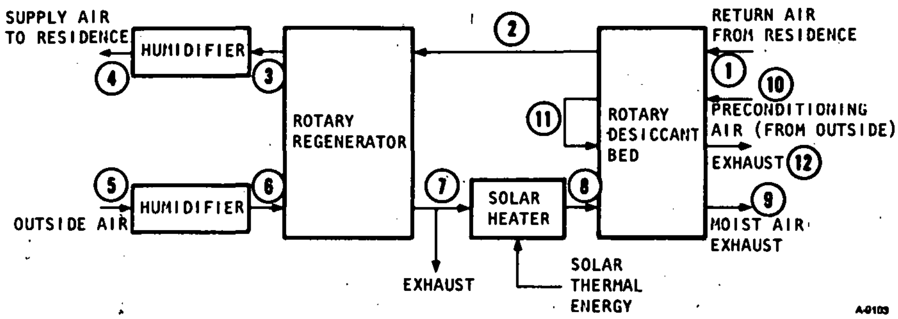

Figure 2-1. Air Conditioner Schematic

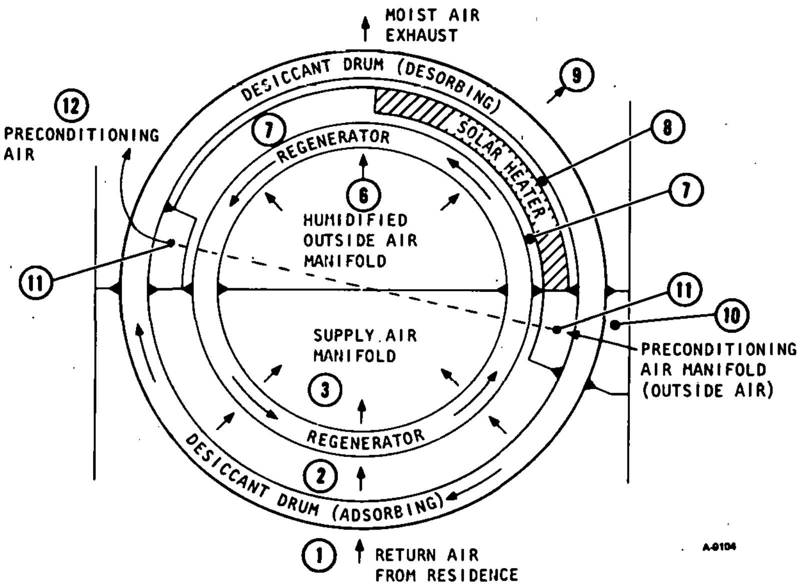

Figure 2-2. Dehumidifier Arrangement

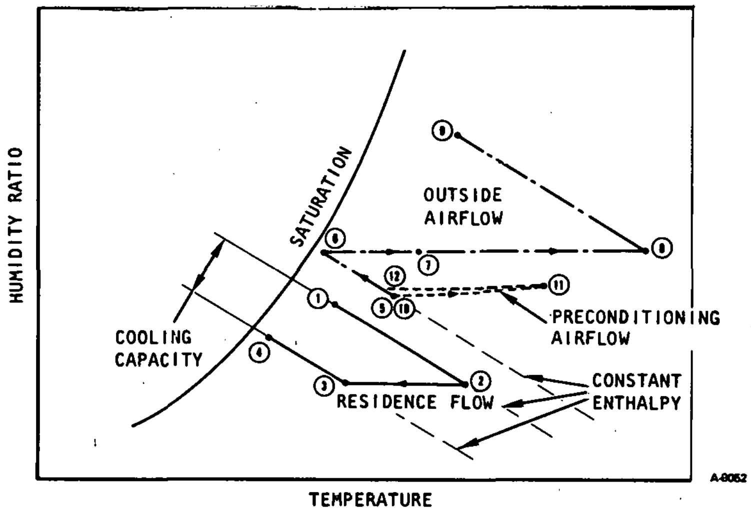

Figure 2-3. Psychrometric Process

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.

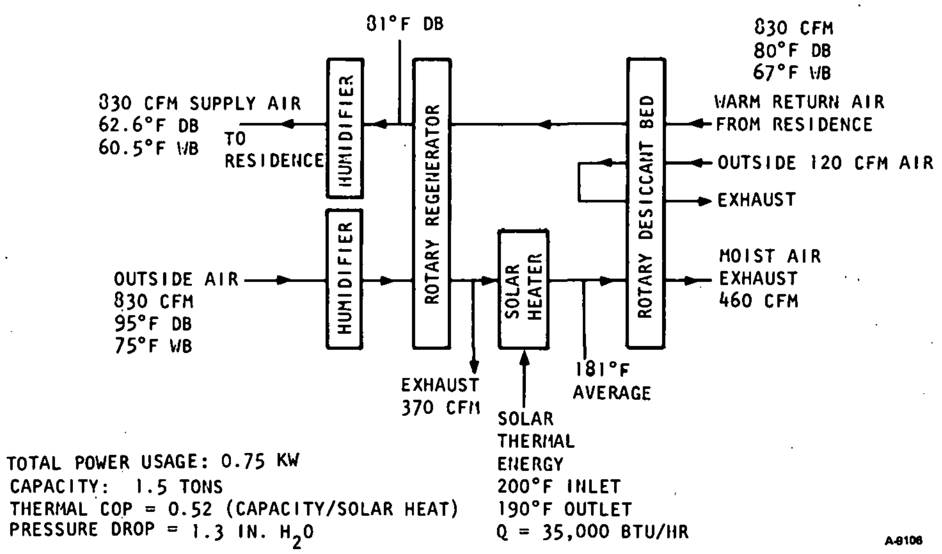

Figure 2-4. Baseline SODAC Performance

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.

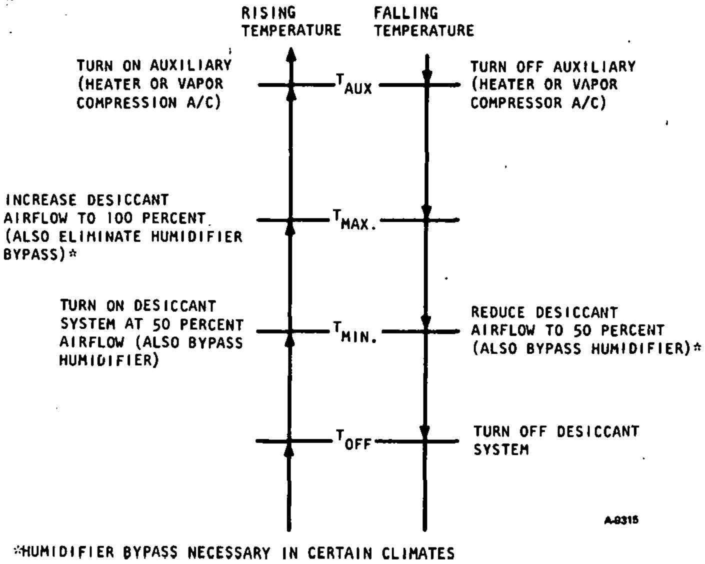

Figure 2-5. Control Scheme

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.

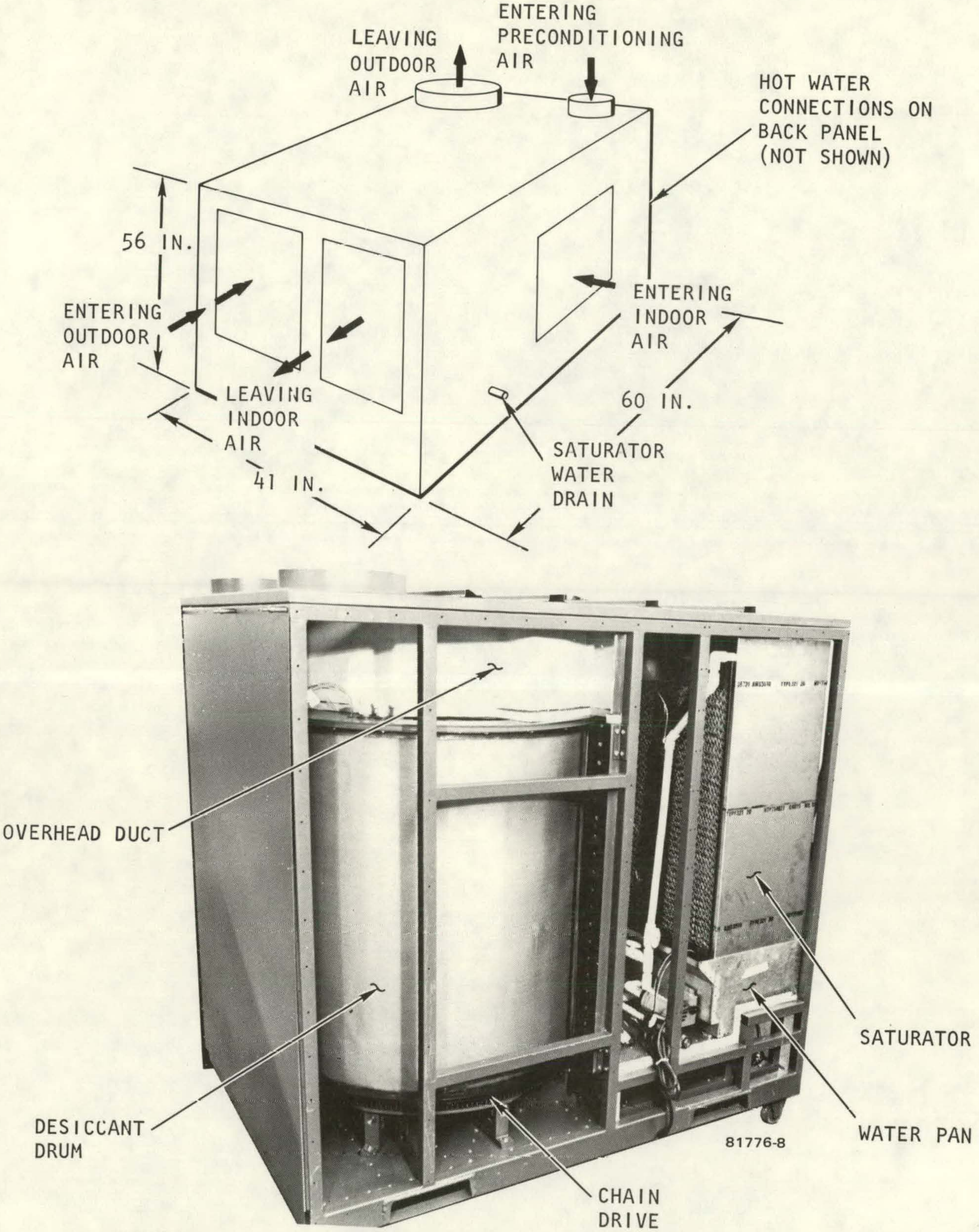

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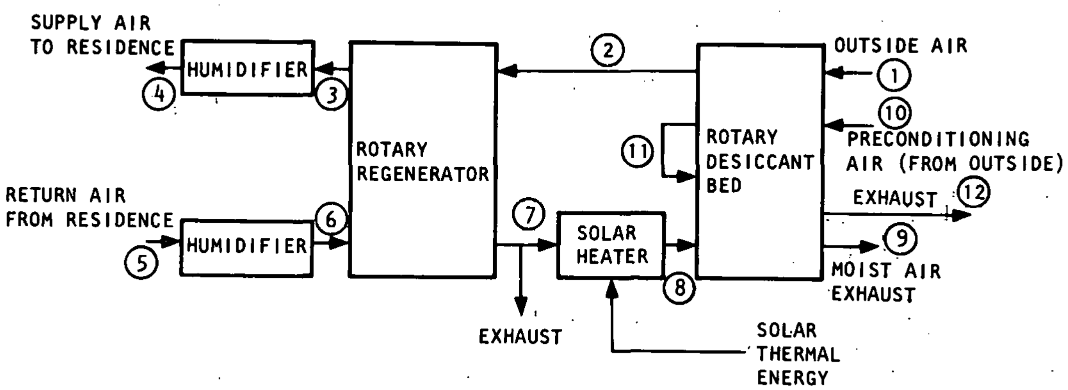
Figure 2-6. SODAC Package

VENTILATED MODE SODAC

In the ventilated mode configuration fresh air is processed through the SODAC and supplied to the conditioned space. Return air from the conditioned space is humidified and used to regenerate the rotary heat exchanger and the sorbent bed. The flow paths of the two airstreams through the unit are illustrated in Figure 2-7.



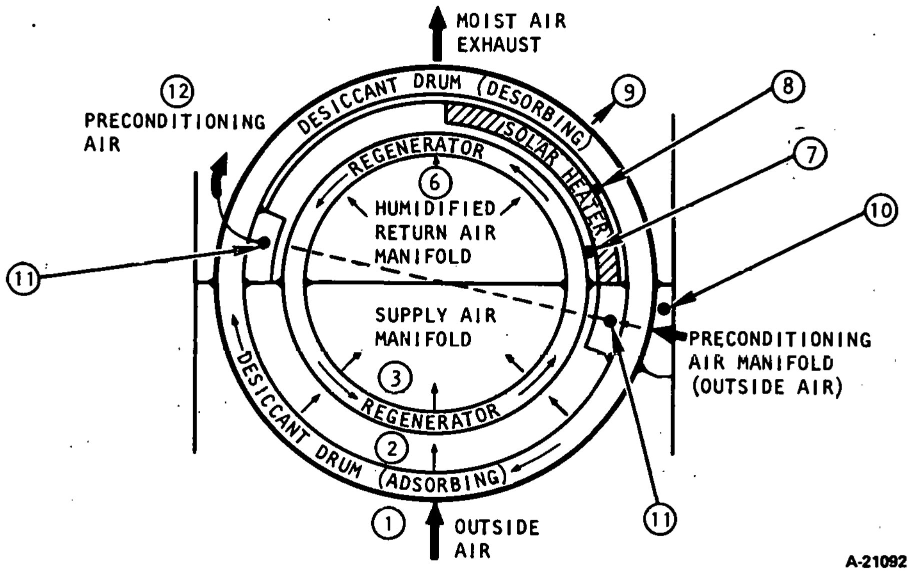


Figure 2-7. Ventilated Mode Configuration

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. Descriptions of the test facilities and equipment, system instrumentation, and test procedures are available in AiResearch Report 81-17773 (Reference 5).

TEST PROGRAM

Testing in the recirculated mode was completed earlier in the program and is also described in Reference 5. This report covers testing in the ventilated mode configuration at full flow. A total of 35 test runs were conducted.

TEST DATA

Table 3-1 summarizes the significant test data. Typical data at near ARI conditions are presented in Figure 3-1. Off-design performance is presented in Figures 3-2 through 3-4; in these plots, capacity and COP are corrected to ARI conditions with the exception of the parameter used in abscissa. The slope of the experimental curves were used to derive the correction factors in times of all interfacing conditions.

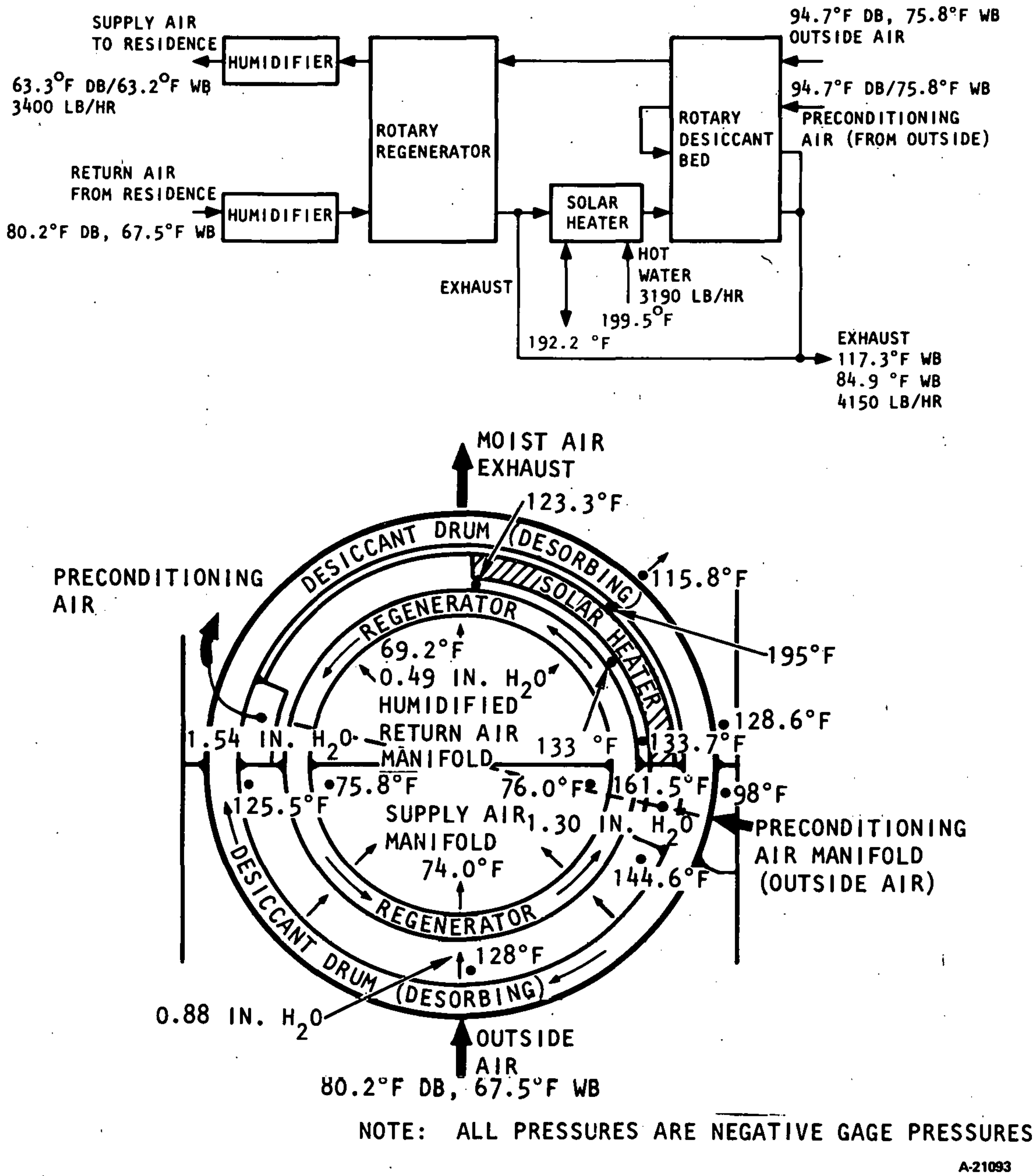
COMPARISON OF RECIRCULATED AND VENTILATED MODE PERFORMANCE

A comparison of the SODAC performance in the recirculated and ventilated modes configurations is shown in Figure 3-5. The data cover the entire range of interfacing parameters investigated and show no advantage for the ventilated mode version of the SODAC except at relatively low outdoor wet bulb temperature (lower than 72°F). This confirms computer predictions.

TABLE 3-1

VENTILATED MODE PERFORMANCE SUMMARY

Run No.	Purpose	Test Conditions							Test Results						
		Outdoor			Indoor				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	Water Temp, °F	Dry Bulb, °F	Wet Bulb, °F	Solar Q, Btu/hr	Capacity, Btu/hr	COP	Capacity, Btu/hr	COP
14/7	ARI conditions	94.7	75.8	921	80.2	67.5	756	199.5	63.3	63.2	24,017	11,265	0.469	12,020	0.485
14/8	Low outdoor wet bulb	94.8	70.6	928	80.3	67.0	763	199.8	59.0	58.3	25,662	21,974	0.856	22,040	0.858
14/10	Low outdoor dry bulb	86.0	75.5	937	79.9	67.5	758	199.4	64.9	64.1	25,333	8,955	0.353	9,190	0.350
15/1	High outdoor dry bulb	105.0	74.4	902	79.8	66.5	749	199.5	62.7	61.1	22,043	13,854	0.628	13,670	0.628
15/2	Low indoor wet bulb	94.5	74.4	931	80.2	63.9	751	200	62.6	61.6	23,688	5,526	0.233	4,430	0.200
15/3	High indoor wet bulb	95.0	75.7	942	80.4	72.0	757	199.5	66.2	65.3	23,688	19,331	0.816	20,900	0.864
15/4	ARI conditions	94.6	74.5	944	80.0	66.8	762	199.6	63.2	62.2	23,668	11,950	0.504	11,290	0.488
15/5	Low indoor dry bulb	95.1	75.3	944	70.5	66.9	768	199.5	63.8	63.0	23,030	10,533	0.457	11,420	0.479
15/6	High indoor dry bulb	94.8	75.2	944	85.0	67.7	767	199.8	64.3	63.2	23,359	11,919	0.510	11,180	0.481
15/7	Dry climate simulation	106.1	74.0	928	79.8	66.6	764	199.6	61.6	60.5	21,385	15,651	0.732	-	-
15/8	Dry climate simulation	106.4	75.7	924	80.3	67.0	757	175.1	64.1	63.1	16,450	10,181	0.62	-	-
15/9	Dry climate simulation	106.3	73.8	932	75.4	63.8	775	199.9	60.1	59.0	21,385	11,838	0.554	-	-
16/1	Humid climate simulation	90.5	79.8	945	80.2	67.6	759	174.2	69.9	69.4	18,095	5,574	0.308	-	-
16/2	Humid climate simulation	90.7	80.5	948	80.3	67.5	757	199.8	69.7	69.0	23,688	4,670	0.197	-	-
16/3	Humid climate simulation	90.3	79.7	949	74.7	64.0	767	200.1	67.7	62.2	23,359	4,542	0.194	-	-
17/1	Effect of water temperature	94.4	75.5	931	80.2	67.3	754	151	66.6	66.1	13,818	2,995	0.217	3,685	0.235
17/2	Effect of water temperature	95.3	74.7	936	80.6	67.6	767	174.8	64.7	64.2	19,082	9,023	0.473	7,116	0.404
17/3	ARI conditions	95.4	75.0	926	80.6	66.3	768	199	63.3	63.1	24,346	8,219	0.337	9,576	0.428
17/6	Low water temperature	94.4	75.5	931	80.2	67.3	754	151	66.6	66.1	13,818	2,995	0.217	3,685	0.235
17/7	Low indoor dry bulb	95.5	75.7	920	70.8	67.6	762	199.5	64.1	63.7	24,017	10,622	0.442	10,850	0.436



Q SOLAR = 24,017 BTU/HR
CAPACITY AT TEST CONDITIONS: 11,300 BTU/HR
COP AT TEST CONDITIONS: 0.469
CAPACITY AT STANDARD CONDITIONS: 12,000 BTU/HR
COP AT STANDARD CONDITIONS: 0.480

Figure 3-1. Typical Design Point Test Data (Ventilated Mode)

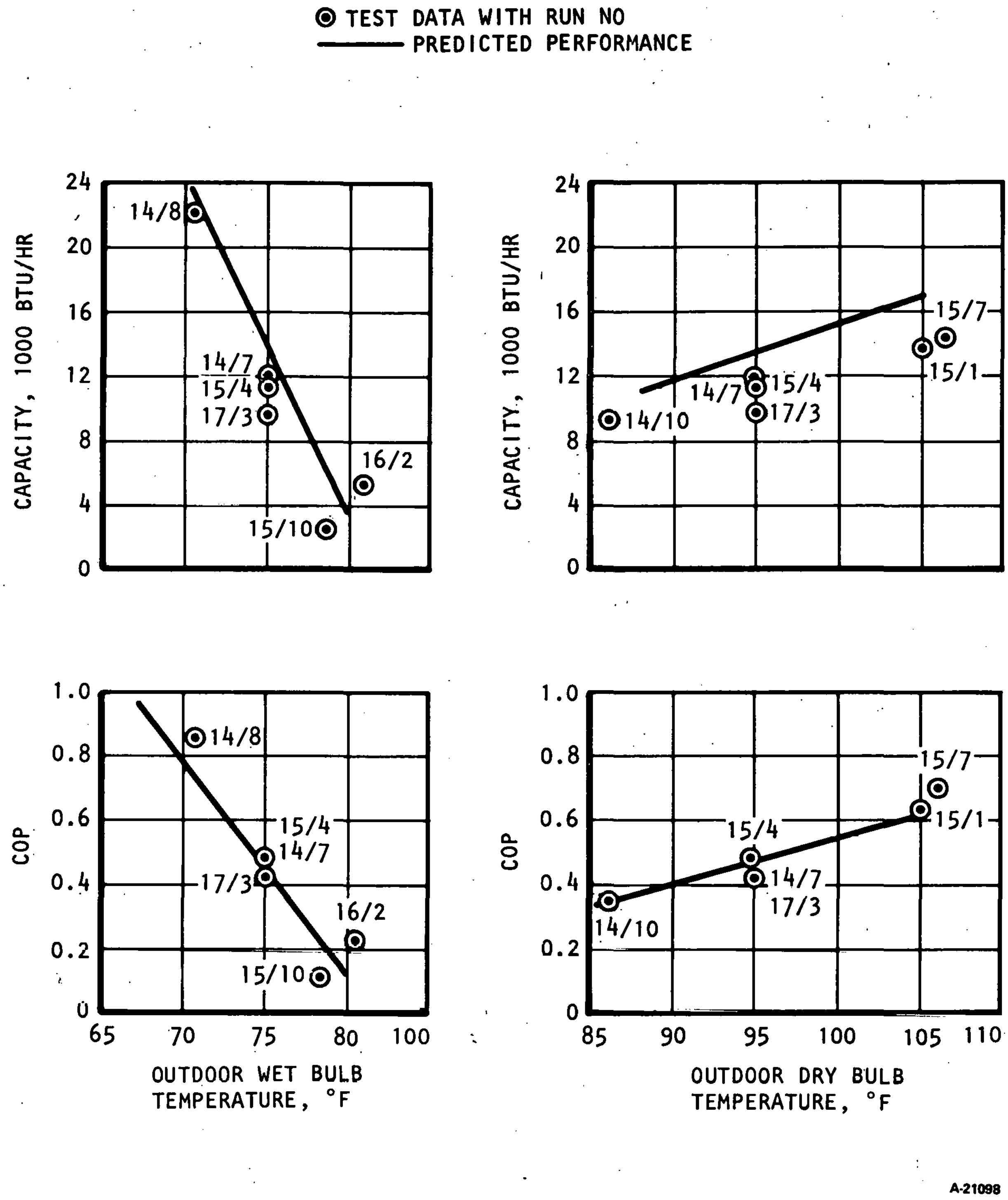


Figure 3-2. Ventilated Mode Test Performance--
Effect of Outdoor Temperature

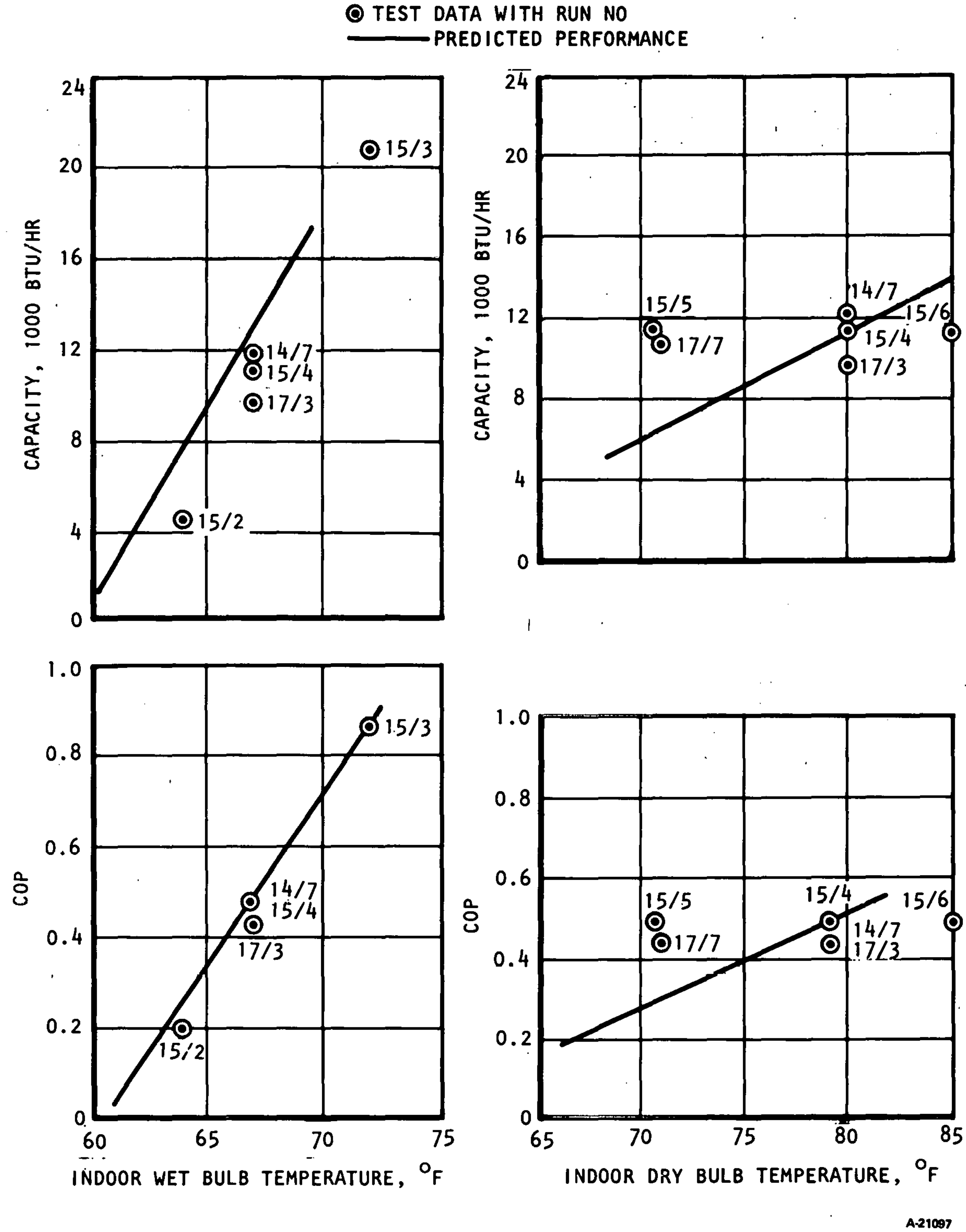


Figure 3-3. Ventilated Mode Test Performance--
Effect of Indoor Temperatures

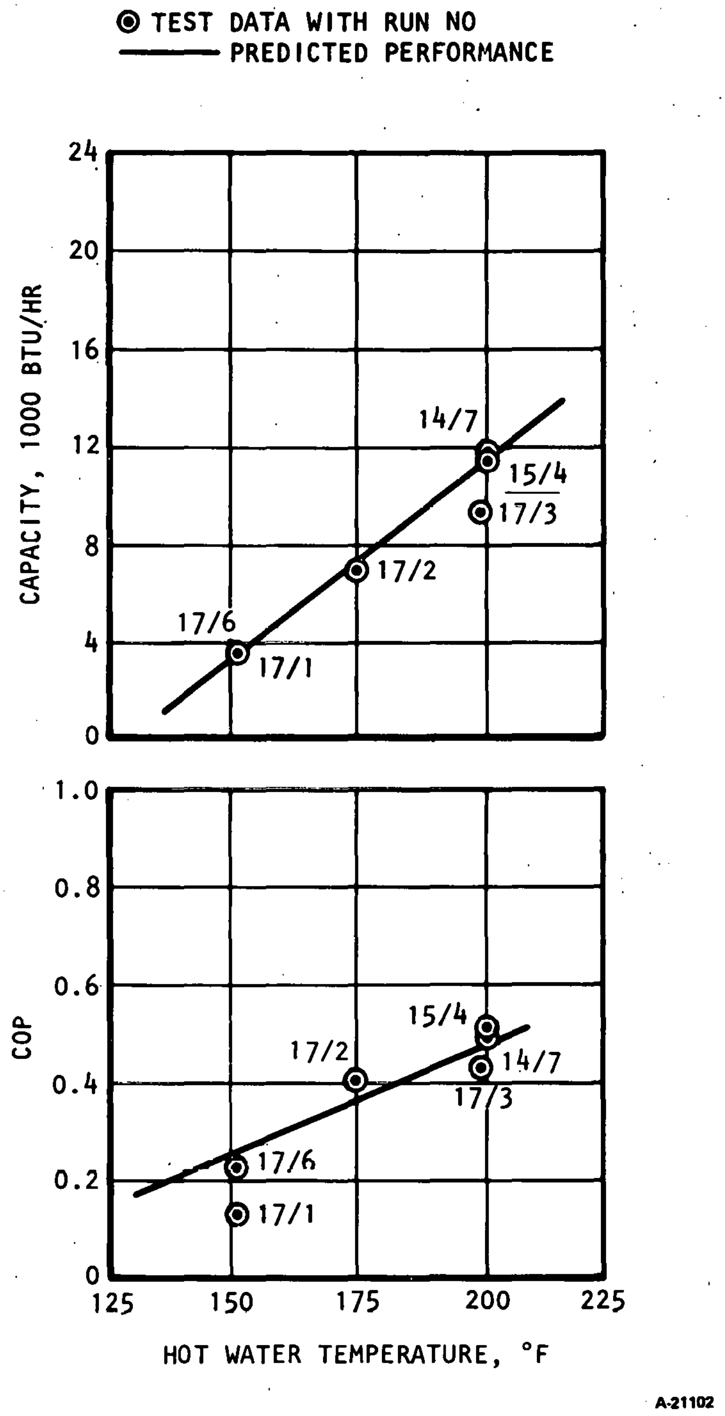


Figure 3-4. Ventilated Mode Test Performance--
Effect of Hot Water Temperature

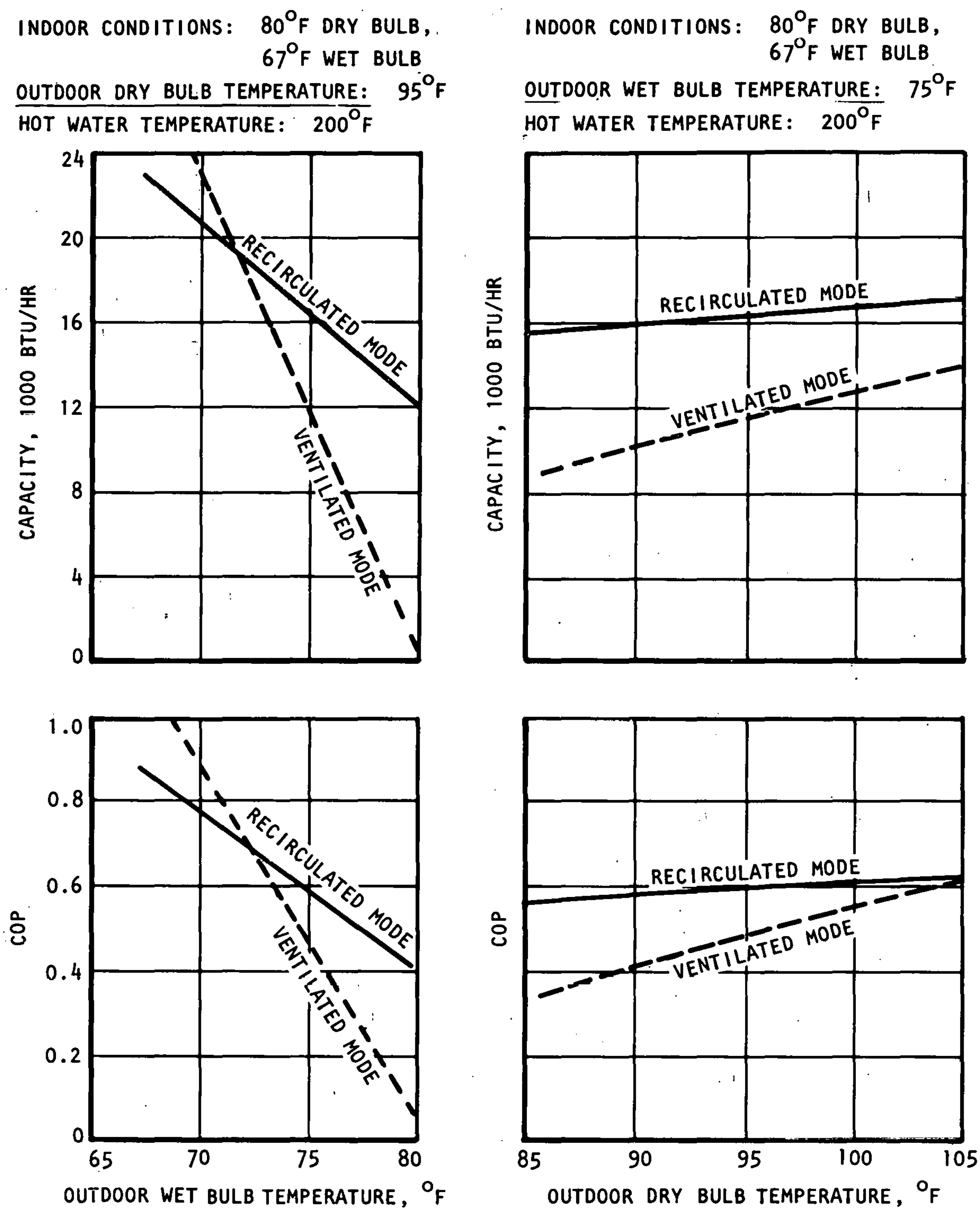


Figure 3-5. Comparison of Recirculated and Ventilated Test Performance

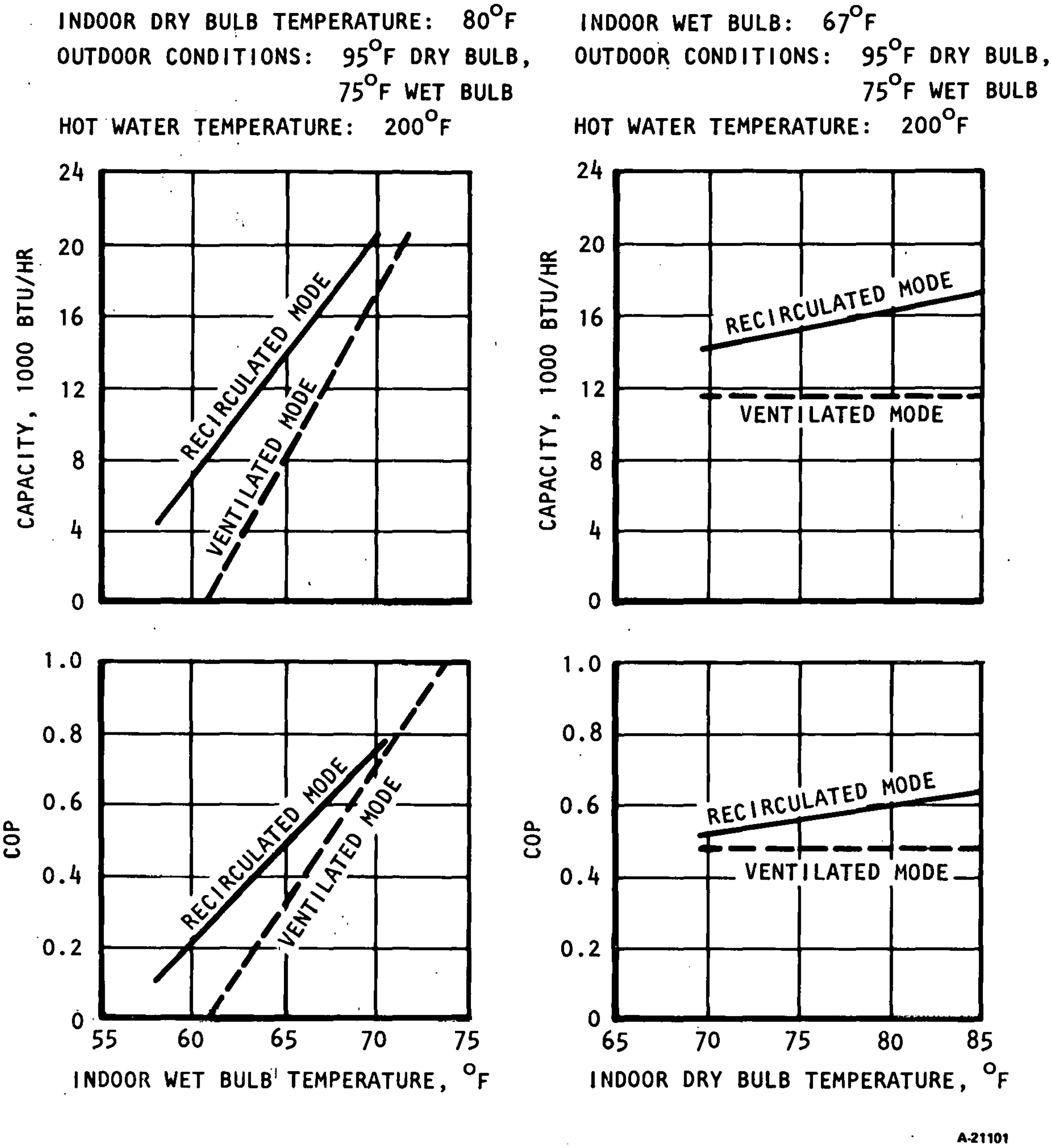


Figure 3-5. Continued

INDOOR CONDITIONS: 80°F DRY BULB, 67°F WET BULB

OUTDOOR CONDITIONS: 95°F DRY BULB, 75°F WET BULB

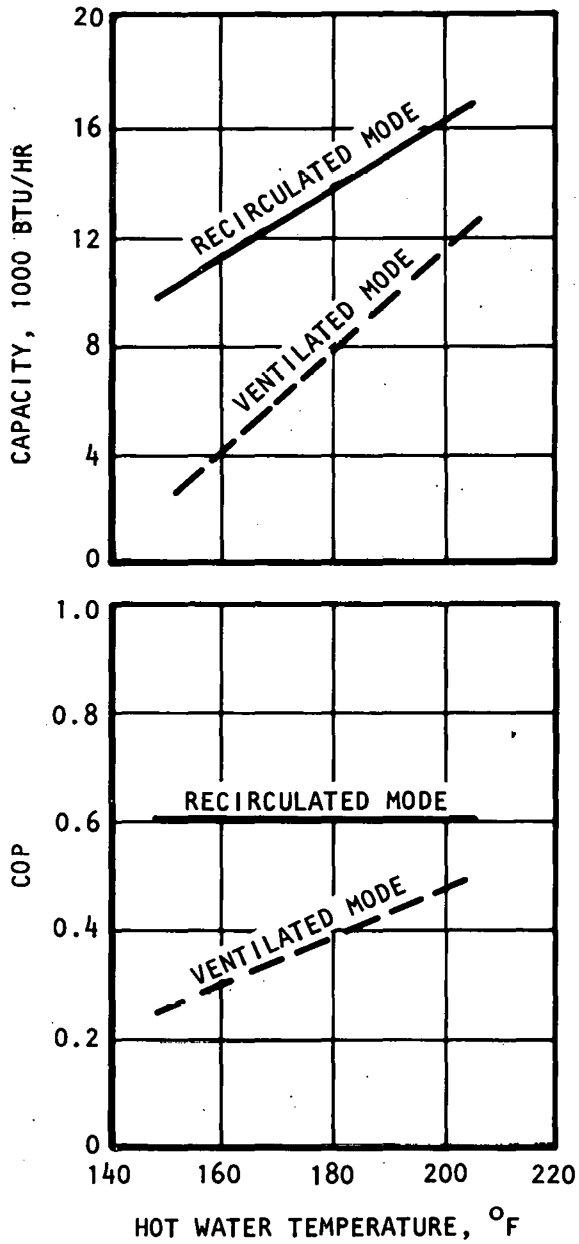


Figure 3-5. Continued