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DEVELOPMENT OF A SOLAR DESICCANT DEHUMIDIFIER

Phase II. Final Summary Report

By
J. Rousseau

November 1, 1982

Work Performed Under Contract No. AC03-77CS31591

Airesearch Manufacturing Company
Torrance, California

U.S. Department of Energy

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GARRETT

AIRESEARCH MANUFACTURING COMPANY

FOREWORD

This 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 second phase of this program and was concerned with off-design testing.

This program was funded by the Department of Energy, San Francisco Operations Office, under contract EG-77-C-03-1591. The period covered by this report is from September 1, 1980 through March 31, 1982. Mr. R. S. Todaro was the DOE contracting officer, and Mr. C. J. Dankowski was the program coordinator. Technical monitoring was the responsibility of Mr. Dennis Schlepp of the Solar Energy Research Institute (SERI).

Mr. J. Rousseau was the program manager and principal investigator for AiResearch, and Mr. J. D. McPherson was the contract administrator. The experimental work on the solar desiccant air conditioner (SODAC) was conducted in the controlled atmosphere chambers of the Dunham-Bush facility in Harrisonburg, Virginia. Mr. P. Rublee was 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 design and test data presented in the Phase I final summary report.

CONTENTS

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 second phase of the program started on September 1, 1980 and was completed in March 1982, except for the publication of this report (which summarizes the Phase II activities).

Phase II is a continuation of the development testing and is concerned with determination of the SODAC performance in the recirculated and ventilated configurations (see Figure 1-1) over the entire range of interfacing parameters.

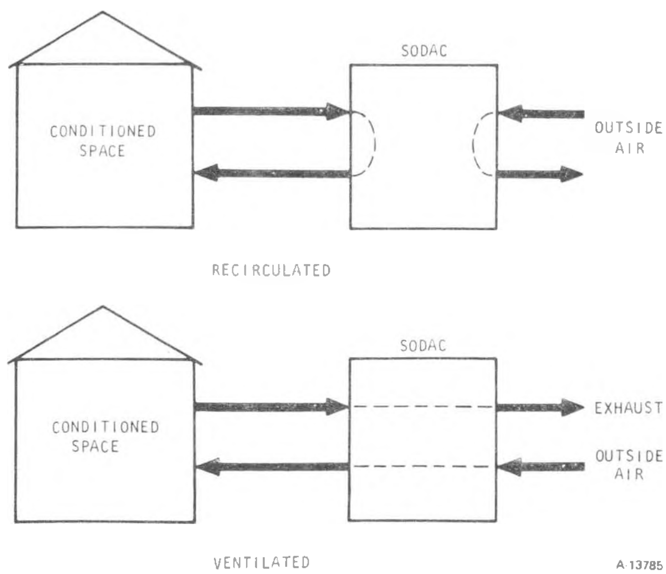

Figure 1-1. Flow Paths for Recirculated and Ventilated Configurations

This report presents the test data for both configurations. Some of the data obtained in Phase I are repeated in this report for completeness. The experimental data are compared to computer predictions.

The system and its operation are described 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 test facility and procedures. Section 4 presents the conclusions and AiResearch recommendations for the SODAC.

Previous documentation under the SODAC program is listed in Section 5 the end of this report (References 1 through 5). References are made to these reports for details on the design, economic evaluation, and construction of the machine.

CONCEPT

The SODAC schematic is shown in Figure 1-2 in the recirculated configuration. The arrangement of the rotary desiccant bed and regenerator is depicted in Figure 1-3. Compared 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.

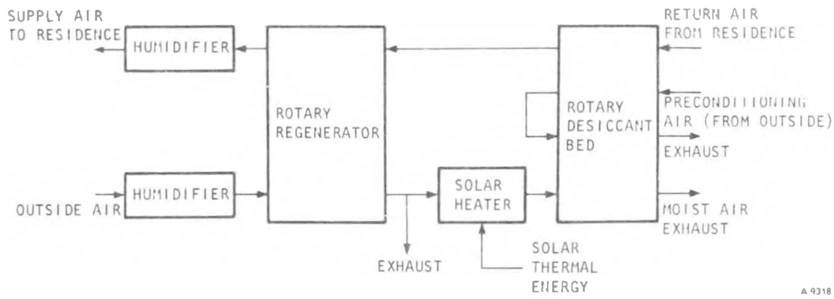

Figure 1-2. System Schematic

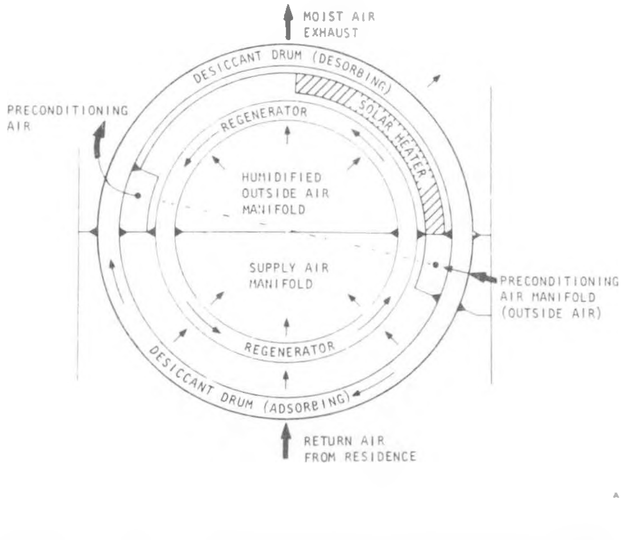

Figure 1-3. Dehumidifier Cross-Section

PHASE II OBJECTIVE

The objective of Phase II was the experimental evaluation of the baseline 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 included the following: (1) two system configurations--recirculated mode and ventilated mode, and (2) operation at design airflow rates and half-flow rates, corresponding to operation at high loads and reduced loads, respectively.

Testing in the ventilated configuration revealed much lower performance than in the recirculated configuration. As a result, this portion of the test program was not completed. The program was redirected toward modifying the SODAC configuration for improved performance.

TECHNICAL ACCOMPLISHMENTS

SODAC testing in the recirculated configuration was initiated in Phase I; in Phase II, 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. Testing in the ventilated configuration was performed at full flow only because of the lower performance obtained over the entire range of operating conditions.

The SODAC design was modified to increase the size of the adsorption zone in an effort to enhance capacity and coefficient of performance. Development in this configuration could not be completed within the present contract.

Design Point Performance

Performance at design point, as defined by the ARI standard rating conditions and a 200°F solar energy (water) temperature level, is listed in Table 1-1 for the recirculated and ventilated configurations.

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

Off-design performance in terms of capacity and COP are plotted in Figures 1-4 and 1-5 for the recirculated and ventilated configurations, respectively. In these plots, all data were reduced to ARI standard conditions and a 200°F water (solar energy) temperature, except for the parameters in the abscissa.

TABLE 1-1

DESIGN POINT PERFORMANCE

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

Parameter	Recirculated Configuration		Ventilated Configuration
Experimental Data	Full Flow	Half Flow	Full Flow
Capacity, Btu/hr	16,300	7,400	12,000
Coefficient of performance	0.60	0.56	0.48
Conditioned space airflow, scfm	760	380	760
Outdoor airflow, scfm	925	490	925
Water usage, gal/hr	3.9	2.0	3.9
Parasitic power (fans, pumps and and drive), kw	0.8	0.24	0.8
Predicted Data			
Capacity, Btu/hr	16,200	8,000	13,500
Coefficient of performance	0.52	0.46	0.47
Parasitic power, kw	0.75	--	0.75

Testing in the ventilated configuration was only performed at the full flow condition. Since examination of the data indicated very poor performance in this flow regime, testing at half-flow was not conducted.

<u>Adsorption Zone Extension</u>

In an effort to improve system performance, the adsorption zone was extended to expose the sorbent to the process airstream for a longer portion of the total cycle time. This was done by relocating two of the seals on the desiccant drum and by modification of the air manifolds. Attempts made to determine the performance of the system in this configuration were unsuccessful due to instrumentation problems and SODAC system problems attributed to leakage. Computer analyses of this extended adsorption zone configuration show potential capacity and coefficient of performance improvements of 30 percent and 12 percent, respectively.

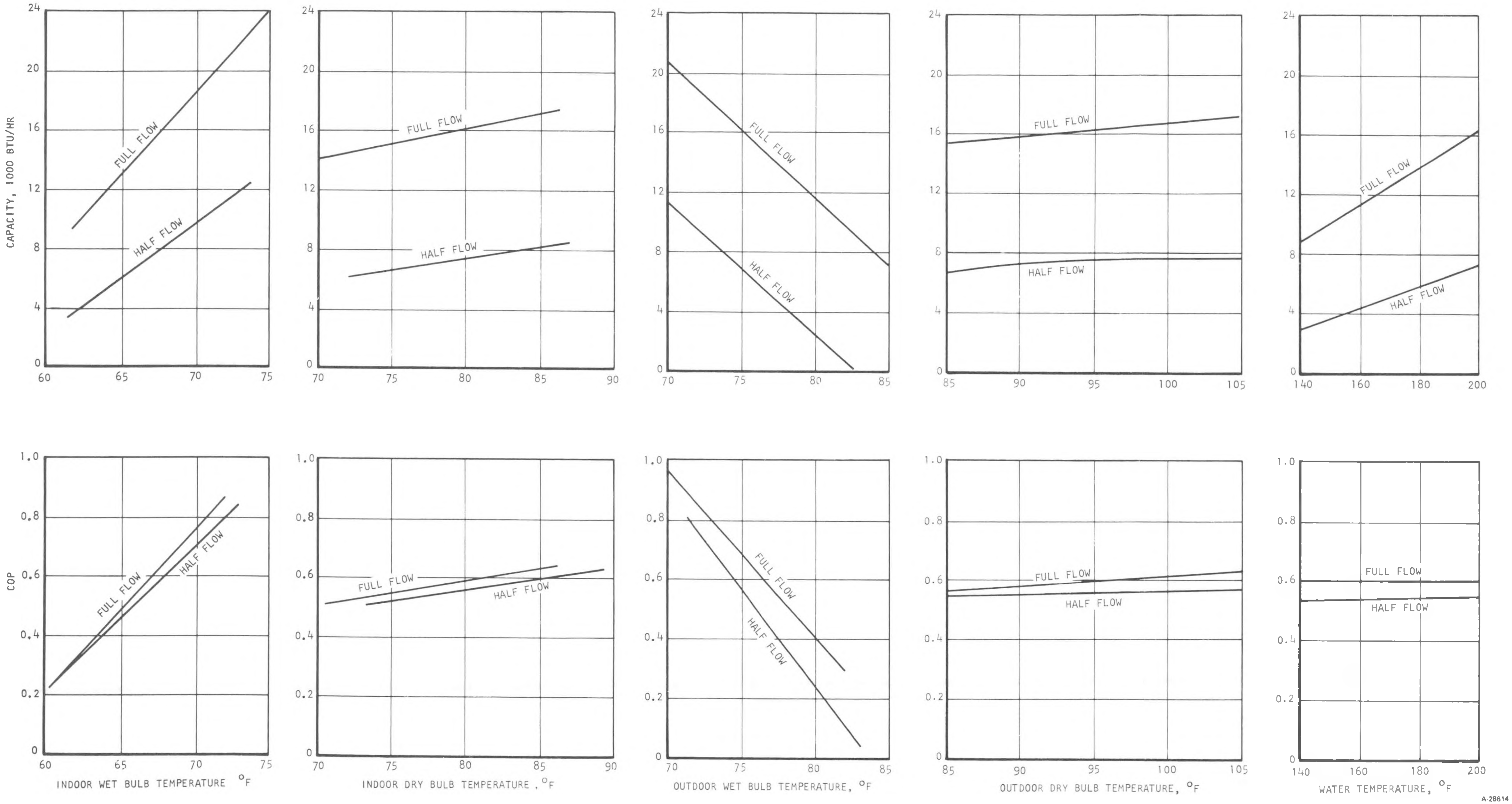

Figure 1-4. SODAC Performance in the Recirculated Configuration

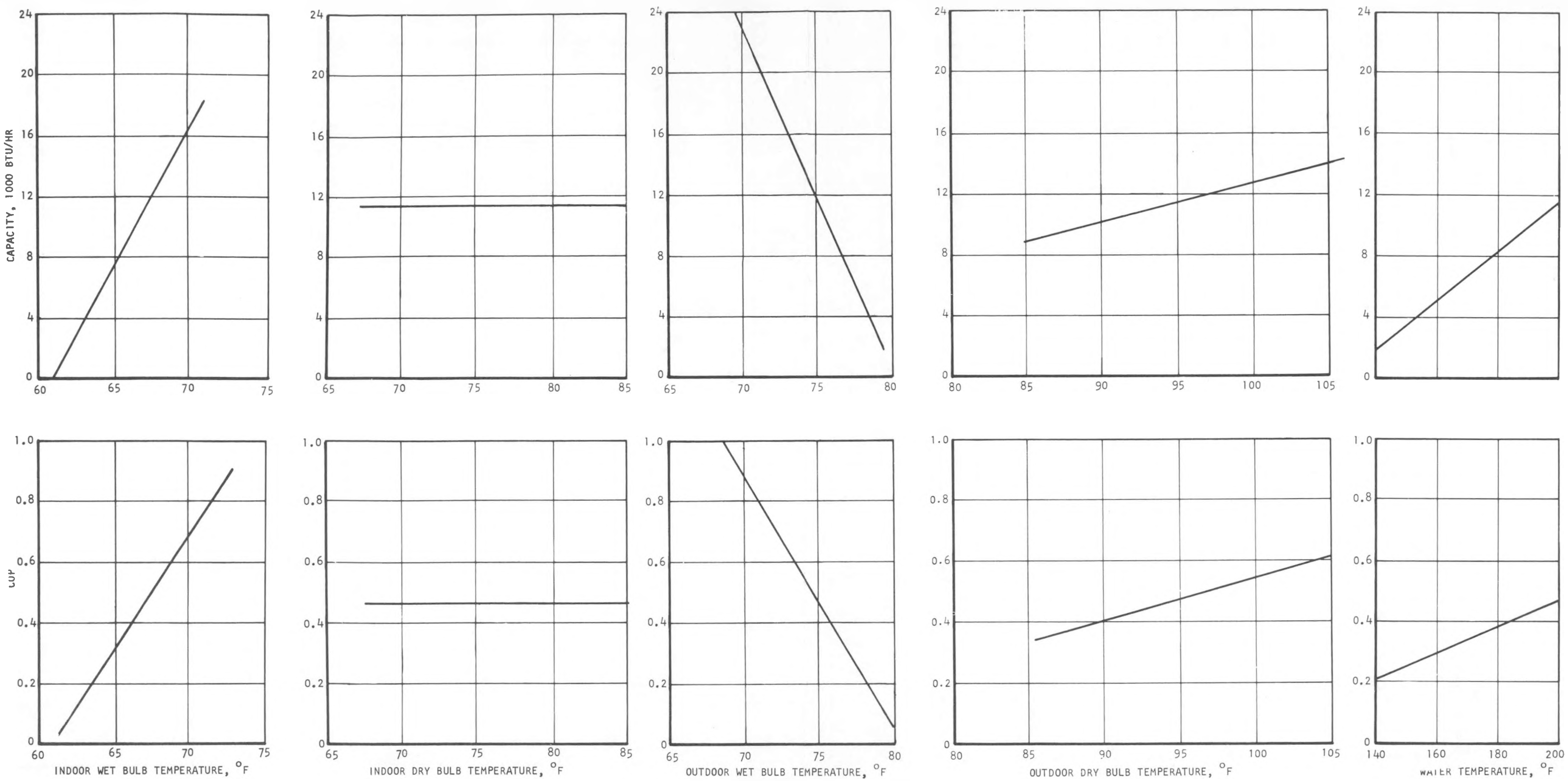

Figure 1-5. SODAC Performance in the Ventilated Configuration, Full Flow

CONCLUSIONS AND RECOMMENDATIONS

A capacity of 1.35 tons at a COP of 0.6 was achieved at the standard ARI rating conditions and at a desorption heat source temperature of 200°F. Significant improvements could be realized by (1) better seal dimensional control and (2) extension of the adsorption zone. Specific capacities of about 400 cfm/ton at a COP of about 0.80 are reasonable expectations for a preproduction prototype unit.

Novel desiccant systems are presently under test, with demonstrated COP's in excess of 1.0. These systems employ sorbent formulations with optimum heat capacity and optimum matching of heat and mass transfer characteristics. The development of such sorbents represents a breakthrough in desiccant system technology. It is believed that this level of performance could not be achieved with granular (or bead) sorbent beds such as used in the SODAC. For this reason, further development of the SODAC is not recommended at this time. The application of these novel sorbent formulations to a SODAC-type package could be investigated to take advantage of the cylindrical SODAC package.

SECTION 2

BASELINE SYSTEM DESCRIPTION

GENERAL

This section contains 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 configuration. Data for the ventilated configuration are presented at the end of this section.

FUNCTIONAL DESCRIPTION

A schematic of the system is given in Figure 2-1; the flow path shown corresponds to recirculation of conditioned space air through the SODAC. Outdoor ambient air is used for regeneration. 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. A detailed description of the system components is contained in References 3 and 4.*

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 zone. 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.

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*References are presented in Section 5.

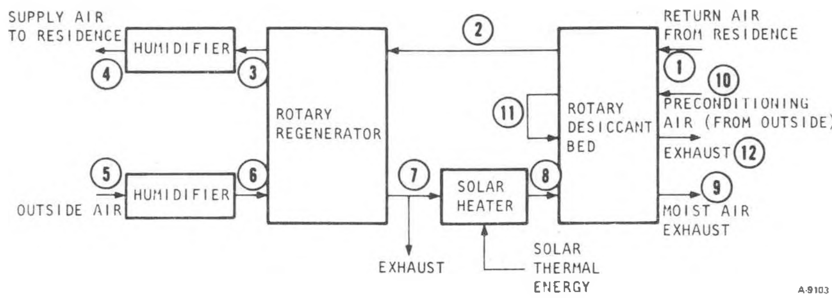


Figure 2-1. Air Conditioner Schematic

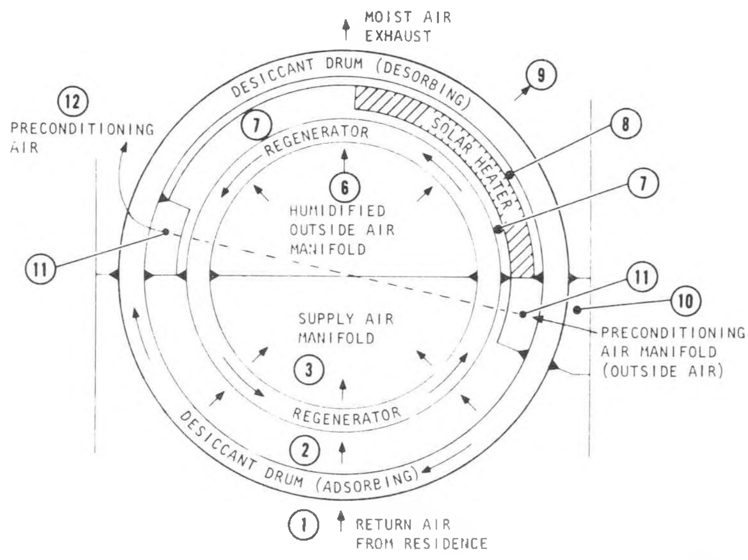


Figure 2-2. Dehumidifier Arrangement

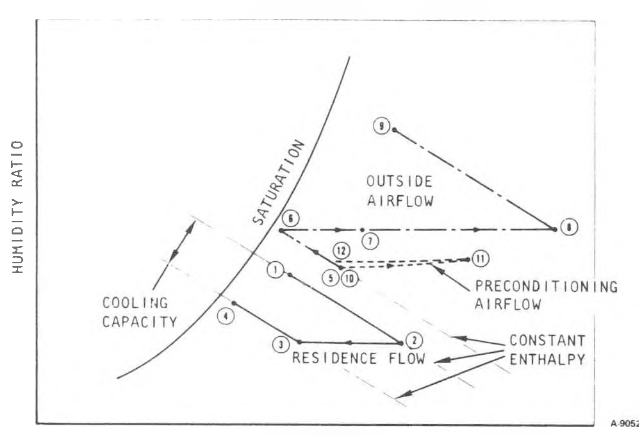


Figure 2-3. Psychrometric Process

Two basic seal configurations are necesssary to isolate the process air streams within the machine: (1) linear seals for interfacing with the perforated sheets of the drums, and (2) circular seals to interface with the drum top and bottom hoops. Figures 2-4 and 2-5 show the seal configurations used.

The linear seal uses a silicone rubber strip which acts to pressurize a Dacron felt pad on the sealing surface. The silicone rubber maintains adequate pressure on the seal with the air pressure in either direction relative to the direction of rotation. Partitions made of 0.032-in.-thick lexan prevent air leakage through the bed, and similarly, the regenerator matrix is partitioned between opposite seals.

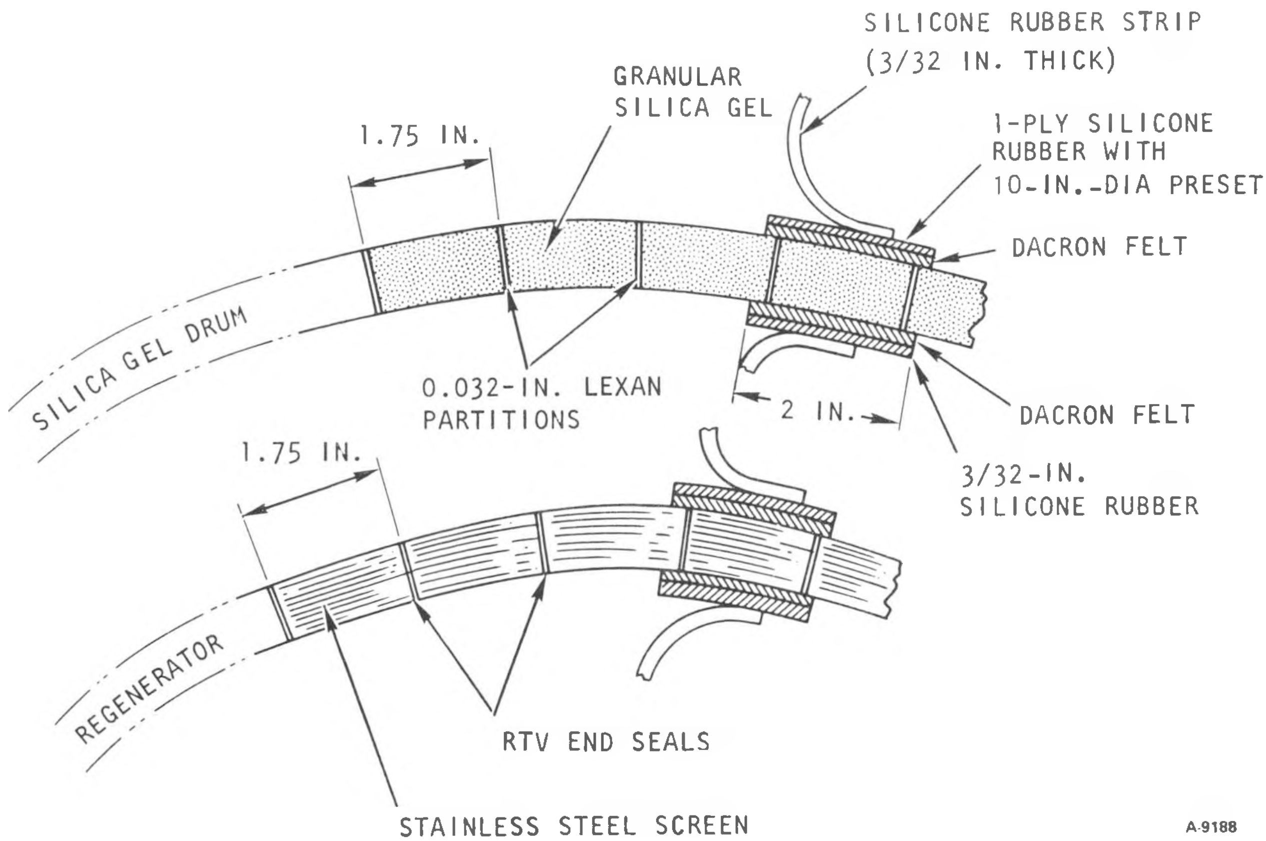


Figure 2-4. Linear Seal Configuration

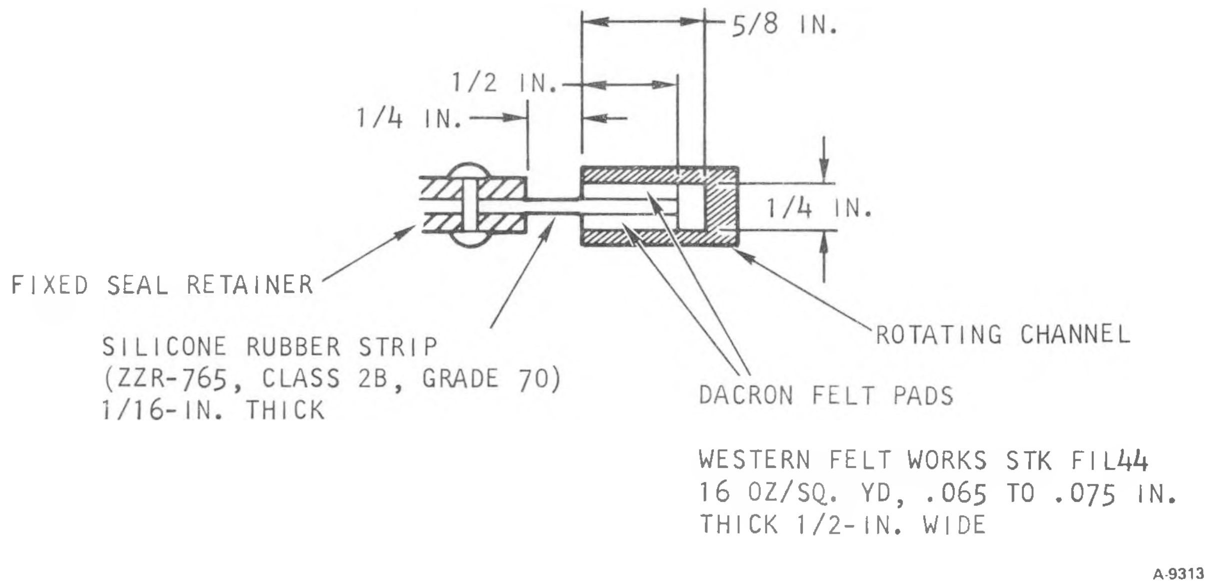


Figure 2-5. Circular Seal Configuration

For the circular seals depicted in Figure 2-5, a thin strip of relatively soft silicone rubber is used, with a Dacron felt pad on each side providing the sealing surface. The seal fits into a circular channel that rotates with the drum. The pressure across the seals changes direction as the drums rotate around their axis. The soft rubber is used to permit easy seating of the seal on either surface.

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 by 24 by 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. H_2O	Pressure drop: 0.19 in. H_2O
Airflow Rates	**Solar Heater**
Residence airstream: 830 scfm	Effectiveness: 85 percent
Preconditioning airstream: 120 scfm	Arc: 86.6 deg
Outside airstream (without pre-conditioning 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. H_2O
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 is defined in Figure 2-6, 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 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 the 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.

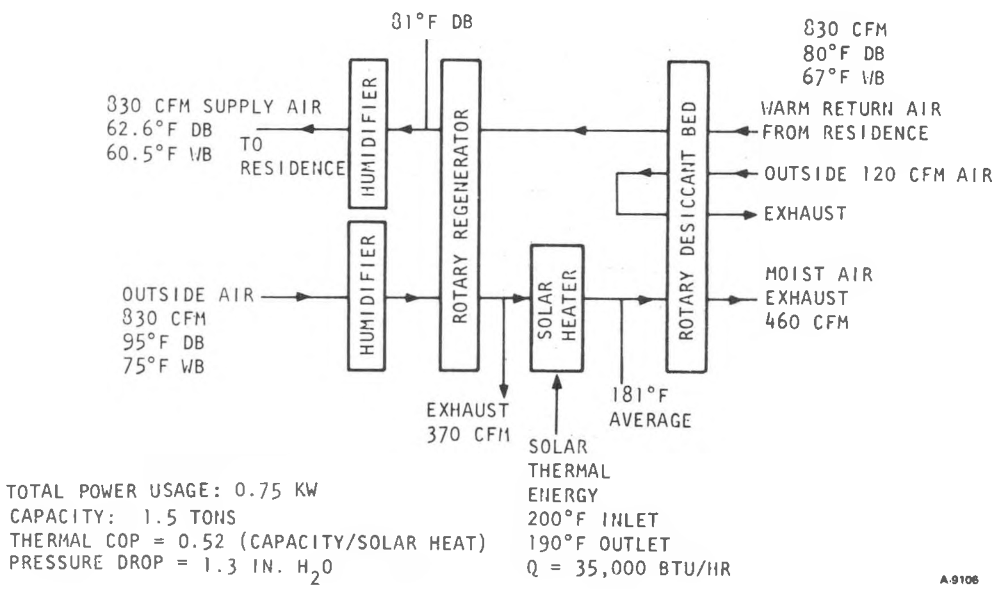


Figure 2-6. Baseline SODAC Performance

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 proportional to the quantity 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 used for testing were reduced by the same ratio.

SYSTEM CONTROL

Comfort is generally given as a 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 2-7 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 on 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.

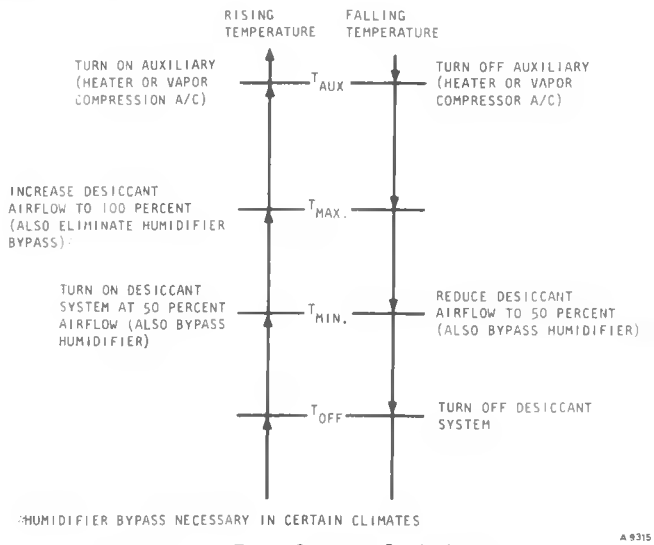
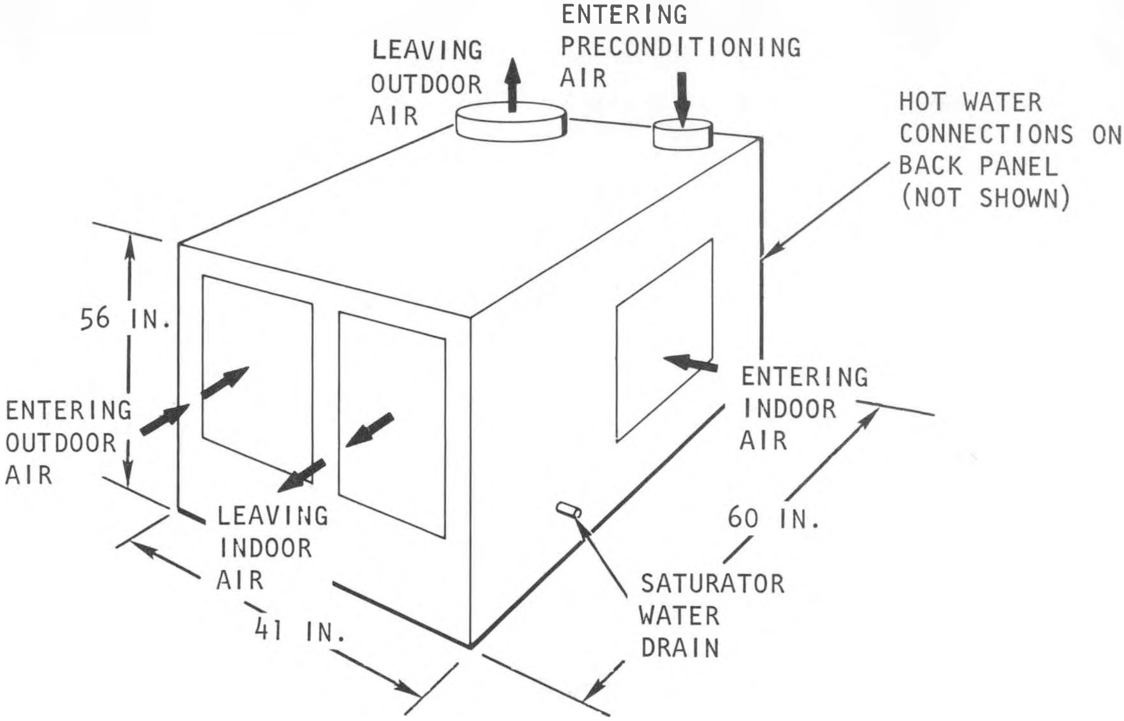


Figure 2-7. Control Scheme

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 2-8. The overall dimensions of the package, excluding duct attachment points, are 60-in. long by 41-in. wide by 56-in. high.



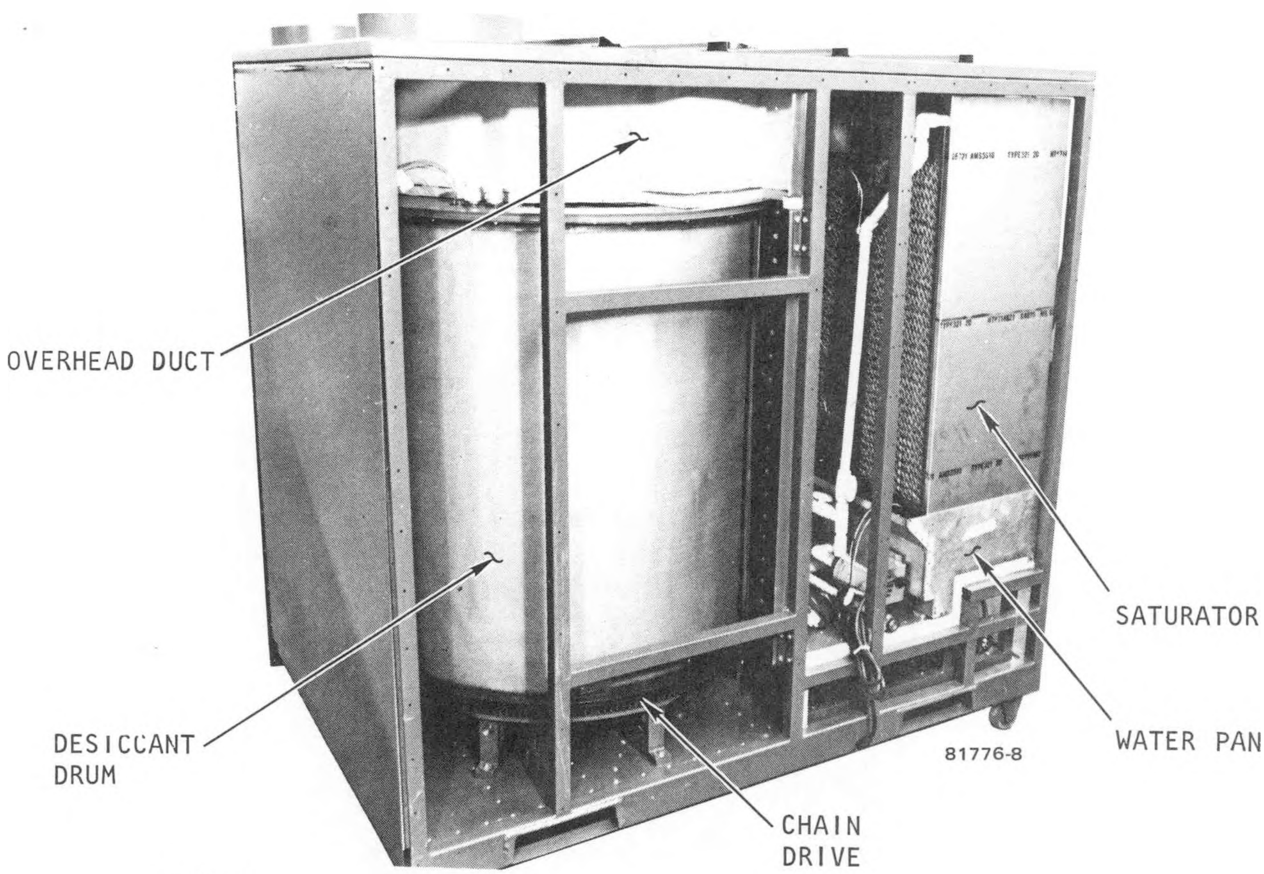


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Figure 2-8. SODAC Package

VENTILATED CONFIGURATION

In the ventilated 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 2-9.

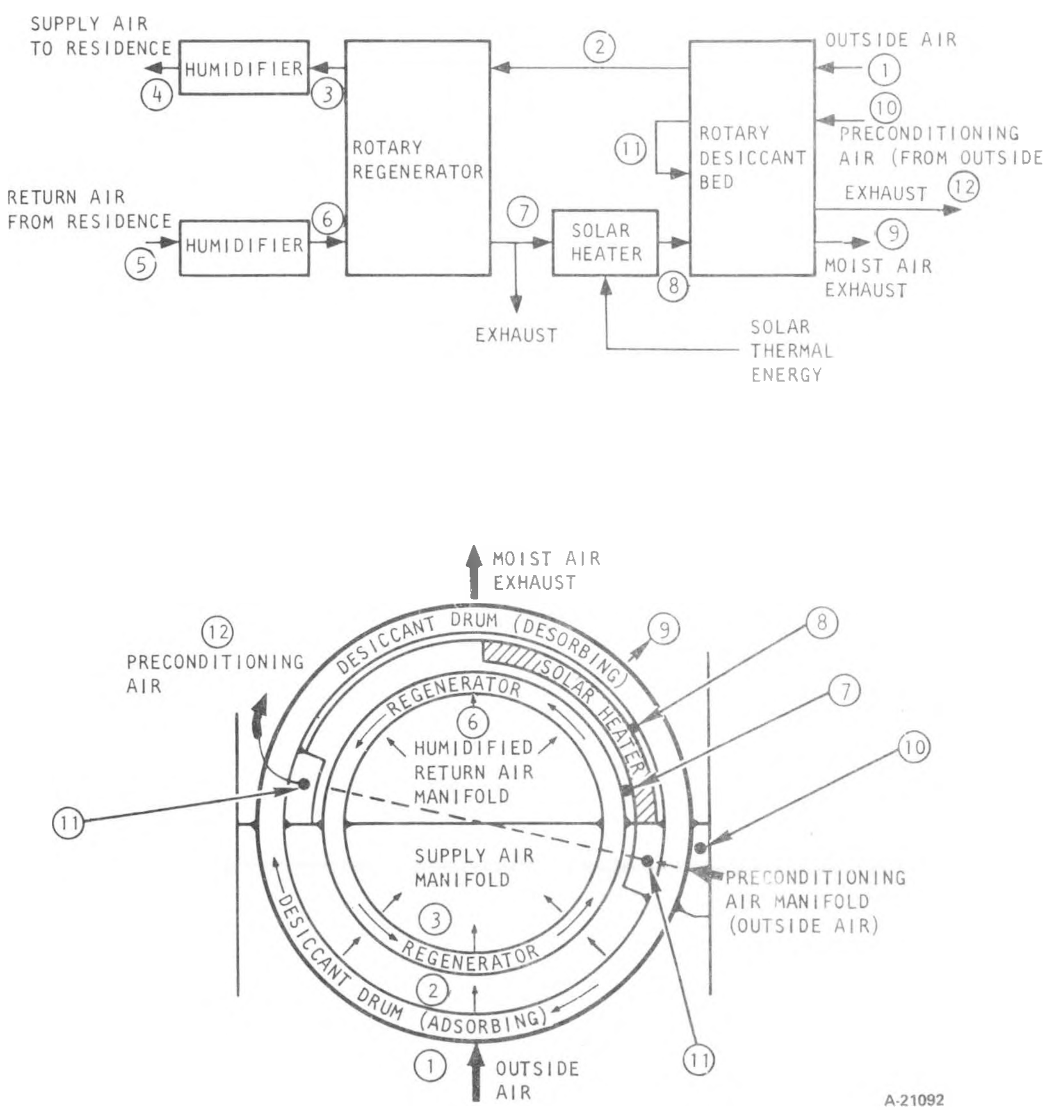


Figure 2-9. Ventilated Configuration

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. Detailed descriptions of the test facilities and equipment, system instrumentation, and test procedures are presented in Reference 3.

TEST FACILITIES AND EQUIPMENT

A schematic of the test stand is shown in Figure 3-1 (recirculated configuration). Testing in the ventilated configuration required rerouting the ducting between the indoor and outdoor chambers and the SODAC. 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 system test.

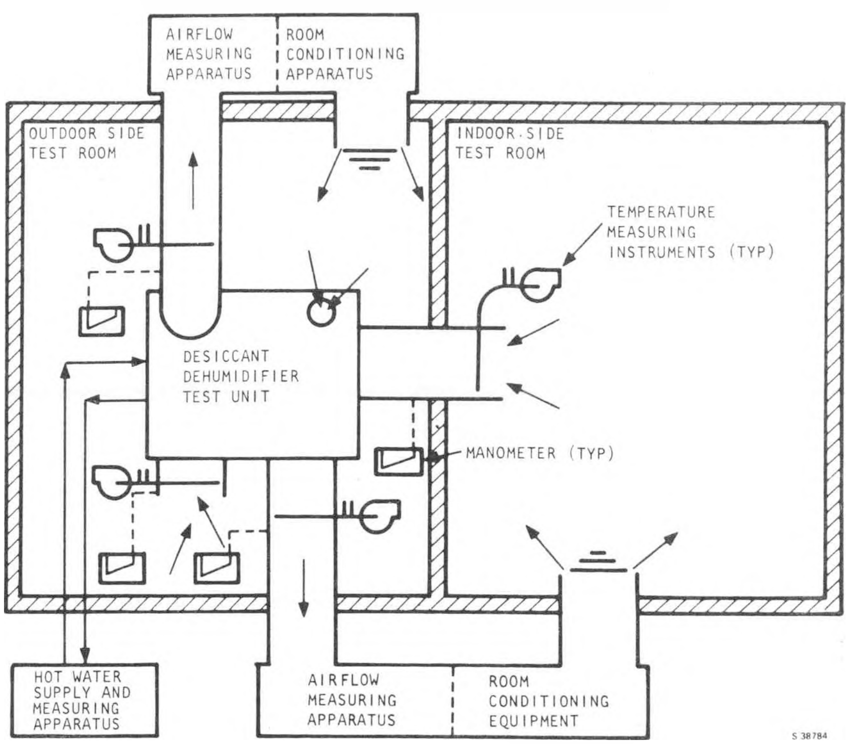


Figure 3-1. Test Stand Schematic

INSTRUMENTATION

The location of most instruments within the SODAC package is illustrated in Figure 3-2. All parameters measured are listed in Table 3-1.

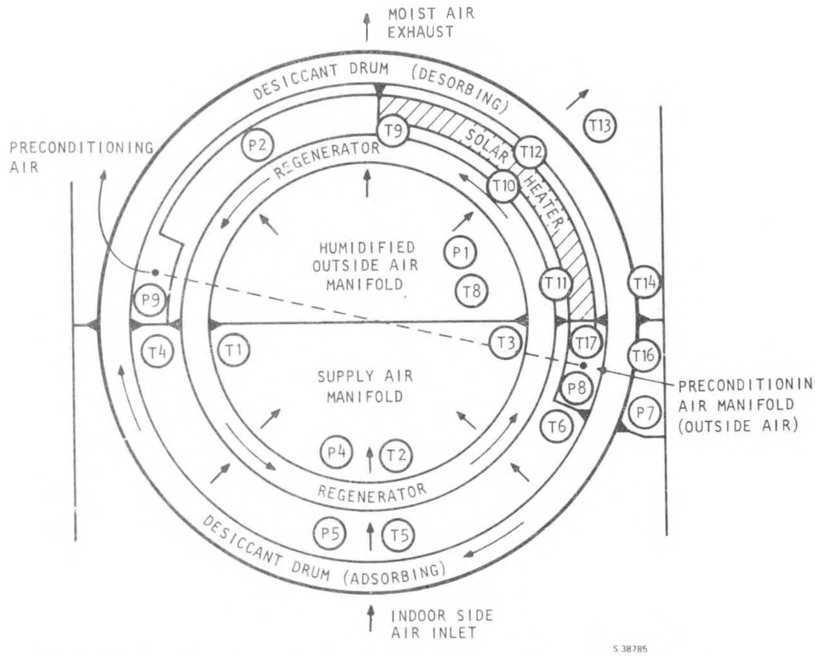


Figure 3-2. Location of Internal Pressure Taps and Temperature Probes

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 was concerned with complete performance characterization of the SODAC in the recirculated and ventilated configurations. Design point performance was used as a reference for instrumentation check and also to verify the operation of the SODAC and of the test chambers. Design point is defined by (1) the standard ARI rating conditions, (2) the SODAC design flow rates, and (3) the desorption temperature. Specifically, the following parameters define the design point conditions at full flow.

(a) Indoor conditions: 80°F db, 67°F wb, 760 scfm

(b) Outdoor conditions: 95°F db, 75°F wb, 930 scfm (total)

(c) Water heat source: 200°F, 3290 lb/hr

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

Tag No.	Data	Range	Units	Accuracy
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	--

The SODAC performance in terms of capacity and COP (as affected by indoor, outdoor, and water temperatures) was determined by subjecting the unit to the test sequence of Table 3-2. This was done for the following configuration flow rates:

(a) Recirculated Configuration--Full flow

(b) Recirculated Configuration--Half flow

(c) Ventilated Configuration--Full flow

Testing at half-flow in the ventilated configuration was not done because of the very poor performance obtained at full flow. Rather, the program was reoriented to include modification of the system to increase the adsorption zone; this portion of the program is reported at the end of this section.

A total of 75 test runs were conducted as part of the Phase II test program. This experimental program was started in Phase I, which included system development and performance characterization. All performance characterization data are presented in this report.

TEST DATA

In this subsection, all figures and tables are presented at the end in the order of this reference in the text. Test data are presented as follows:

(a) Recirculated Configuration--Full flow

(1) Design point data (Figure 3-3)

(2) Summary of full flow test data (Table 3-3)

(3) Effect of indoor wet bulb and dry bulb temperatures (Figure 3-4)

(4) Effect of outdoor wet bulb and dry bulb temperatures (Figure 3-5)

(5) Effect of water temperature (Figure 3-6)

(b) Recirculated Configuration--Half flow

(1) Performance at ARI conditions (Figure 3-7)

(2) Summary of half-flow test data (Table 3-4)

(3) Effect of indoor wet bulb and dry bulb temperatures (Figure 3-8)

(4) Effect of outdoor wet bulb and dry bulb temperatures (Figure 3-9)

(5) Effect of water temperature (Figure 3-10)

(c) Ventilated Configuration--Full flow

(1) Design point data (Figure 3-11)

(2) Summary of full flow test data (Table 3-5)

(3) Effect of indoor wet bulb and dry bulb temperatures (Figure 3-12)

(4) Effect of outdoor wet bulb and dry bulb temperatures (Figure 3-13)

(5) Effect of water temperature (Figure 3-14)

In all cases, the test data plots are compared to the computer predictions, and the test run numbers shown are used for purposes of cross-reference with the appropriate summary data in tabular form. Off-design performance plots were prepared by correcting the test data to exact ARI conditions with the exception of the parameter in the abscissa. The slope of the experimental curves were used to derive the correction factors in terms of all interfacing parameters.

COMPARISON OF RECIRCULATED AND VENTILATED PERFORMANCE

A comparison of the SODAC performance in the recirculated and ventilated configurations is shown in Figure 3-15. 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.

SORBENT STABILITY

Through the entire test period, approximately 1100 hr of operating time were accumulated on the SODAC, with no indication of a decrease in capacity of the silica gel sorbent in that period. Settling of the sorbent within the thin (1.25-in.-thick) drum was experienced, and refilling of the drum was necessary at periodic intervals. It is believed that settling occurs as a result of breaking of the sharp edges of the sorbent particles when subjected to the pressure cycling imposed by periodic flow reversals.

As the sorbent settles within the bed, the sorbent particles are carried by the airstream such that the void created is located in a particular area of the bed, resulting in a triangular void at the top of the sorbent bed. This void constitutes a bypass area for the airstreams, and its presence is easily detected by monitoring the air temperature at the outlet of the desorbing bed.

It is believed that this settling problem could be controlled by exercising a slight pressure on the top surface of the sorbent bed. This could be done with foam rubber. In this arrangement, the sorbent granules at the top of the bed could not be violently shaken and broken by the airstream, and the effect of settling would be greatly minimized, if not eliminated entirely.

Another solution to this problem could be the use of silica gel beads which have no sharp edges, thus eliminating the high stresses at the contact point between the sorbent particles.

ADSORPTION ZONE EXTENSION

In the baseline system design, a portion of the sorbent bed adjacent to the solar heater (See Figure 2-1) is not used. In an effort to increase the performance of the SODAC, a computer analysis was conducted to determine the capacity and performance improvement which could be realized by extending the adsorption zone. This can be done by relocating the hot preconditioning air duct on the desorption side of the bed so as to increase the adsorption zone. Figure 3-16 shows the present and modified SODAC configurations. The modification involves reworking the desorption side ducting and relocating the seals identified as "A" in Figure 3-16.

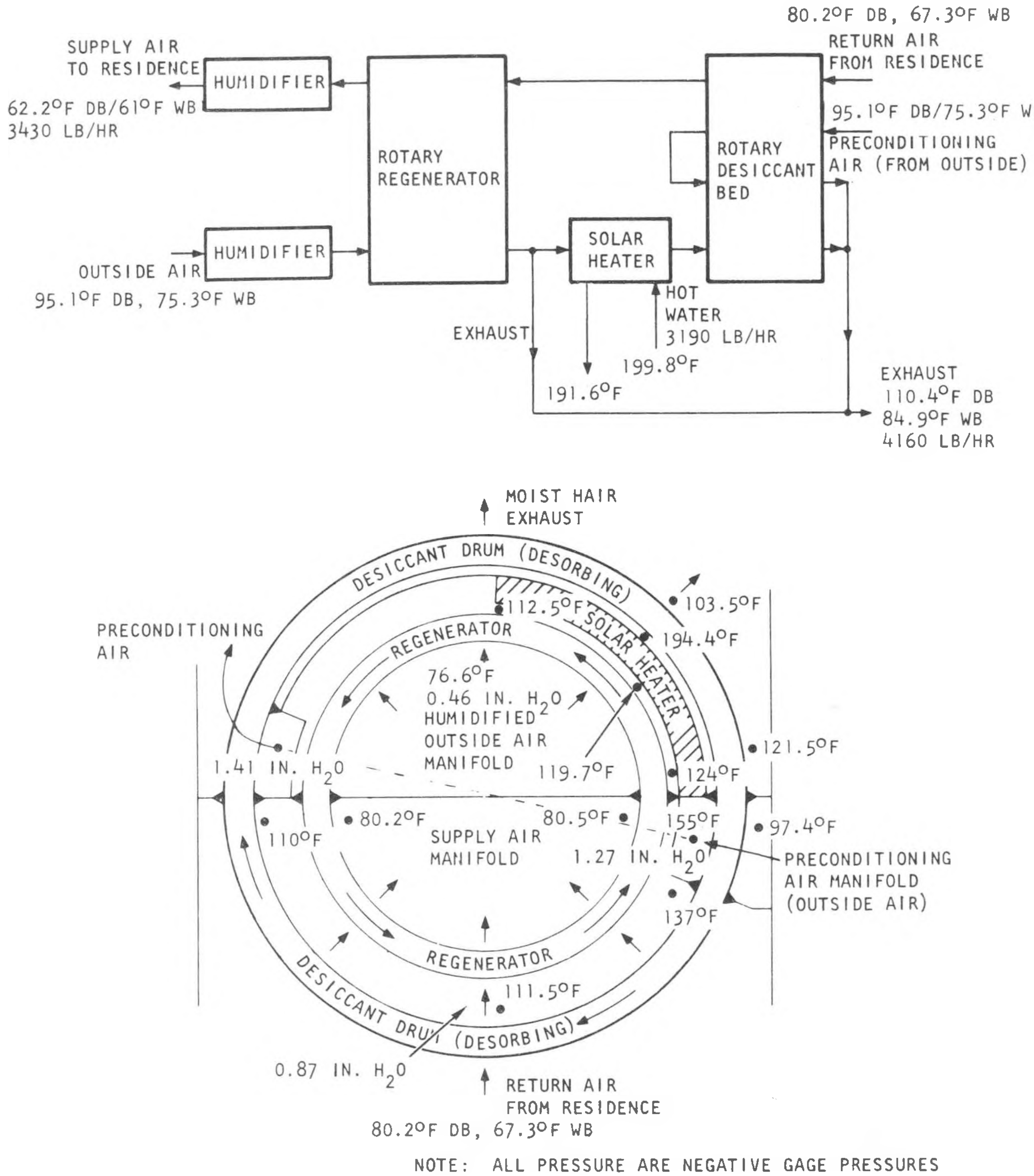
The estimated effects on SODAC capacity and COP are plotted in Figure 3-17 as a function of the adsorption zone extension measured in degrees around the periphery of the drum. In the baseline configuration, the adsorption zone extends through a 158-deg arc; the actual baseline system construction limits the maximum shifting of the seals by 35 deg. This represents a 22 percent increase in the adsorption zone. Corresponding to this maximum value, increases in capacity and COP of 30 and 12 percent, respectively, are estimated.

These modifications were incorporated in the design. Unfortunately, experimental evaluation could not be completed prior to completion of the program because of instrumentation and system problems.

TABLE 3-2

TEST SEQUENCE

Run No.	Hot Water Temperature, °F	Indoor Temperature		Outdoor Temperature		Remarks
		DB, °F	WB, °F	DB, °F	WB, °F	
1	200	80	67	95	75	Design point
2	175	80	67	95	75	Effect of water temperature
3	150	80	67	95	75	Effect of water temperature
4	200	80	67	95	70	Effect of outdoor WB
5	200	80	67	95	80	Effect of outdoor WB
6	200	80	67	85	75	Effect of outdoor DB
7	200	80	67	105	75	Effect of outdoor DB
8	200	80	60	95	75	Effect of indoor WB
9	200	80	72	95	75	Effect of indoor WB
10	200	80	67	95	75	Design point
11	200	70	67	95	75	Effect of indoor DB
12	200	85	67	95	75	Effect of indoor DB
13	200	80	67	107	71	Dry climate (Phoenix)
14	175	80	67	107	71	Humid climate (Phoenix)
15	200	75	63	107	71	Humid climate (Phoenix)
16	200	80	67	90	80	Humid climate (Key West)
17	175	80	67	90	80	Humid climate (Key West)
18	200	75	63	90	80	Humid climate (Key West)



Q SOLAR = 26,980 BTU/HR
CAPACITY AT TEST CONDITIONS: 16,400 BTU/HR
COP AT TEST CONDITIONS: 0.61
CAPACITY AT STANDARD CONDITIONS: 16,300 BTU/HR
COP AT STANDARD CONDITIONS: 0.60

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Figure 3-3. Typical Design Point Test Data for Recirculated Mode, Full Flow

TABLE 3-3

TEST DATA SUMMARY FOR RECIRCULATED CONFIGURATION, FULL FLOW

Run No.	Purpose	Test Conditions: Outdoor: Inlet Dry Bulb, °F	Test Conditions: Outdoor: Inlet Wet Bulb, °F	Test Conditions: Outdoor: Total Flow, scfm	Test Conditions: Indoor: Inlet Dry Bulb, °F	Test Conditions: Indoor: Inlet Wet Bulb, °F	Test Conditions: Indoor: Flow, scfm	Test Conditions: Water Inlet Temp., °F	Test Results: Indoor Outlet: Dry Bulb, °F	Test Results: Indoor Outlet: Wet Bulb, °F	Test Results: Solar Q, Btu/hr	Test Results: Capacity, Btu/hr	Test Results: COP	Corrected Data: Capacity, Btu/hr	Corrected Data: COP
5/7	Design point	95.1	75.3	924	80.2	67.3	762	199.8	62.2	61.0	26,980	16,400	0.608	16,330	0.601
5/8	Design point	94.7	74.8	926	80.5	67.1	763	198.6	62.1	61.0	26,980	15,820	0.586	15,550	0.573
7/1	Low water temperature	94.9	75.3	925	80.4	67.6	757	152.2	64.8	64.3	17,100	8,758	0.487	8,245	0.515
7/3	Low outdoor WB	95.2	69.2	926	80.9	67.4	766	199.5	58.5	58.4	27,640	22,990	0.832	22,280	0.809
7/4	High outdoor WB	94.5	79.9	922	80.2	67.2	759	199.2	64.7	63.8	28,290	8,940	0.316	8,770	0.306
7/5	Design point	92.5	75.0	919	80.5	66.8	762	199.8	61.9	60.5	27,970	16,210	0.580	16,390	0.586
7/6	Low water temperature	94.8	75.0	929	80.2	67.4	761	174.4	63.1	62.0	22,700	14,180	0.624	13,640	0.605
7/7	Low inside WB	95.0	75.3	921	80.5	64.0	762	200.0	60.3	59.1	28,950	11,930	0.412	12,150	0.417
7/8	High indoor WB	94.9	75.4	921	80.4	71.2	755	200.4	64.7	63.7	26,650	21,130	0.793	21,420	0.803
7/9	Low outdoor DB	85.3	74.9	924	80.0	67	760	200.0	62.3	61.0	27,970	15,570	0.557	15,470	0.554
8/1	Low indoor DB	95.1	75	929	75	67.4	763	199.4	62.7	61.3	27,970	16,070	0.575	15,660	0.557
8/2	High outdoor DB	105	75.2	925	80.6	67.2	764	200.1	61.8	60.5	27,640	17,330	0.627	17,160	0.62
8/3	High indoor DB	94.6	75.1	924	85.2	67.2	764	200.2	61.5	60.1	28,290	18,190	0.643	18,030	0.637
8/4	Low water temperature	95.2	75	936	80.5	67.1	762	153.0	64.1	63.1	17,440	10,460	0.60	10,230	0.592
11/2	Hot-dry climate simulation	106.9	75.3	908	80.2	67.6	761	198	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	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	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	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	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	65.3	64.7	23,700	5,958	0.25	--	--
11/8	Design point	95.1	75.1	910	80.6	67.5	759	198.8	61.5	60.6	29,300	17,900	0.61	--	--
12/1	Design point	95.2	75.0	917	80.3	66.9	761	199.2	61.8	60.6	26,978	16,216	0.601	16,334	0.605
14/2	Design point	95.2	75.7	913	80.5	66.8	758	198.6	613	60.7	26,978	15,697	0.581	16,784	0.606

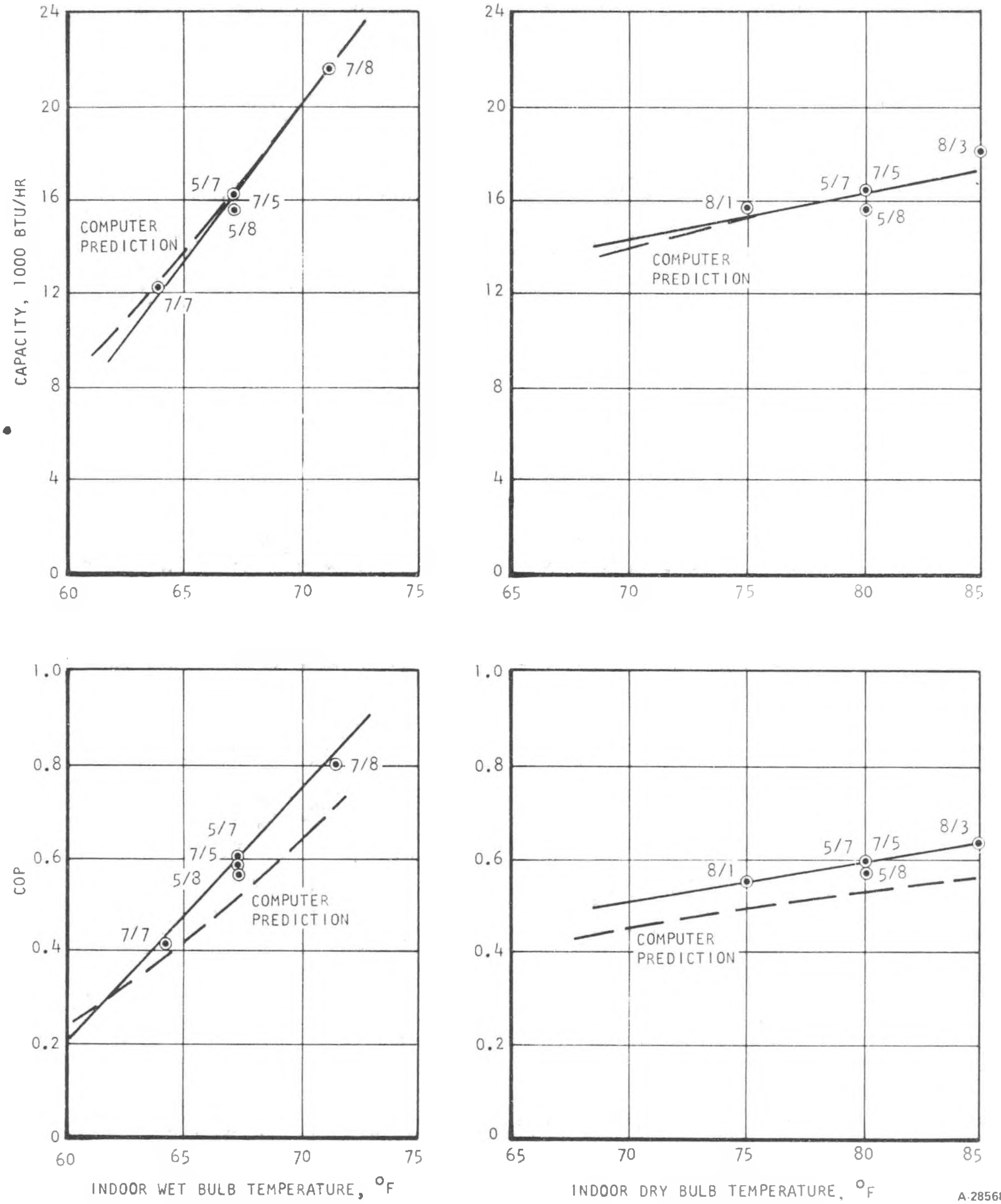


Figure 3-4. Effect of Indoor Temperatures on Recirculated Configuration, Full Flow

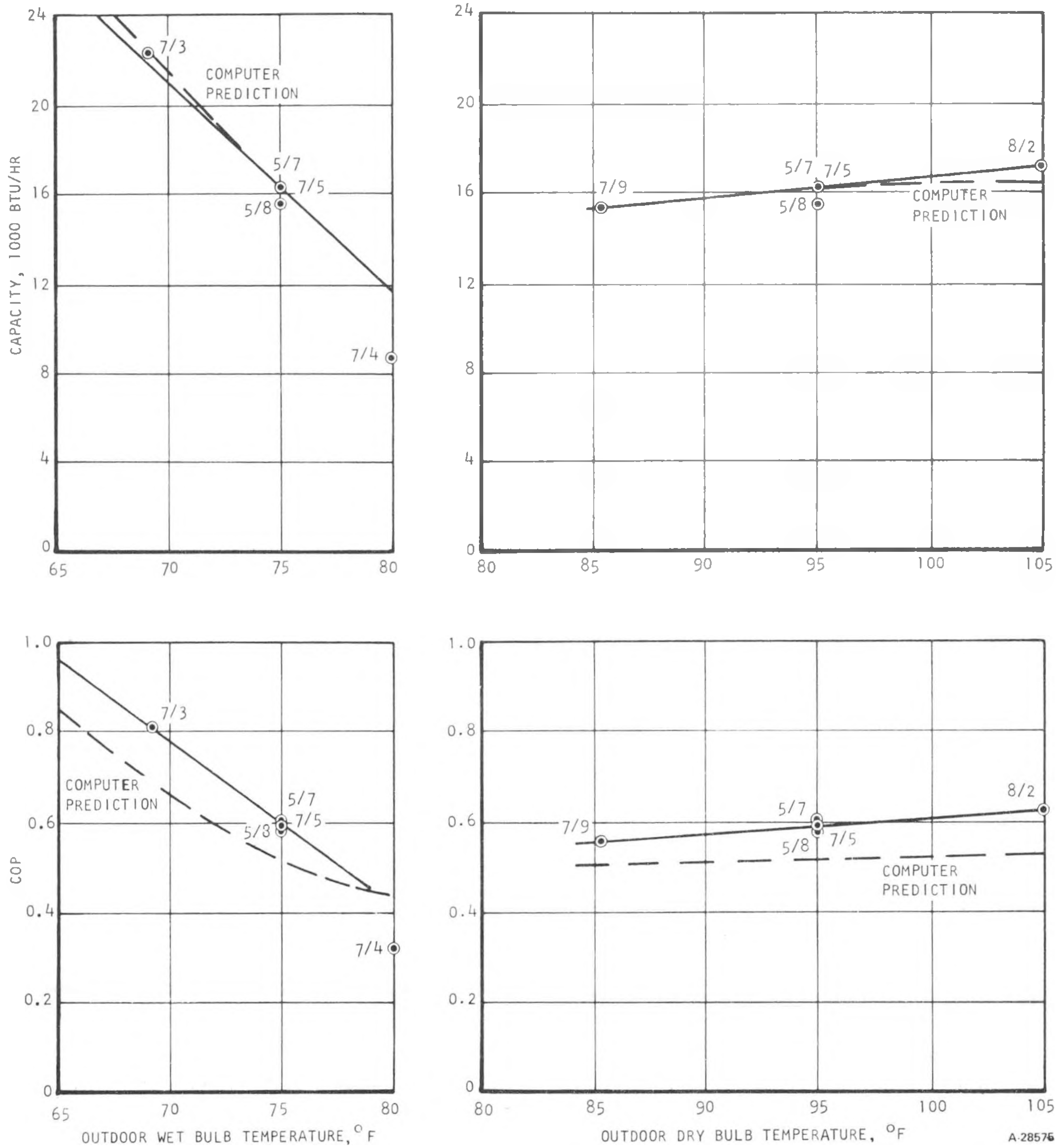


Figure 3-5. Effect of Outdoor Temperatures on Recirculated Configuration, Full Flow

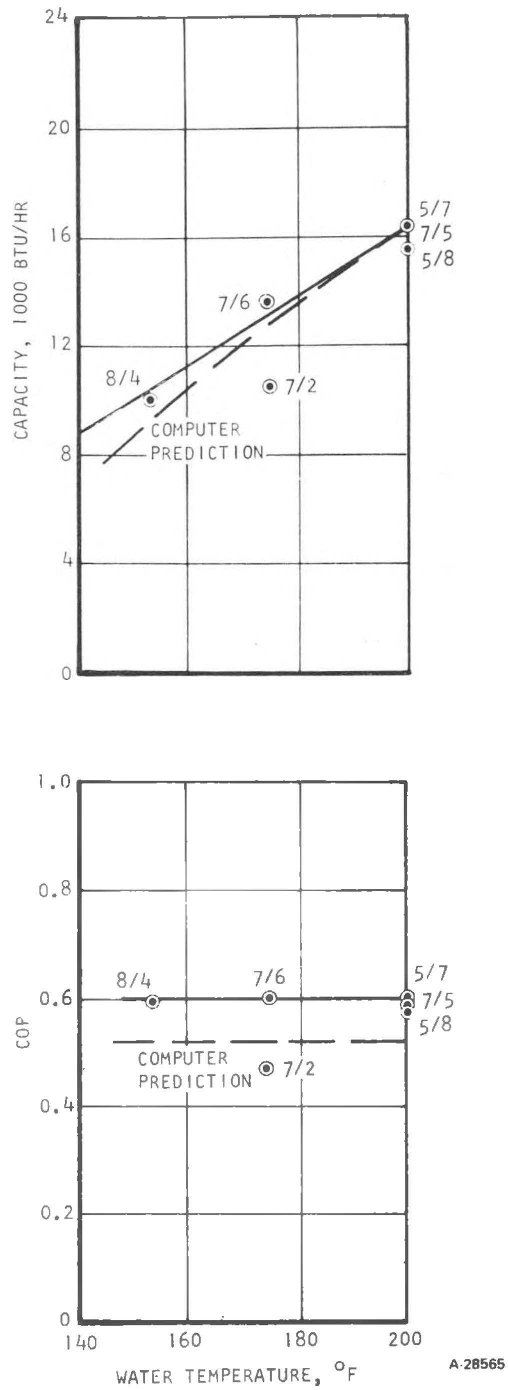
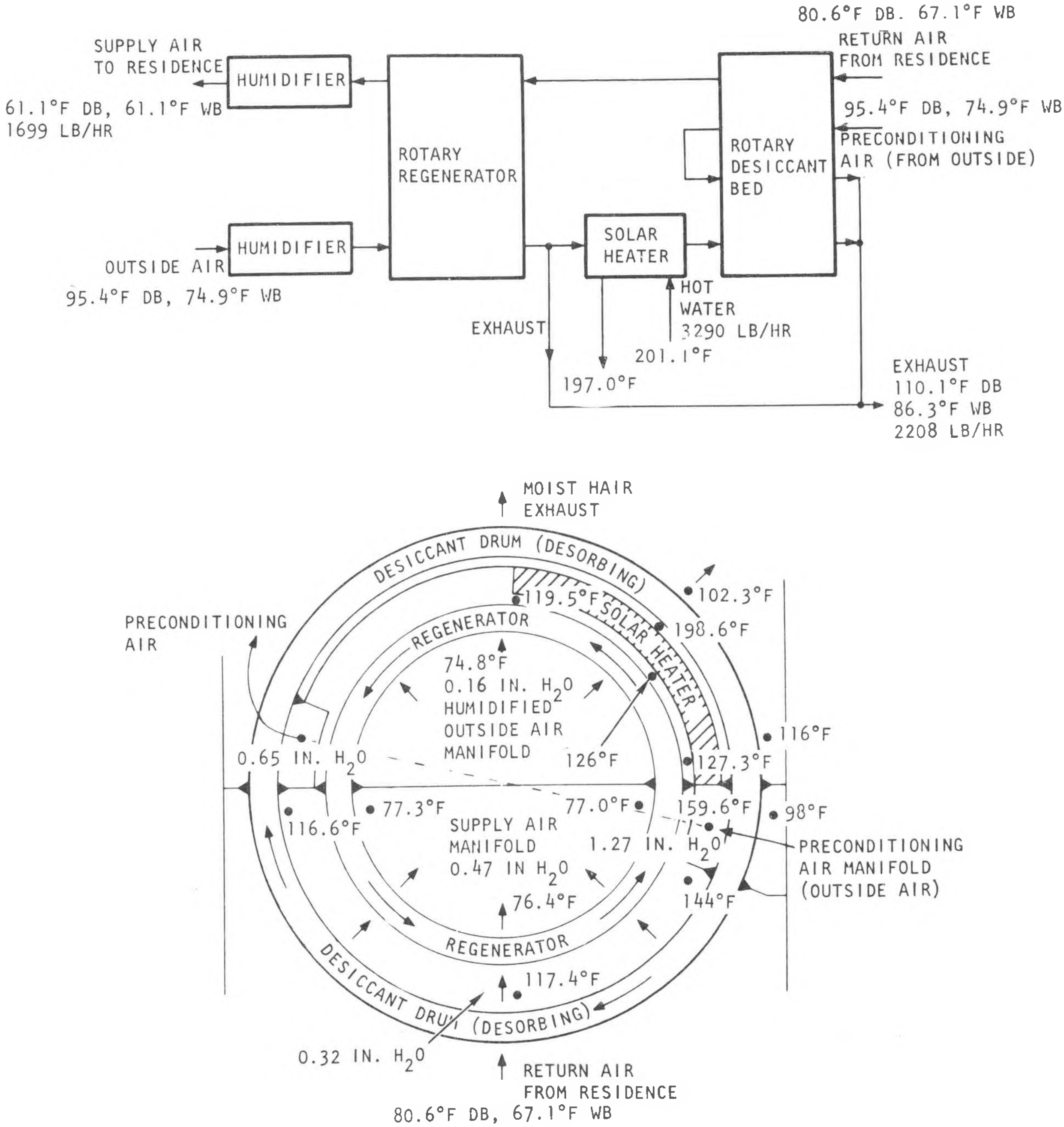


Figure 3-6. Effect of Water Temperature on Recirculated Configuration, Full Flow



NOTE: ALL PRESSURE ARE NEGATIVE GAGE PRESSURES

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

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Figure 3-7. Typical Performance at ARI Conditions for Recirculated Configuration, Half Flow

TABLE 3-4

TEST DATA SUMMARY FOR RECIRCULATED CONFIGURATION, HALF FLOW

		Test Conditions							Test Results					Corrected Data	
		Outdoor			Indoor				Indoor Outlet						
Run No.	Purpose	Inlet Dry Bulb, °F	Inlet Wet Bulb, °F	Total Flow, scfm	Inlet Dry Bulb, °F	Inlet Wet Bulb, °F	Flow, scfm	Water Inlet Temp., °F	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	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	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	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	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	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	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	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	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	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	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	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	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	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	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	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	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	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	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	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	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	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	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	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	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	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	57.9	57.9	13,160	6095	0.463	-	-

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

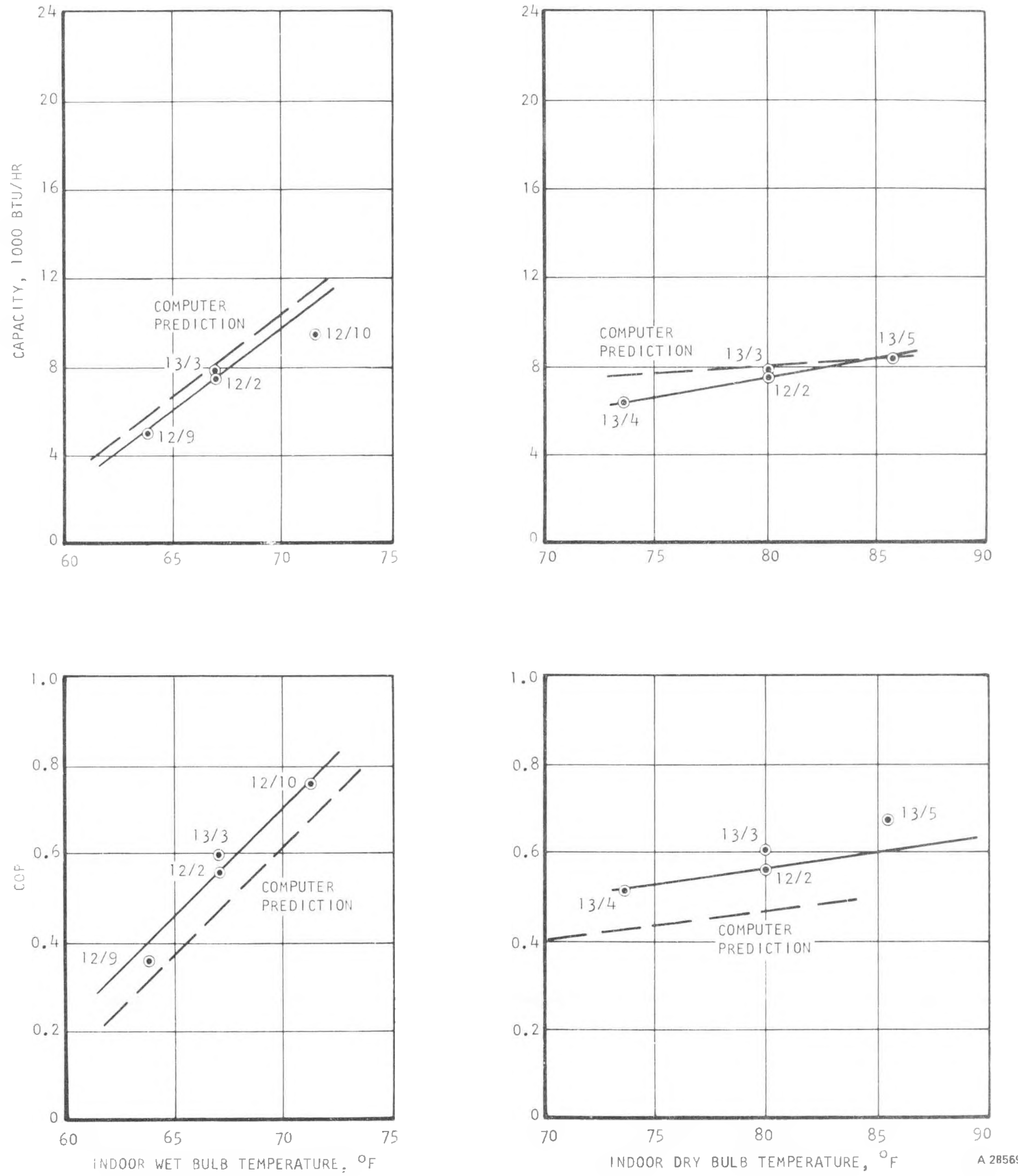


Figure 3-8. Effect of Indoor Temperatures on Recirculated Configuration, Half-Flow

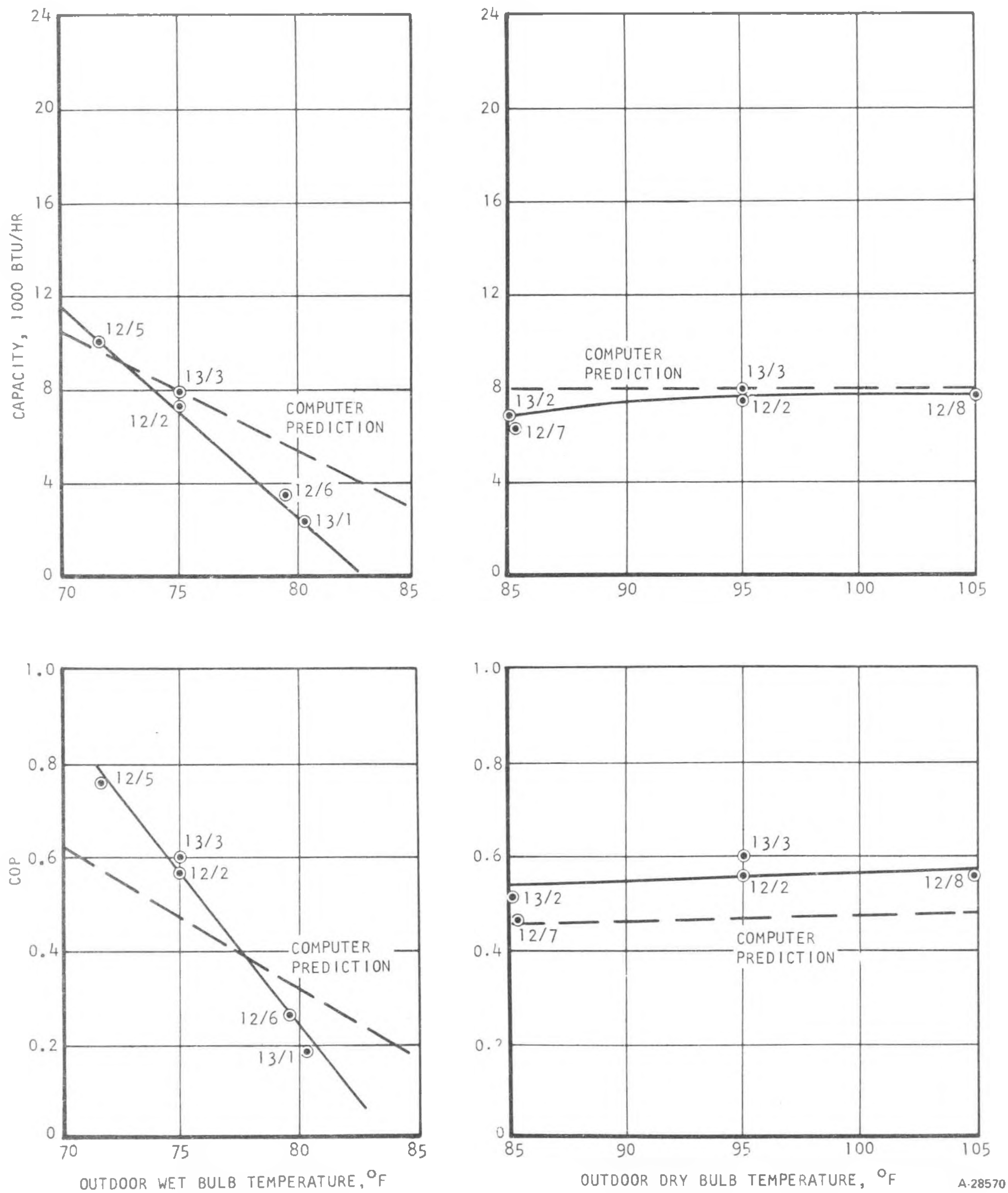


Figure 3-9. Effect of Outdoor Temperatures on Recirculated Configuration, Half-Flow

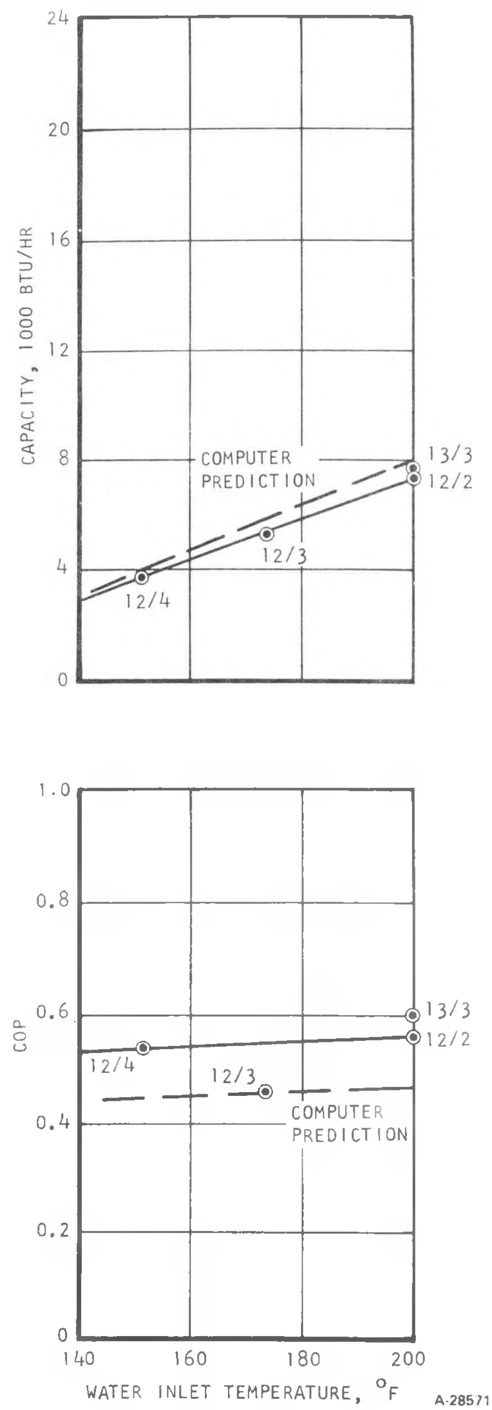
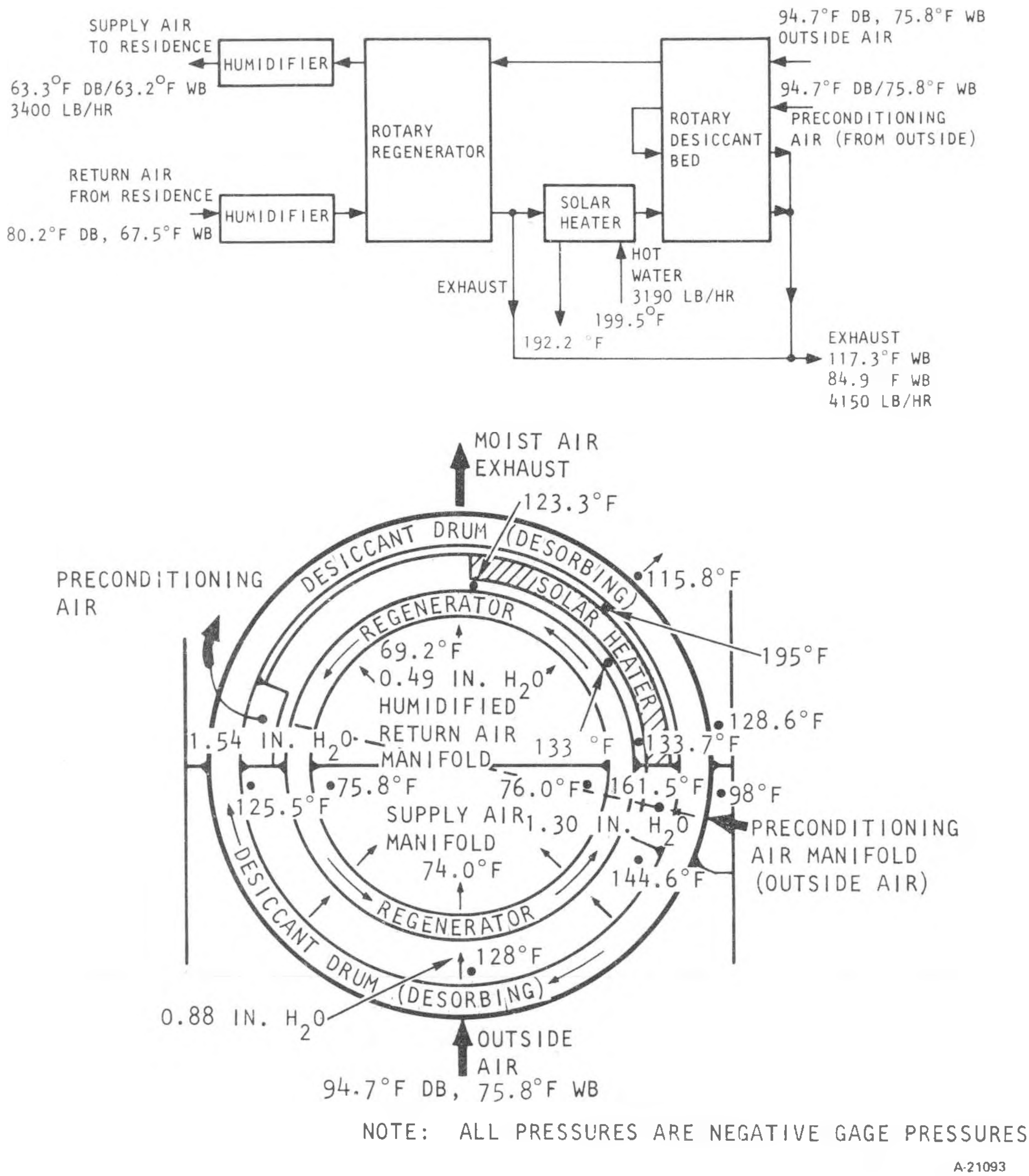


Figure 3-10. Effect of Water Temperature on Recirculated Configuration, Half-Flow



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 3-11. Typical Performance at ARI Conditions for Ventilated Configuration, Full Flow

TABLE 3-5

TEST DATA SUMMARY FOR VENTILATED CONFIGURATION, FULL FLOW

Run No.	Purpose	Test Conditions: Outdoor Inlet Dry Bulb, °F	Outdoor Inlet Wet Bulb, °F	Outdoor Total Flow, scfm	Indoor Inlet Dry Bulb, °F	Indoor Inlet Wet Bulb, °F	Indoor Flow scfm	Water Inlet Temp., °F	Test Results: Indoor Outlet Dry Bulb, °F	Indoor Outlet Wet Bulb, °F	Solar Q, Btu/hr	Capacity, Btu/hr	COP	Corrected Data: Capacity, Btu/hr	Corrected 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

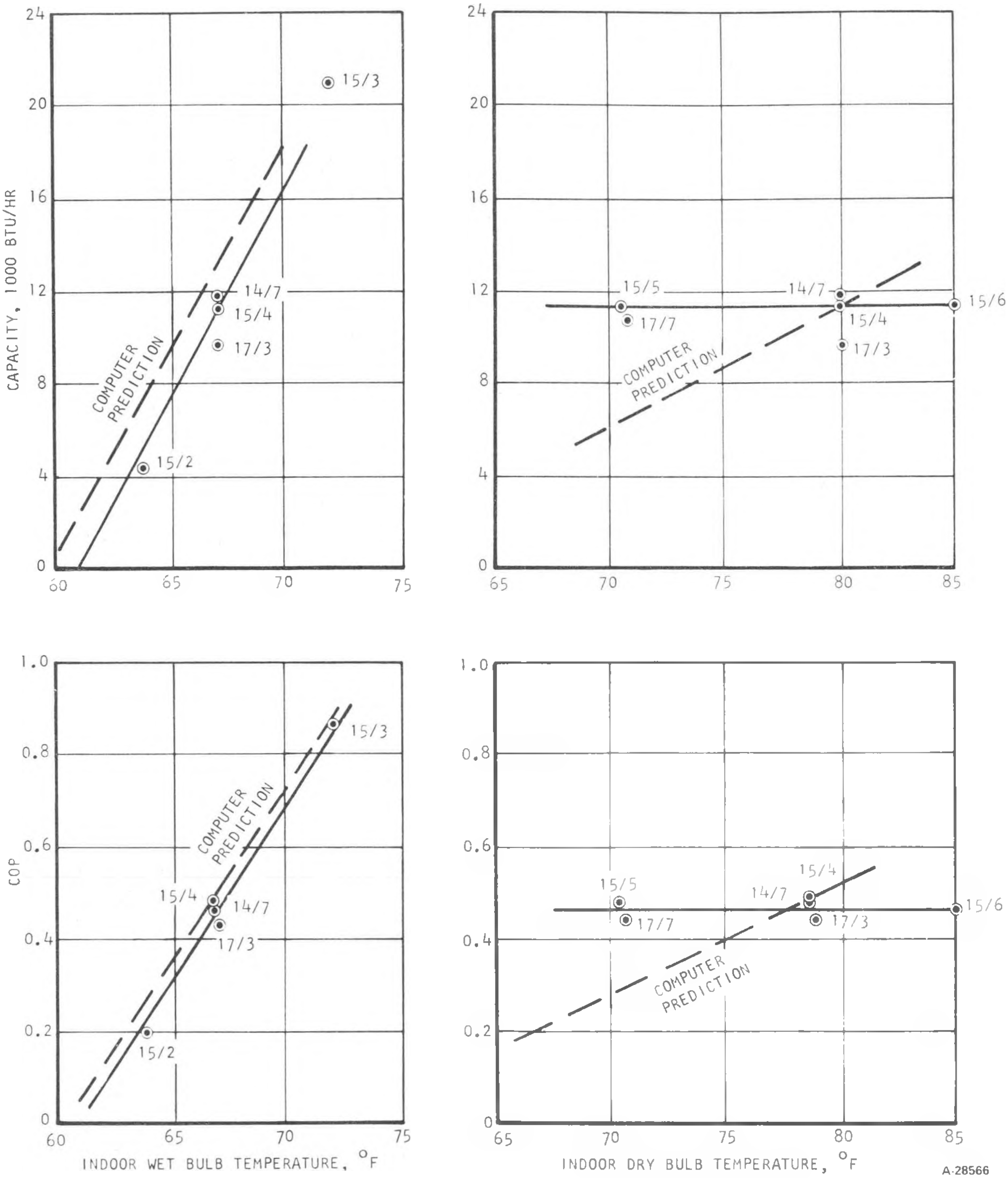


Figure 3-12. Effect of Indoor Temperatures on Ventilated Configuration, Full Flow

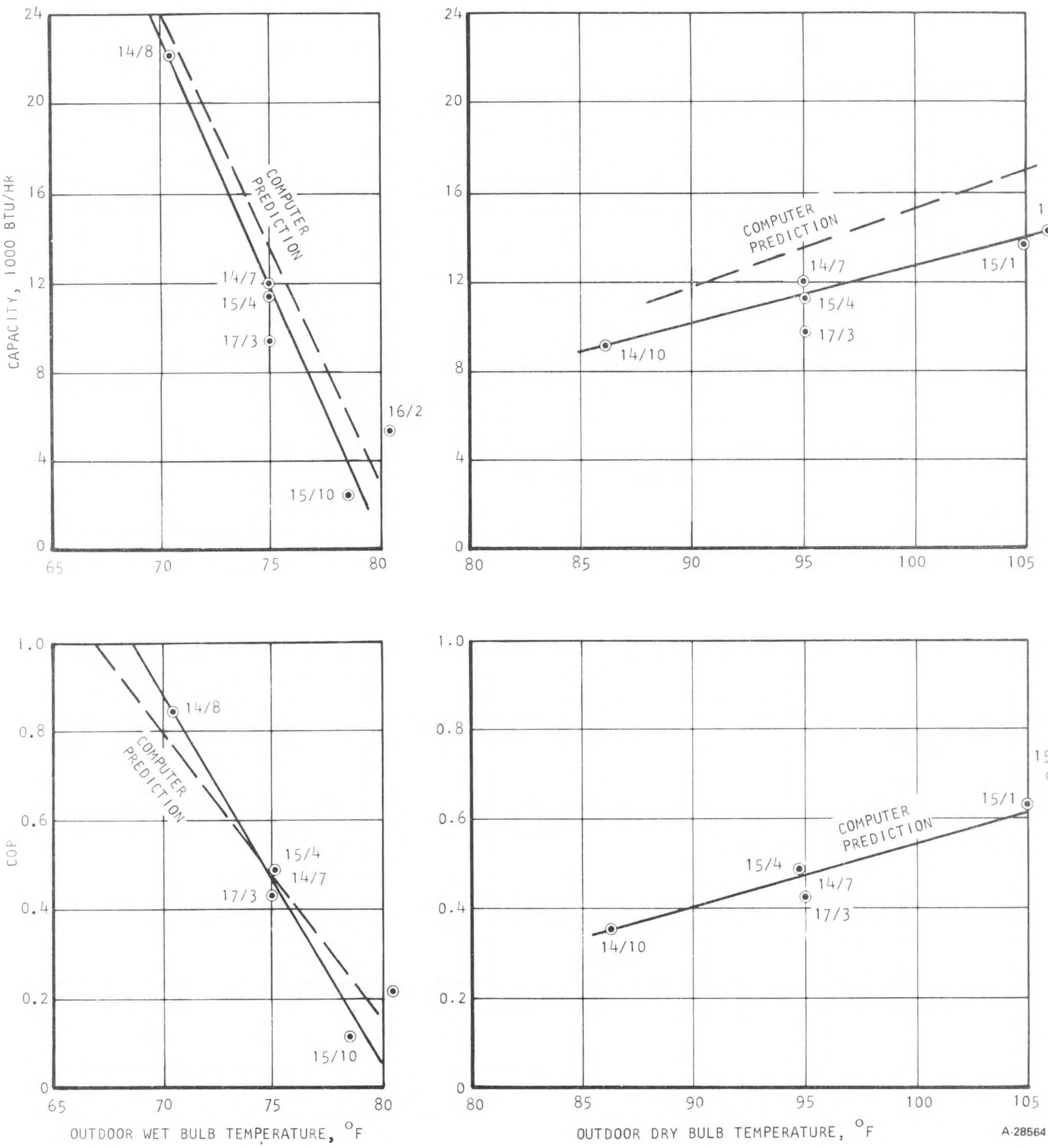


Figure 3-13. Effect of Outdoor Temperatures on Ventilated Configuration, Full Flow

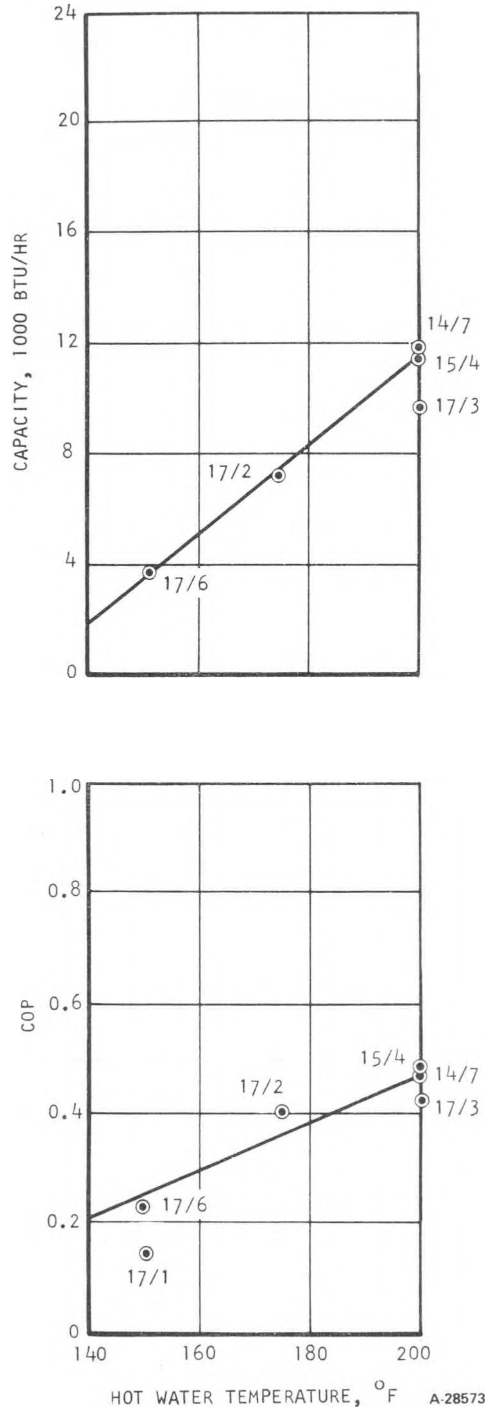


Figure 3-14. Effect of Hot Water Temperatures on Ventilated Configuration, Full Flow

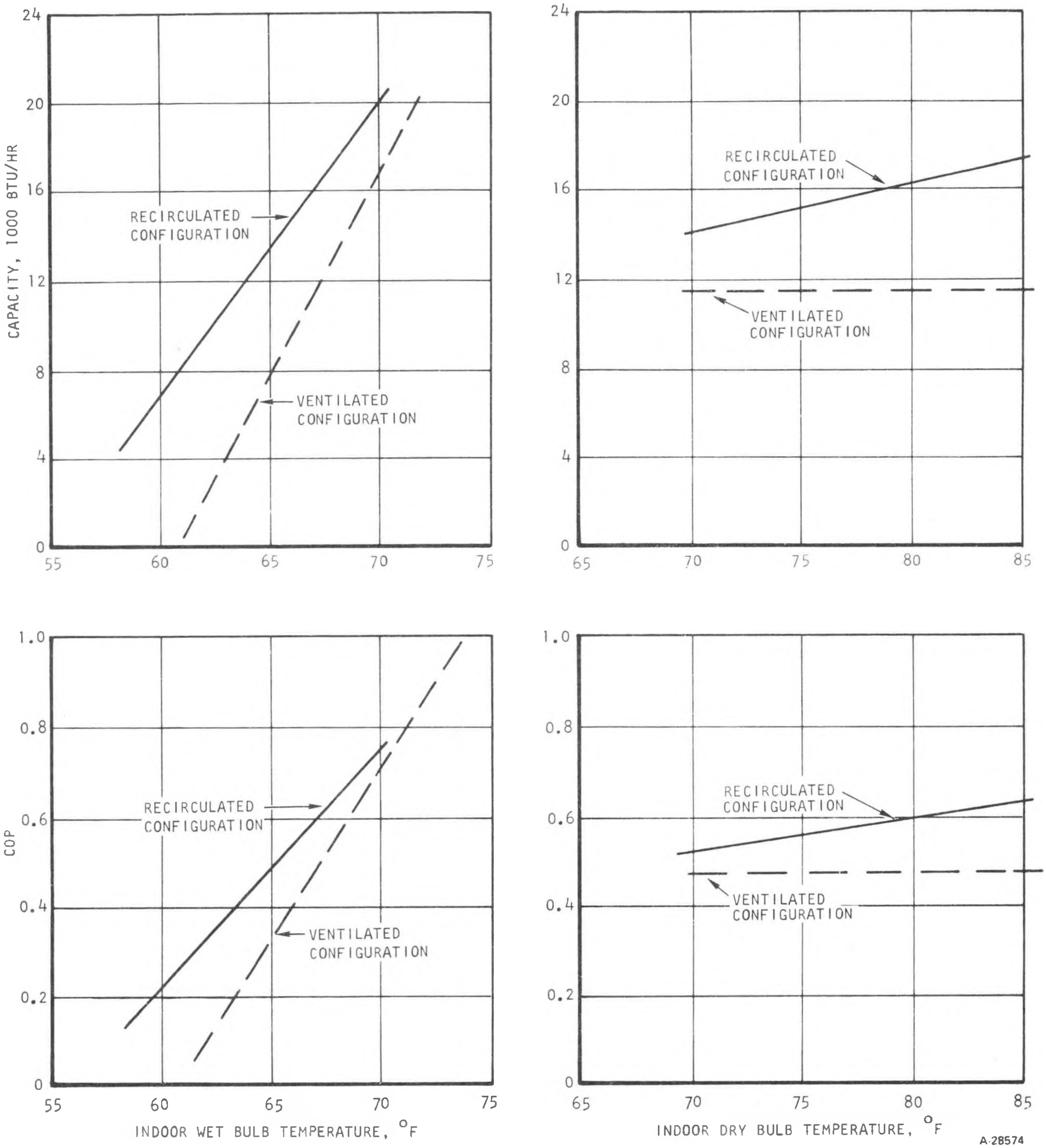


Figure 3-15. Comparison of Recirculated and Ventilated Configuration, Full Flow

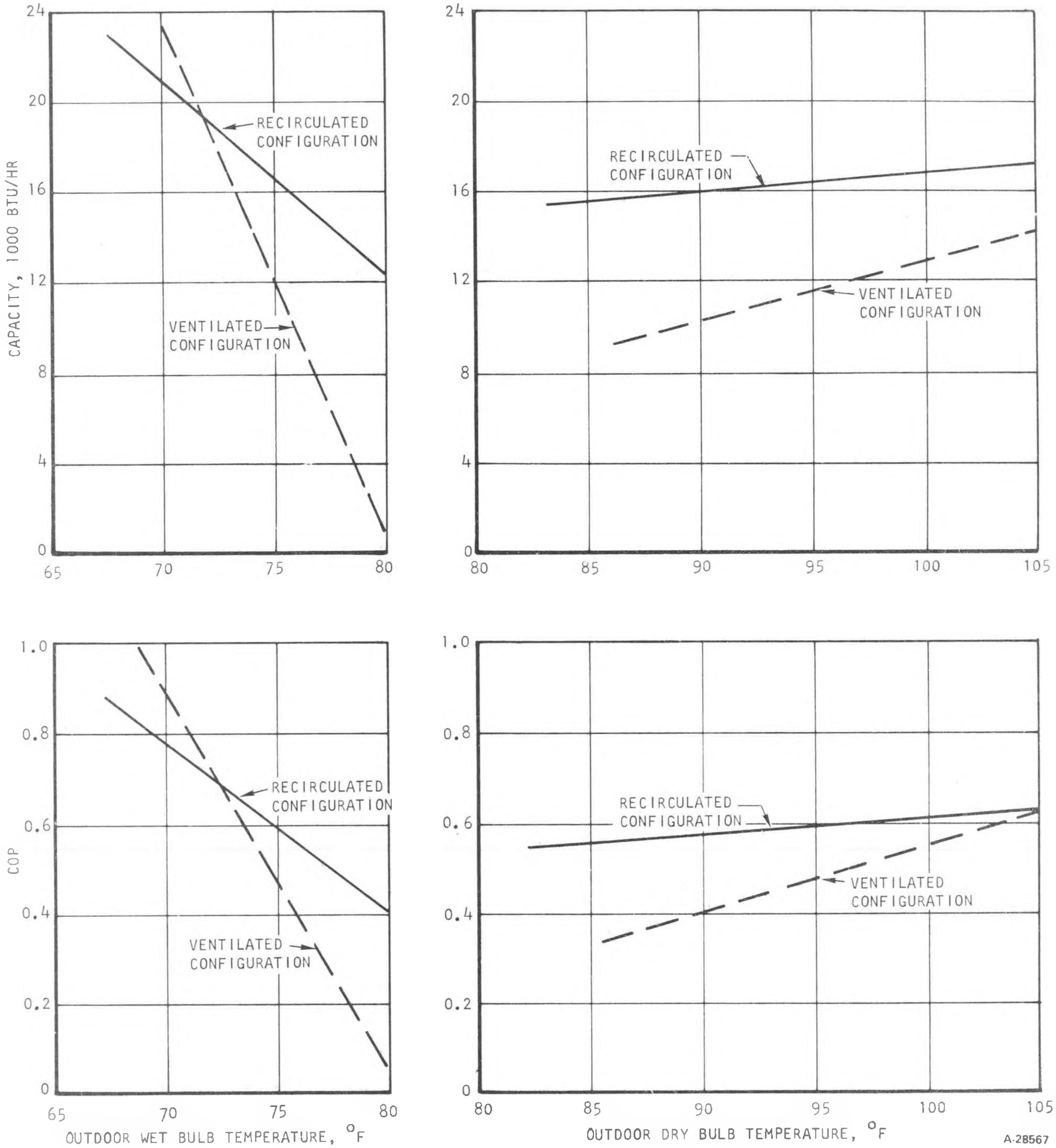


Figure 3-15. Continued

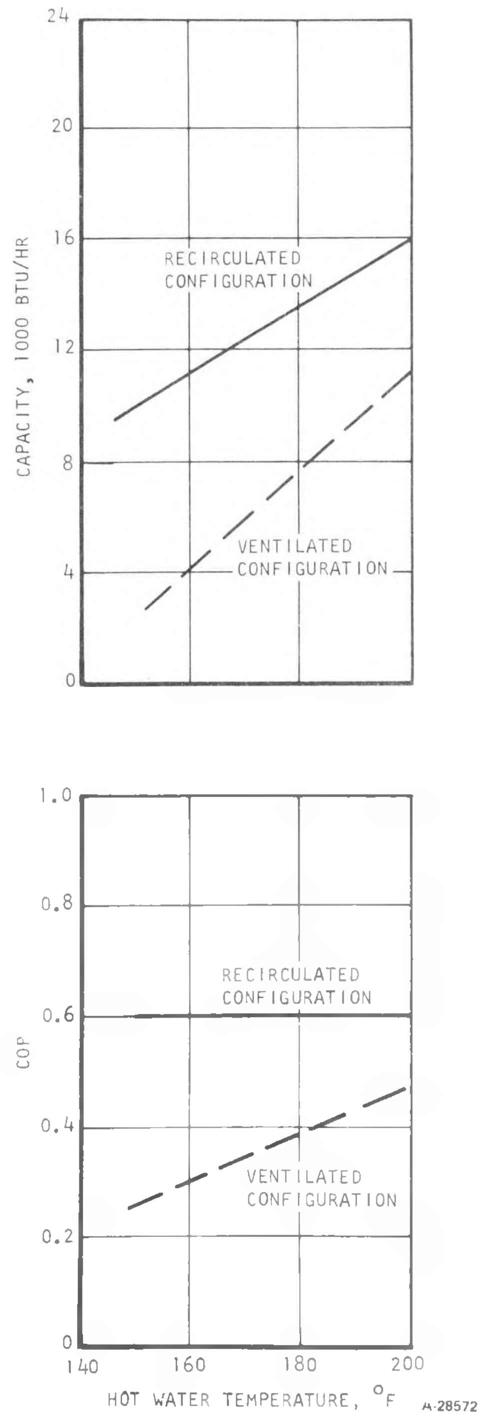


Figure 3-15. Continued

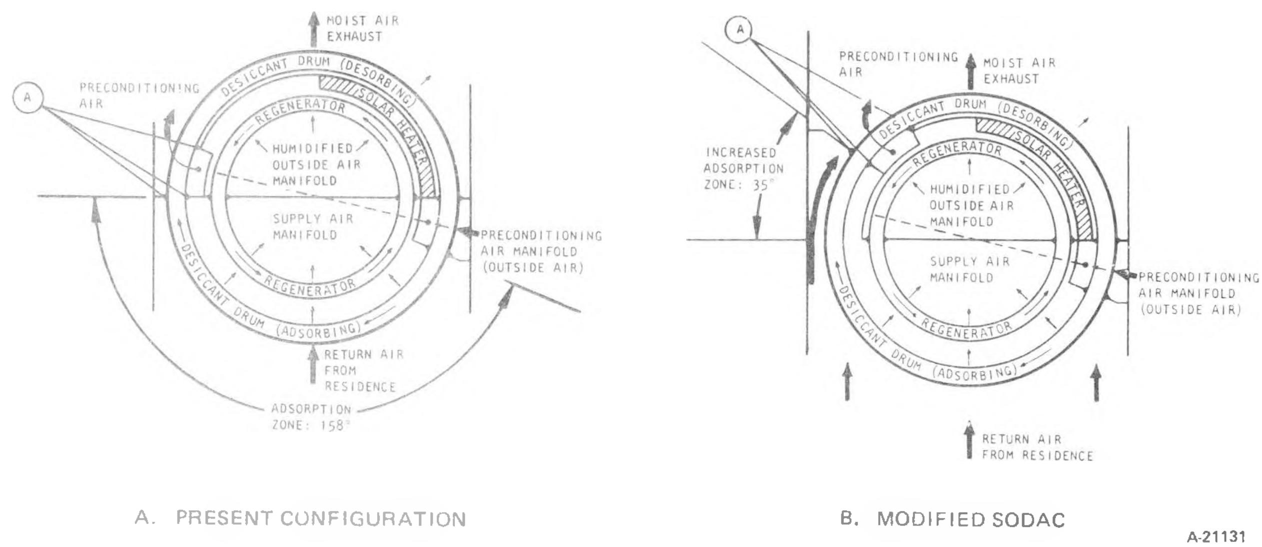


Figure 3-16. Seal Relocation

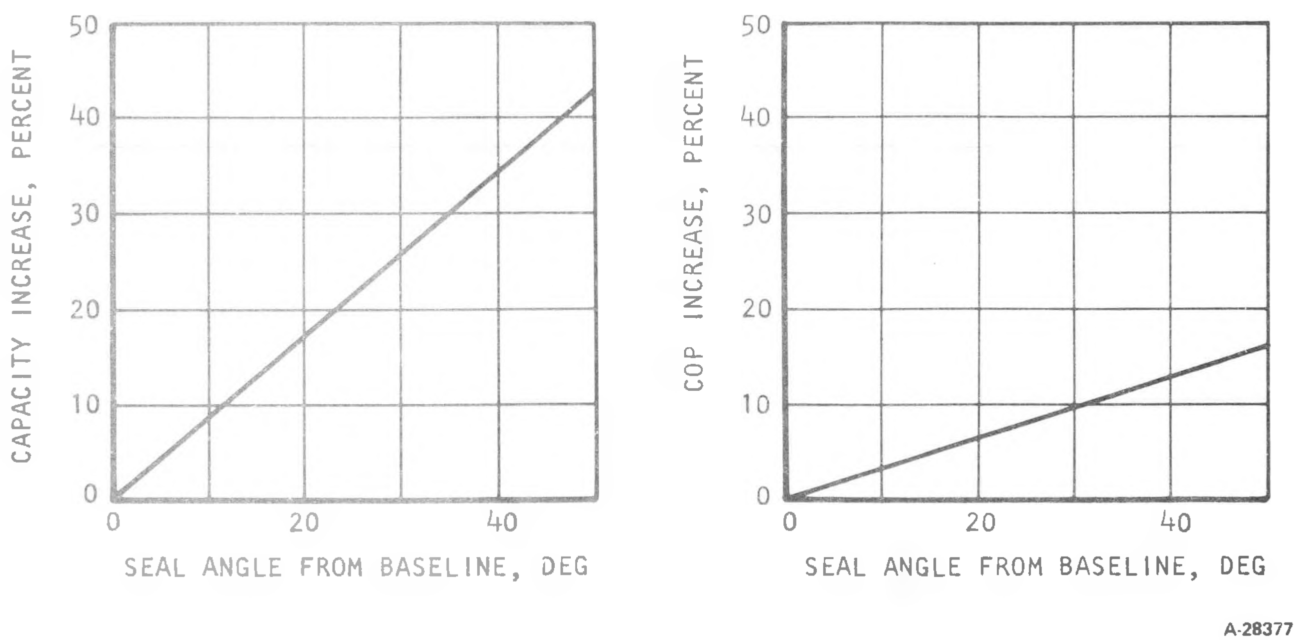


Figure 3-17. Estimated Performance Improvements

SECTION 4

CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

The comprehensive test program conducted on the SODAC has been successfully completed. The data obtained has demonstrated the accuracy of the analytical tools used earlier in the evaluation of the system, and thus validates the conclusions reached relative to the economic viability of the system.

A number of conclusions can be reached as a result of the SODAC test program. These conclusions are underlined and briefly discussed below. For ease of reference, pertinent performance data at the standard ARI rating conditions are listed below:

Capacity:	1.35 ton
COP:	0.60
Indoor airflow:	760 cfm
Outdoor airflow:	925 cfm
Parasitic power:	800 watts (estimated)

Performance of the SODAC in the recirculated configuration is significantly higher than in the ventilated configuration. The SODAC was designed for recirculation of space conditioned air. Operation in the ventilated configuration results in approximately 15°F higher sorbent bed temperatures. Since the silica gel desiccant capacity is very sensitive to temperature, the performance of the entire system is reduced accordingly. This lower performance prevails over the entire range of indoor and outdoor dry and wet bulb temperatures, and also over the 150° to 200°F desorption water temperatures investigated. Operation in the ventilated mode is desirable for certain applications, such as a restaurant. The use of molecular sieve rather than silica gel may obviate this problem, but will result in a higher system cost.

Predicted capacity was achieved and predicted COP was exceeded by 10 to 15 percent. Although a number of assumptions were made in the development of the system computer program, the test data demonstrate the ability of the computer program to predict the behavior of the system. Some of the assumptions made relative to mass transfer data were somewhat conservative, and seem to make up for performance losses due to internal air leakage and heat leaks within the SODAC and to the surrounding environment.

The wiper-type seals with dacron felt pads developed under this program were very effective in minimizing leakage. This type of seal appears to have a life adequate for its intended purpose. The SODAC wiper-type seals and partitions within the sorbent bed and rotary regenerator were developed as a result of the Phase I development program. Incorporation of these seals and partition was done by rework of the existing hardware. The alignment of the seals and the distance between the partitions within the desiccant bed could not be accurately controlled. As a result, leakage could not be eliminated entirely. However, with proper care taken in the design of a machine designed for production, it is believed that the use of this type of seals could be used to effectively eliminate leakage.

The capacity and COP of the SODAC could be improved significantly by eliminating leakage. Analysis of the test data shows a consistent negative heat balance between the heat input to the SODAC (cooling load and hot water load) and the heat output from the SODAC (heat dumped to the outdoor process airstream). The ratio of heat input to heat output is on the order of 0.90. This is indicative of internal leakages. A model was developed to determine the effect of leakage on performance using heat balance data as a basis. Using this model, it is estimated that by eliminating leakage, the capacity and COP of the system could be increased by about 20 percent.

The capacity and COP of the SODAC can be improved further by increasing the size of the adsorption zone. An attempt was made to demonstrate this effect; contract limitations did not permit successful experimental demonstration. However, computer investigations show a potential increase of 30 percent in capacity and 12 percent in COP by increasing the adsorption zone by 22 percent. This involves moving two of the seals so as to better utilize the entire periphery of the desiccant drum.

The high process airflow rates (760 cfm) necessary to achieve reasonable capacity (1.35 tons) would be reduced considerably at the higher capacities, which could be obtained by incorporation of the modifications mentioned above. The maximum 400-cfm/ton currently specified for rating air conditioners may be achievable. However, it is doubtful that return air wet bulb temperatures lower than about 58°F could ever be achieved with this type of system. This is considerably higher than normally obtained with vapor compression and absorption air conditioners.

The coefficient of performance obtained with the SODAC as presently configured is comparable to that attainable with a small absorption cooling system. Further development should yield significant improvements with COP's (on the order of 0.8, which is well within reach). Commercialization studies conducted in Phase I show that a SODAC designed for high production could be sold at much lower cost than a comparable absorption machine. The analysis conducted earlier indicated that the SODAC could be an economically viable option as an add-on to an economical solar heating system. These conclusions are still valid, especially with the higher performance that could be achieved by redesign of this first SODAC development prototype.

RECOMMENDATIONS

The SODAC development program was very successful in demonstrating hardware approaches and in identifying areas for further improvements. Further testing of this first development unit is not recommended. Rather, future work should be conducted on a preproduction prototype incorporating the following features.

(a) Silica gel beads rather than granular silica gel

(b) Wiper-type seals with dacron felt pads and bed partitions

(c) An extended adsorption zone

(d) A drum drive designed specifically for the desiccant system

(e) A plastic screen regenerator matrix

(f) Frames and drums designed to minimize leakage

By proper seal-partition design and also by extending the adsorption zone, COP's of 0.8 could be achieved and considerably higher capacity could be derived from a unit of the size of the present development unit. With this improved capacity, process flow rates (cfm/ton) and parasitic power will be reduced to acceptable levels.

Although the SODAC offers advantages by comparison to many desiccant systems and also by comparison to absorption systems, development is not recommended at this time because of the availability of other novel sorbent approaches which have demonstrated potential performance that is significantly better. Some of these novel sorbent systems have demonstrated COP's in excess of 1.0 at standard ARI conditions. This high performance was achieved as a result of breakthroughs in sorbent technology, whereby the heat and mass transfer properties of the sorbent are matched in an optimum fashion to achieve higher loading and minimum heat input for desorption.

SECTION 5

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