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# DEVELOPMENT OF A SOLAR-POWERED RESIDENTIAL AIR CONDITIONER

CONTRACT NAS8-30758

## FINAL SUMMARY REPORT

75-12098

NOVEMBER 28, 1975



(NASA-CR-144234) DEVELOPMENT OF A  
SOLAR-POWERED RESIDENTIAL AIR CONDITIONER  
Final Summary Report (AiResearch Mfg. Co.,  
Torrance, Calif.) 146 p HC \$6.00 CSCL 10A

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AIRESEARCH MANUFACTURING COMPANY  
OF CALIFORNIA

S-99821

# **DEVELOPMENT OF A SOLAR-POWERED RESIDENTIAL AIR CONDITIONER**

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OF CALIFORNIA**

**S-29821**

## INTRODUCTION

This report summarizes the work performed by AiResearch Manufacturing Company for NASA Marshall Space Flight Center under Contract NAS8-30758. The program is a 13-month study of solar-powered air conditioners designed specifically for residential application. The contract was sponsored by the Life Support and Environmental Branch of MSFC; David Clark was the program technical monitor. At AiResearch, the program was under the direction of J. Rousseau; acknowledgement is made to Dr. K. C. Hwang who developed the various computer programs used in the analyses.

Initially the program was concerned solely with the development of an optimum Rankine-cycle mechanical refrigeration system. However, feasibility investigations of the adsorption process revealed that a desiccant-type air conditioner offered many significant advantages. As a result, limited efforts were expended toward the optimization of such a system.

The material presented in this report covers the following topics:

- (1) The program itself
- (2) A summary of the work and data generated under the various program tasks
- (3) Conclusions and recommendations

An overall outline of the report is presented below.

# INTRODUCTION

## PROGRAM DEFINITION

## PROGRAM RESULTS

- RANKINE-CYCLE AIR CONDITIONER
- DESICCANT AIR CONDITIONER

## CONCLUSIONS AND RECOMMENDATIONS



## PROGRAM DEFINITION

An overview of the program is presented in the following pages, where the program is defined in terms of its objectives, approach, and schedule. A listing of the documents furnished to NASA under contract NAS8-30758 is also given.

# PROGRAM DEFINITION

- OBJECTIVES
- APPROACH
- SCHEDULE
- DOCUMENTATION



## PROGRAM OBJECTIVES

The temperature level of the energy source available from the solar collector is about 200°F; the heat sink level necessary for operation of the heat machine is about 95°F. Under these conditions, the maximum efficiency attainable as expressed by Carnot cycle efficiency is about 15 percent. Actually, a Carnot cycle efficiency of approximately 10 to 12 percent is more realistic, if the temperature differences necessary for heat exchange at the source and the sink are taken into account. It is apparent that the actual efficiency of an air conditioner designed for operation within these temperature levels will be low and will require large quantities of source heat. This translates into large solar collector and thermal energy storage devices.

For this reason, in order to achieve maximum utility from such a system, it is necessary to maximize the effectiveness of all system components and configure the system to minimize cost of ownership. This was the overall goal of the study program. Specific program objectives are listed on the opposite page.

The optimization of the Rankine-cycle air conditioning system in terms of cost and performance requires trade studies at the overall system level, including the solar collector. Previous economic studies concerned with solar thermal energy have shown that generally the cost of the solar collector is by far the largest factor in determining the selection of a cost-effective system. Therefore, Rankine air conditioning system cost data have been generated in a parametric fashion and were used for overall system optimization.

A minor portion of the total program effort has been directed toward the optimization of a desiccant air conditioner in terms of desiccant bed size and operating parameters. Parametric data relating performance to ambient conditions and internal subsystem parameters were generated to permit the preparation of a preliminary specification.

# PROGRAM OBJECTIVES

- OPTIMIZATION OF RANKINE-POWERED AND DESICCANT AIR CONDITIONERS DESIGNED SPECIFICALLY FOR OPERATION USING THERMAL ENERGY FROM A FLAT PLATE SOLAR COLLECTOR
- CHARACTERIZATION OF THE OPTIMUM RANKINE-CYCLE SYSTEM IN TERMS OF COST AND PERFORMANCE TO PERMIT OVERALL SYSTEM TRADE STUDIES
- DEVELOPMENT OF PRELIMINARY SPECIFICATIONS



S-94775-A

## PROGRAM LOGIC

Task 1, candidate system classification, involves a state-of-art survey covering the absorption and adsorption processes as well as the Rankine-cycle air conditioner. As part of this task, a 3-ton Lithium Bromide/water (LiBr/H<sub>2</sub>O) absorption system was characterized in terms of performance and cost; these data established a baseline for comparison. Data on systems and equipment applicable to the Rankine air conditioner were collected to establish a data bank for performance characterization of the system component. In the performance of Task 1, preliminary data indicated that the desiccant process could offer many advantages if applied to a solar-powered air conditioner. Evaluation of this approach was included as part of Task 1 to determine feasibility.

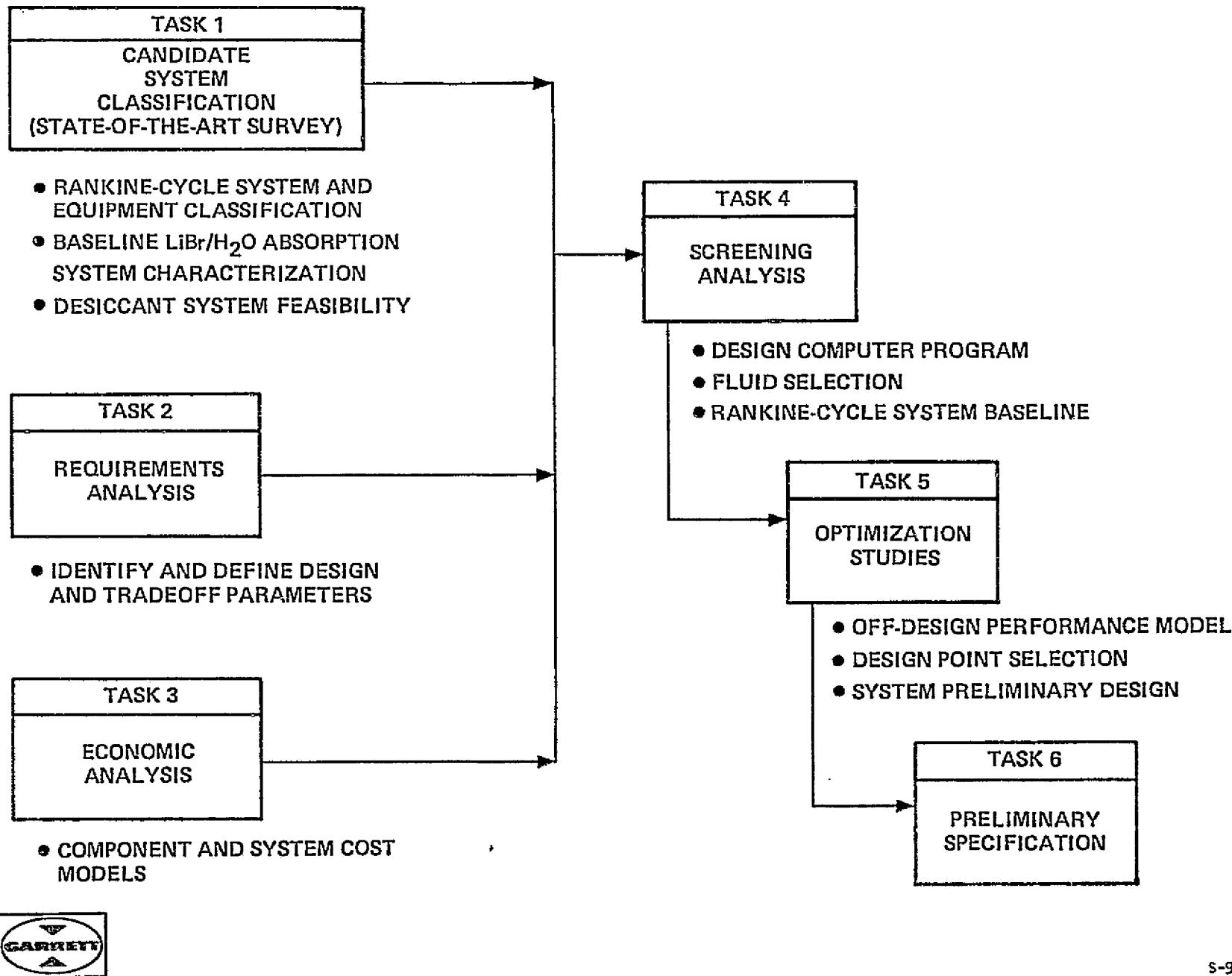
Task 2, requirements analysis, was concerned with the definition of (1) parameters involved in the optimization procedure, and (2) conditions for the design and evaluation of competing system approaches.

Under Task 3, economic analysis, system and equipment cost data were collected and correlated. Models were developed to permit characterization of candidate air conditioning systems in terms of cost.

The results of Tasks 1 through 3 were used in a screening analysis. Task 4 was aimed at selection of an optimum Rankine-cycle air conditioner configuration. Because of the number of interface and internal system parameters involved, a computer program was developed to mechanize system design calculations. Fluid selection studies were performed. The computer program initially developed as a cycle analysis program was modified into a system design program by incorporating component performance and cost models.

Any solar-powered system will require augmentation for operation during periods when solar thermal energy is not available. The selection of a technique for system augmentation requires off-design performance evaluation; this was done under Task 5. A second computer program describing the off-design performance of the Rankine system was developed. This program was used with a solar collector-thermal storage system to determine overall system performance and auxiliary power usage. System performance data were used to generate component problem statements for the preliminary specifications.

# PROGRAM LOGIC



S-94779-A

## PROGRAM SCHEDULE

The performance period for the contract was 13 months. Task 1 was extended for the duration of the program to permit coverage of ongoing study and hardware programs directly related to the development of solar-powered air conditioners.

The requirements analysis was performed early to provide data for the activities of Tasks 3 and 4. Additional requirements data were obtained recently from the NASA solar house to permit evaluation of the system at off design.

The economic analysis yielded cost models that were incorporated in the design computer program of Task 4. The screening analyses were completed with the definition of the baseline system in August. Finally, the optimization studies of Task 5 concerned with the Rankine cycle air conditioner extended through the middle of November. Data from Task 5 were used in the preparation of the system specifications.

Documentation was furnished as shown. Task reports were submitted to summarize results at appropriate intervals. This constitutes the final summary report under the contract.

# PROGRAM SCHEDULE

TASK DESCRIPTION	MONTHS												
	NOV 1	DEC 2	JAN 3	FEB 4	MAR 5	APR 6	MAY 7	JUN 8	JUL 9	AUG 10	SEP 11	OCT 12	NOV 13
TASK 1. CANDIDATE SYSTEM CLASSIFICATION													
TASK 2. REQUIREMENTS ANALYSIS													
TASK 3. ECONOMIC ANALYSIS													
TASK 4. SCREENING ANALYSIS													
TASK 5. OPTIMIZATION STUDIES													
TASK 6. PRELIMINARY SPECIFICATIONS													
DOCUMENTATION													
MONTHLY PROGRESS REPORTS													
TASK REPORTS													
PROGRAM REVIEWS													
FINAL REPORT													

S-94778 -A



## PROGRAM DOCUMENTATION

A number of reports were published throughout the contract period to summarize study results and document particular areas of investigation. These reports are listed below. This summary report contains the findings of the study leading to the development of the schematics and specifications presented later.

# PROGRAM DOCUMENTATION

<u>DOCUMENT</u>	<u>DATE</u>	<u>AIRESEARCH REPORT NO.</u>
PROGRAM PLAN	NOVEMBER 14, 1974	74-10996(1)
DESIGN REQUIREMENTS AND TRADE-OFF PARAMETERS	NOVEMBER 22, 1974	74-10996(2)
STATE-OF-THE-ART SURVEY	JANUARY 13, 1975	74-10996(3)
ECONOMIC ANALYSIS	MARCH 28, 1975	74-10996(4)
PROGRAM REVIEW	APRIL 8, 1975	74-10996(5)
FEASIBILITY STUDY – SOLAR-POWERED AIR CONDITIONER USING THE ADSORPTION PROCESS	MAY 30, 1975	74-10996(6)
SCREENING ANALYSIS	JULY 25, 1975	74-10996(7)
OPTIMIZATION STUDIES	NOVEMBER 7, 1975	74-10996(8)
MONTHLY PROGRESS REPORTS (12)		74-11021(1) THROUGH (12)



S-99814

## PROGRAM RESULTS

### RANKINE-CYCLE AIR CONDITIONER

A summary of the investigations conducted on the Rankine-cycle air conditioner is presented. The data are arranged according to the six program tasks described previously. Data on the desiccant air conditioner are given later. The state-of-art survey of Task 1 covers the three types of systems currently considered: absorption, Rankine, and desiccant air conditioners. The data compiled on the LiBr/H<sub>2</sub>O absorption system were used as a basis for comparison since this approach represents the state of the art in air conditioners powered by low-grade thermal energy.

# PROGRAM RESULTS

## RANKINE-CYCLE AIR CONDITIONER

STATE-OF-THE-ART REVIEW (TASK 1)

REQUIREMENTS ANALYSIS (TASK 2)

ECONOMIC ANALYSIS (TASK 3)

SCREENING ANALYSIS (TASK 4)

SYSTEM OPTIMIZATION (TASK 5)

SPECIFICATION (TASK 6)

DESICCANT AIR CONDITIONER (TASK 1)



S-99815

STATE-OF-ART REVIEW (TASK 1)

This report section briefly summarizes the data collected under the state-of-art review. These data are concerned with the following:

- (1) The LiBr/H<sub>2</sub>O absorption system
- (2) The Rankine-powered air conditioner
- (3) Air conditioning by the adsorption process

# STATE-OF-ART REVIEW (TASK 1)

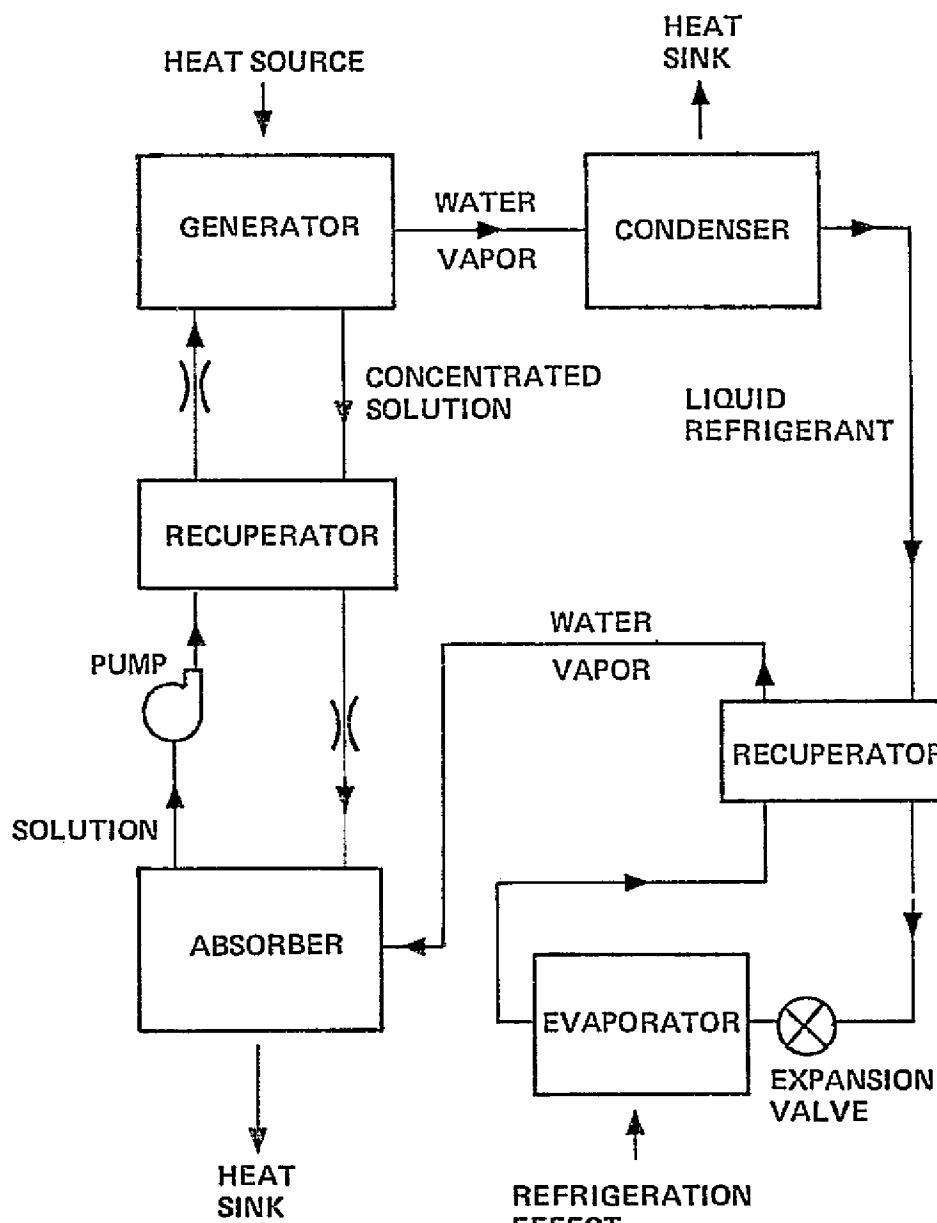


## LiBr/H<sub>2</sub>O ABSORPTION CYCLE

A simple schematic of a LiBr/H<sub>2</sub>O absorption air conditioner is shown. The system arrangement corresponds roughly to the Arkla unit installed in the NASA solar house. The original Arkla unit featured a gas-fired generator, water-cooled absorber/condenser, and a thermosyphon solution pump. This was modified by substituting a water-fired generator for use with a low-temperature collector; this version of the Arkla unit has been used in a number of solar house programs throughout the country, primarily because it represents the only available hardware directly applicable to the use of low-grade thermal energy for air conditioning.

Currently Arkla is under contract to NSF to further modify its basic water-fired LiBr/H<sub>2</sub>O unit by incorporation of (1) a mechanical pump (as shown on the schematic), and (2) an evaporative condenser. Both modifications may improve the performance of the system in comparison with the version in the NASA solar house. The solution pump will eliminate the increase in solution boiling point due to the thermosyphon pumping technique currently used. Further, the evaporative condenser/absorber will result in slightly lower temperatures due to the elimination of the intermediate water transport loop between the cooling tower and the air conditioner.

# LiBr/H<sub>2</sub>O ABSORPTION CYCLE



5-94783

## BASELINE LiBr/H<sub>2</sub>O SYSTEM CHARACTERISTICS

The data listed below were taken from Arkla proposal to NSF covering the modification of an existing water chiller design (not currently marketed) to incorporate (1) a water-fired generator, (2) a LiBr/H<sub>2</sub>O solution pump, and (3) an evaporative condenser-absorber. The estimated capacity of the unit is 3 tons under the conditions listed below.

The coefficient of performance of the unit is estimated at 0.65 with a water temperature at generator inlet of 195°F and a wet bulb ambient of 78°F. In addition, the unit will consume 875 watts of parasitic power for fans, pump, and controls. No test data are available on this unit at this time. These COP data are consistent with data obtained on the NASA solar house unit. However, analysis of the test data obtained on the NASA unit show that with generator water temperatures of approximately 195°F and cooling water temperatures of about 80°F the test unit capacity was generally between 2 and 2.5 tons.

The cost of the LiBr/H<sub>2</sub>O 3-ton unit modified for solar operation is not available at this time. Data obtained from Arkla indicate that such units would be marketed at about the same price as current water chillers. An installed cost of \$2700 for a 3-ton unit is considerably higher than the cost of a conventional vapor compression air conditioner.

# BASELINE LiBr/H<sub>2</sub>O SYSTEM CHARACTERISTICS

COOLING CAPACITY	3 TONS
HOT WATER SOURCE TEMPERATURE	195° F IN/185° F OUT
CHILLED WATER TEMPERATURE	55° F IN/45° F OUT
WATER CONSUMPTION	24 GAL./HR
EVAPORATIVE HEAT REJECTION	78° F WB AIR IN
COEFFICIENT OF PERFORMANCE	0.65
ELECTRICAL CONSUMPTION	875 WATTS MAX. (EXCLUDING CIRCULATION FAN)
SYSTEM COST	\$2700 (INSTALLED COST)



S-94777

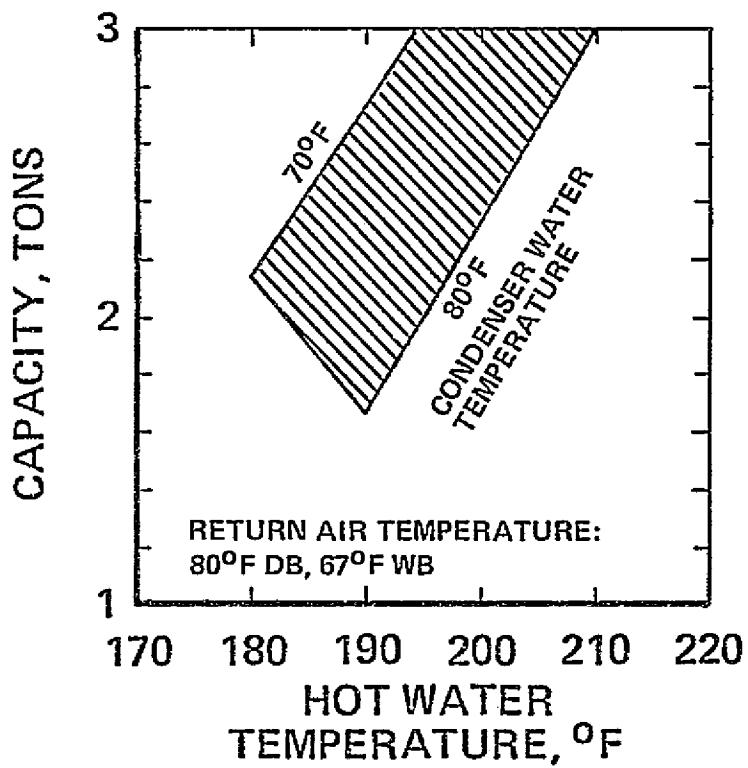
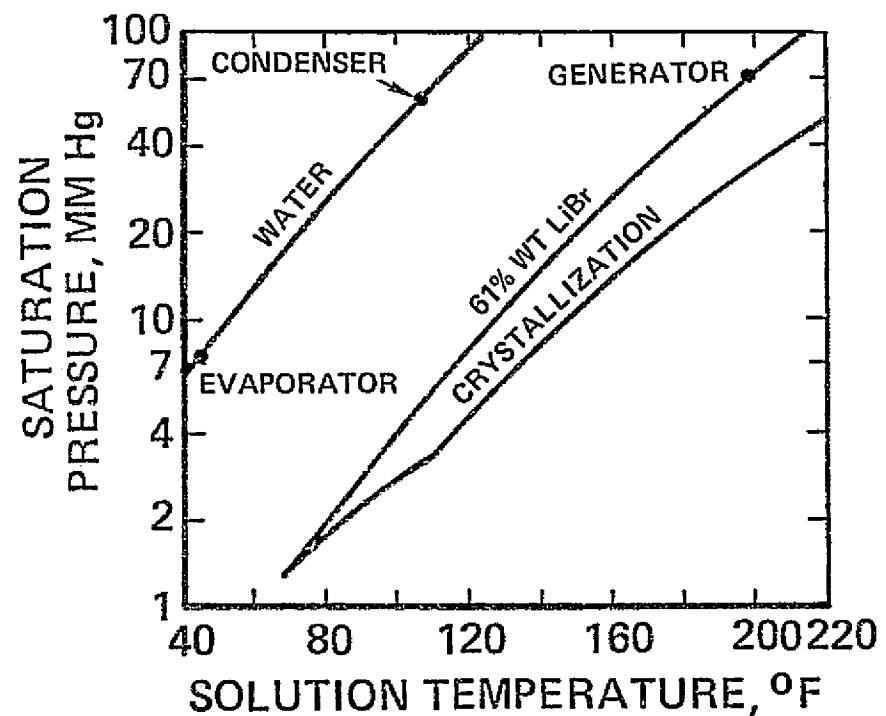
## LiBr/H<sub>2</sub>O SYSTEM OFF-DESIGN PERFORMANCE

Off-design performance of a typical LiBr/H<sub>2</sub>O absorption air conditioner is plotted in terms of the temperature levels of the heat source and heat sink. This plot is reproduced from data published by Arkla for the Solaire unit (Model 501-WF). The unit is designed for 100 percent capacity at a source temperature of 210°F and a cooling water sink temperature of 85°F. As the source temperature drops from 210 to 190°F, for example, at constant cooling water temperature, system capacity will drop to about 55 percent of design capacity.

This significant drop in cooling capacity is due primarily to a reduction in the water pressure in the generator; this in turn corresponds to a rapid decrease in condenser capacity. This effect is shown on the vapor pressure plot shown on the left side of the accompanying figure. It is important to note here that the temperatures on this vapor pressure plot are actual working fluid temperatures.

In the actual installation of the NASA solar house, the control system initially defined by Arkla was to maintain the generator water temperature (by addition of auxiliary power) so that the difference between source and sink temperatures would be at least 130°F. This control system was later changed by NASA to control the minimum generator water temperature at about 185°F; cooling tower water temperature was maintained between 70 and 80°F over the operating range.

# LiBr/H<sub>2</sub>O SYSTEM OFF-DESIGN PERFORMANCE



## RANKINE CYCLE SURVEY DATA

A number of organizations are currently involved in analytical and experimental investigations directly related to the application of the Rankine cycle to solar-powered air conditioners. The most significant work found in the literature is summarized below.

Barber Nichols has developed a 3-ton air conditioner for evaluation in the Honeywell solar laboratory. The Rankine power loop (R-113) features an expansion turbine, which drives a conventional R-12 compressor through a gearbox. A motor-generator is used to supplement the Rankine turbine. The turbine is designed to develop 1.87 kw of power at 52,000 rpm; efficiency is 72 percent. An overall system COP of 0.5 was obtained with a collector water temperature of 215°F and a cooling tower water temperature of 85°F. This unit is currently being upgraded to improve COP; modifications involve lower pressure drop piping and substitution of a more efficient motor-generator.

General Electric Company has been engaged in the development of a multivane expander to drive a conventional 3-ton refrigerant compressor. Efficiencies as high as 80 percent have been achieved with R-11 as the working fluid. Test units have been subjected to 1000 hr of development testing. Overall COP's of 0.6 have been reported for this unit.

In 1970, AiResearch delivered to the U.S. Army two heat-powered 5-ton refrigeration units. The turbocompressor is a hermetic unit featuring a two-stage compressor and a single-stage turbine; design speed is 48,000 rpm. Turbine and compressor efficiencies of 80 and 75 percent respectively were achieved on test. This unit could be used to provide 4.5 tons of cooling in typical solar-powered application. Currently, AiResearch is engaged in the development of a 10-ton R-12 centrifugal compressor. The 1.5-in. dia compressor wheel is driven through a magnetic coupling at a speed of 90,000 rpm. Efficiencies of 75 percent have been achieved on test.

United Aircraft Research Laboratories is engaged in a turbocompressor feasibility demonstration program. The test unit is a modified turbocompressor designed for 8 tons of cooling with R-114 and gas turbine exhaust as the heat source. Matching problems in the solar powered application has resulted in relatively low efficiencies.

Thermo Electron Corporation has developed Rankine systems using reciprocating machinery since 1963. Using this basic technology, the characteristics of a solar-powered system were estimated, and turbine and compressor efficiencies of 72 percent were claimed to be consistent with data obtained on similar machines.

# RANKINE CYCLE SURVEY DATA

● BARBER NICHOLS

3-TON SOLAR POWER RANKINE-DRIVEN AIR CONDITIONER  
SYSTEM COP = 0.5 WITH SOURCE TEMPERATURE AT 215°F  
AND SINK TEMPERATURE AT 85°F

● GENERAL ELECTRIC

MULTIVANE EXPANDER DEVELOPMENT  
EXPANDER EFFICIENCIES OF 80 PERCENT OBTAINED ON  
TEST WITH R-11

● AIRESERACH

WASTE HEAT TURBOCOMPRESSOR AIR CONDITIONER  
DEVELOPED FOR ARMY; DEMONSTRATED TURBINE AND  
COMPRESSOR EFFICIENCIES OF 80 AND 75 PERCENT  
RESPECTIVELY

10-TON R-12 CENTRIFUGAL COMPRESSOR UNDER DEVELOP-  
MENT; 75 PERCENT EFFICIENCY AT 90,000 RPM OBTAINED  
PRESENT STUDY PROGRAM

● UNITED AIRCRAFT  
RESEARCH LABORATORIES

TURBOCOMPRESSOR FEASIBILITY DEMONSTRATION;  
ESTIMATED EFFICIENCIES OF 62 AND 75 PERCENT FOR  
COMPRESSOR AND TURBINE RESPECTIVELY

● THERMOELECTRON CORP.

RECIPROCATING MACHINERY APPLICABLE TO SOLAR-  
POWERED RANKINE AIR CONDITIONER; COMPRESSOR AND  
TURBINE EFFICIENCIES OF 72 PERCENT ESTIMATED  
WITH R-22



S-94770-A

## DESICCANT AIR CONDITIONER

A simplified schematic of a desiccant air conditioner is depicted showing the principle of operation.

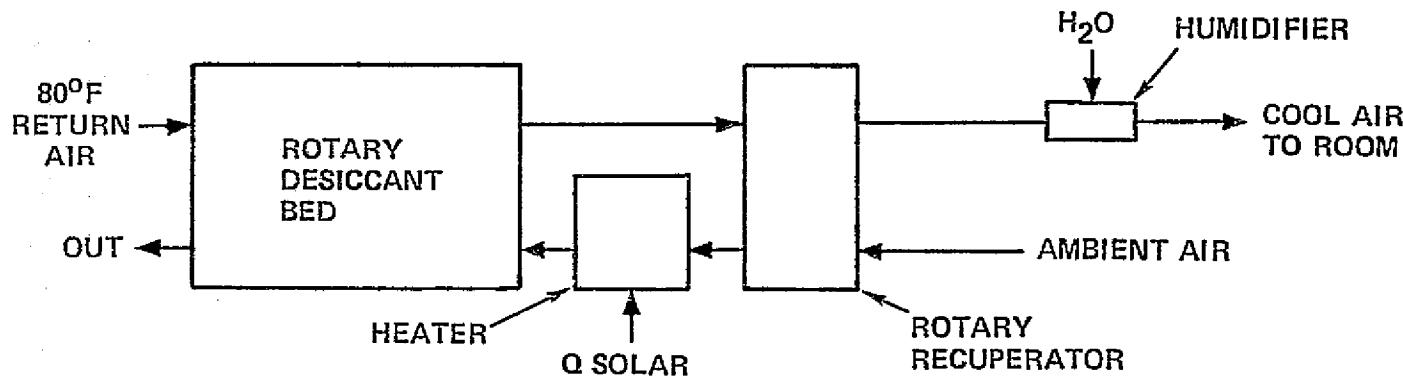
Hot humid air from the room is dried adiabatically in a rotary desiccant bed. Heat is released in the process, and the hot air is cooled in a rotary recuperator. Further cooling is effected by humidifying this dry air stream prior to ducting it to the room. After humidification, the dry and wet bulb temperatures of this air stream are satisfactory for air conditioning purposes.

Ambient air flowing through the unit in the opposite direction is used as the heat sink in the rotary recuperator. (The dry bulb of this air also can be reduced by humidification or recuperation upstream of the rotary recuperator.) The ambient air is further heated in a fixed boundary heat exchanger and used to regenerate the desiccant bed.

Several variations of this basic arrangement are possible, depending on the degree of regeneration and humidification of the cooling stream. Data reported in the literature indicate that a coefficient of performance of 0.7 to 0.8 can be achieved using silica gel as the desiccant with a heat source temperature of 200°F. This compares favorably with the COP's attainable with the absorption and Rankine cycles.

The adsorption process offers a number of significant advantages compared to the other two candidate approaches. In the light of these advantages, feasibility investigations of the adsorption process were conducted. AiResearch has developed, under prior space life support activities, a comprehensive bank of sorbent data and analytical computer routines which were used in the performance of this task.

# DESICCANT AIR CONDITIONER



## ADVANTAGES

1. AIR IS WORKING FLUID
2. USES HEAT DIRECTLY
3. POTENTIAL FOR AIR COLLECTOR
4. COOLING MODE COP'S OF 0.7-0.8 AT MAX. TEMP OF 180°F



Considerable work has been expended by The Institute of Gas Technology (IGT)\* aimed at the development of a gas-fired desiccant air conditioner. This basic system has been modified for integration with a solar collector system to realize the advantages of thermal solar energy in significantly reducing gas consumption. The heart of the system is a rotary desiccant wheel fabricated of corrugated molecular sieve on asbestos paper. This solar MEC system, as it is called by IGT, is currently under field test at various locations in this country. The basic capacity of the unit has been measured at 2.5 tons under test; and overall thermal COP's as high as 0.73 are reported at desorption temperatures of 290°F. Under these conditions approximately 65 percent of the desorption thermal energy is from a solar source.

NSF is currently funding a preliminary design study of a desiccant air conditioner using silica gel as the sorbent. This program is conducted by The Center for the Environment and Man, Inc.\*\* In its present configuration, this system utilizes four rotating beds of silica gel separated by fixed boundary heat exchangers (hot and cold) to approach isothermal bed operation in the adsorption and desorption modes. In addition, the system features two regenerative heat exchangers in the ambient air and recirculated airstream. The baseline system is designed to provide 2 tons of air conditioning at a COP of 0.6 under ambient db and wb of 90°F and 78°F respectively, and return air temperatures from the house of 75°F db and 63°F wb. This system is designed essentially to interface with a water collector.

R. V. Dunkle of the Commonwealth Scientific and Industrial Research Organization\*\*\* in Australia has been involved in the development of rotary regenerators for a number of years. As early as 1965, Dunkle has advocated the use of desiccant systems as a means of substantially reducing electrical power requirements. As a part of research programs, this organization has developed rotary regenerators and conducted limited testing of drying beds. A prototype system using fixed beds with periodic switching for adsorption and desorption has been built and subjected to performance evaluation. Analyses indicate that this type of system can be operated in hot humid regions.

\*William Rush, "Solar Desiccant Systems, Workshop Proceedings--Solar Cooling for Buildings," NSF-RA-N-74-063, February 1974.

\*\*Peter Lunde, paper presented at the workshop on The Use of Solar Energy for the Cooling of Buildings, Los Angeles, August 1975.

\*\*\*R. V. Dunkle, "A Method of Solar Air Conditioning," Mechanical and Chemical Engineering Transactions of The Institution of Engineers, Australia, Vol. MCI, No. 1, May 1965.

# DESICCANT AIR CONDITIONER SURVEY DATA

- INSTITUTE OF GAS TECHNOLOGY

2.5-TON GAS-FIRED SOLAR-ASSISTED UNIT

SORBENT: MOLECULAR SIEVE

OVERALL THERMAL COP: 0.73 WITH 290°F DESORPTION  
TEMPERATURE AND ARI STANDARD CONDITIONS

AUXILIARY POWER: 0.75 KW

- THE CENTER FOR ENVIRONMENT AND MAN, INC.

PRELIMINARY DESIGN OF A SOLAR-POWERED UNIT (WATER  
COLLECTOR)

SORBENT: SILICA GEL

THERMAL COP: 0.6

- COMMONWEALTH SCIENTIFIC AND INDUSTRIAL RESEARCH ORGANIZATION

PRELIMINARY DESIGN OF A SOLAR-POWERED UNIT (AIR COLLECTOR)

SORBENT: SILICA GEL

AUXILIARY POWER: 0.53 KW

DESORPTION TEMPERATURE: 180 TO 210°F



## REQUIREMENTS ANALYSIS (TASK 2)

Early in the study, requirements were defined for the purpose of concept evaluation and comparison. These requirements were taken as the standard values specified by the Air Conditioning and Refrigeration Institute (ARI Standard 240) for the purpose of rating unitary heat pump equipment.

Later actual ambient and environmental data, including dry bulb temperature, relative humidity, solar insolation, and house heat load, were obtained from experimental data collected as part of the Marshall Space Flight Center solar house test program. These data were used in the evaluation of the Rankine cycle air conditioner under simulated conditions.

## **REQUIREMENTS ANALYSIS (TASK 2)**



S-94773

## SYSTEM INTERFACES

Basically, a heat-powered air conditioner interfaces with the subsystems shown in the block diagram below.

Because of the nature of the subsystems and equipment constituting the air conditioner and the heat sink, these two subsystems were considered together in the study. The distribution subsystem establishes the air conditioner cooling requirements and does not include any equipment that affects the design of the air conditioner itself. The distribution fan was taken as part of the air conditioner package. The solar source subsystem was considered separately and defined by its performance interfaces.

The rationale for this breakdown is that the solar heat source will be used for heating in winter. Whether or not it is used to drive a heat-powered air conditioner is the subject of this study. On the other hand, the requirement for a heat sink subsystem is essentially dependent on the presence of the air conditioner. Consequently, in this study the air conditioner was considered to incorporate all equipment necessary to provide heat sink capability to the ambient air.

Two types of air conditioners are shown, depending on the heat sink technique used:

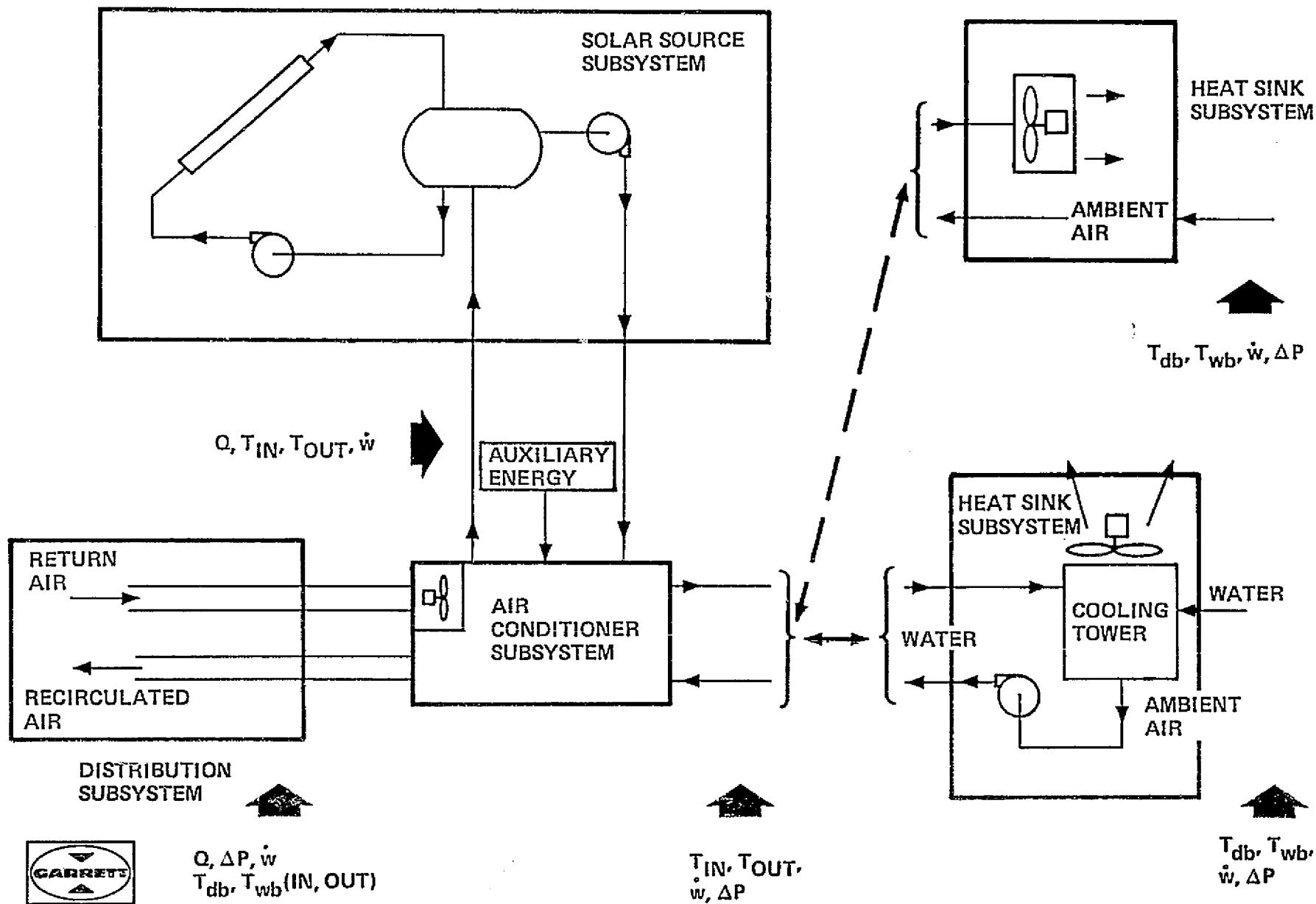
Air-Cooled Air Conditioners--Where the cycle waste heat is dumped directly to an ambient air stream

Water-Cooled Air Conditioners--Where the cycle waste heat is dumped into water from a cooling tower

In either case, ambient air is the ultimate heat sink. Since the heat sink is considered together with the air conditioner in the study, the interfaces between the two subsystems are internal optimization parameters.

A number of parameters internal to the air conditioner were used to optimize the system in terms of cost and performance. These internal optimization parameters include working fluid temperatures at the boiler, condenser, and evaporator, as well as the efficiencies of the turbine and compressor.

# SYSTEM INTERFACES



## DESIGN AND OPERATING PARAMETERS

The table below summarizes the data used for design and optimization of the air conditioner. Listed in the table are design point values used for concept comparison and selection. Given in parentheses are the ranges of the parameters investigated in the evaluation of the selected concept.

# DESIGN AND OPERATING PARAMETERS

COOLING CAPACITY	3 TONS
AMBIENT AIR	95°F DB, 75°F WB (70-100°F DB; 65-80°F WB)
SOLAR HEAT SOURCE	200°F (140 TO 240°F)
RETURN AIR	400 CFM/TON (MAX.); 80°F DB, 67°F WB (58-67°F WB)
ENERGY COST	
● ELECTRICAL	\$0.02 TO \$0.04/KW-HR
● NATURAL GAS	\$0.0013 TO \$0.0025/CU FT
● FUEL OIL	\$0.25 TO \$0.50/GAL
LOAD PROFILE	FROM THE NASA SOLAR HOUSE TEST DATA
COLLECTOR-TANK PERFORMANCE	FROM THE NASA SOLAR HOUSE TEST DATA



## TYPICAL DAILY VARIATION OF PARAMETERS

Experimental data obtained from the NASA-MSFC solar house were analyzed to provide daily variations of the parameters pertinent to the evaluation of the Rankine-cycle air conditioner. These parameters include:

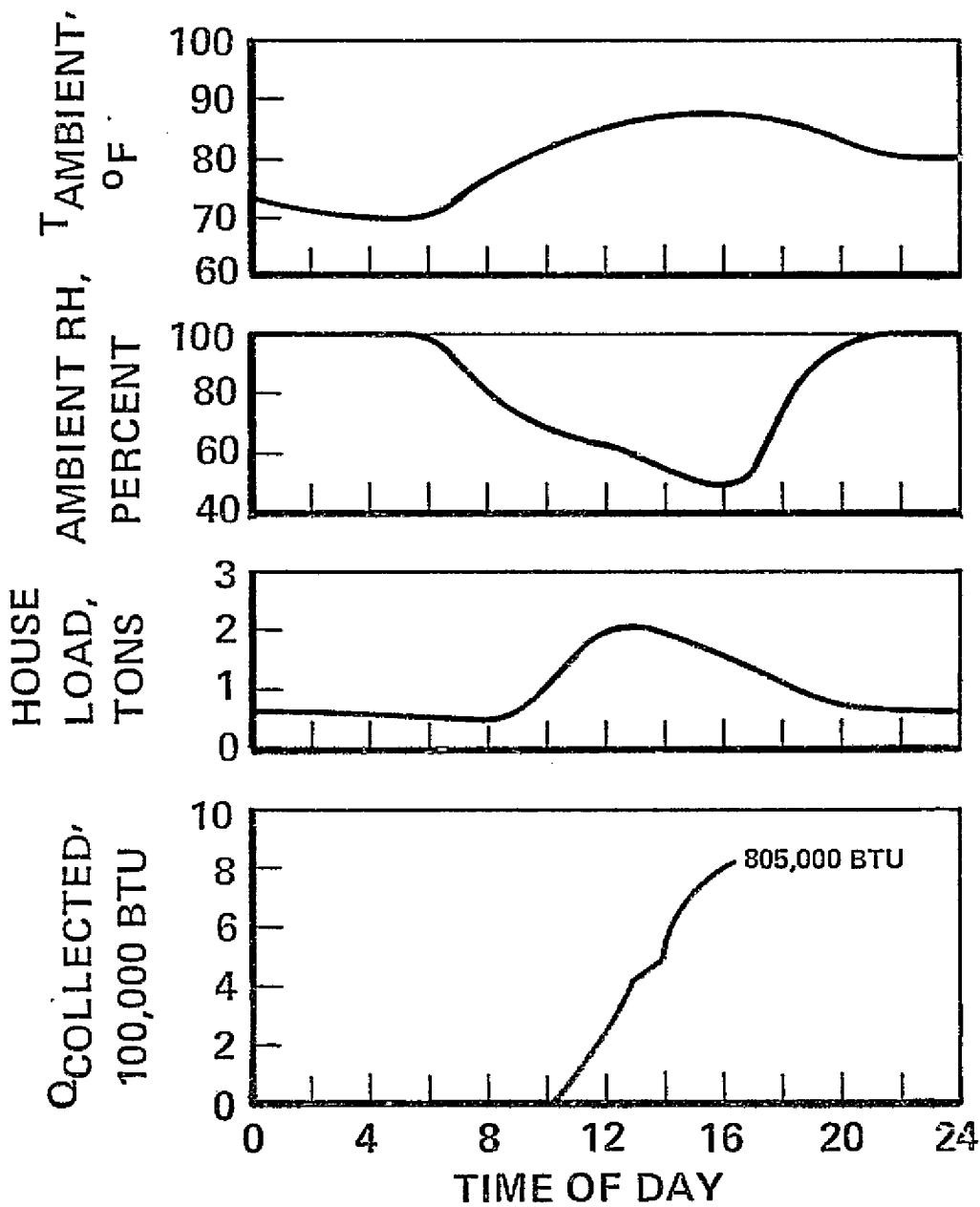
- (1) Ambient temperature and relative humidity--necessary to determine condenser performance
- (2) House air conditioning load--used in estimating on-off cycling, parasitic power usage, and auxiliary power requirements
- (3) Energy collected--basis for heat balance, system performance, and auxiliary power
- (4) System heat losses--used in computation of energy available at the air conditioner

These data were plotted as a function of time for five successive days covering the period from August 18 through 23, 1974. Examination of the data for these five days shows the following typical conditions, corresponding to maximum and minimum load situations occurring at about 6 and 16 hr respectively.

Time of Day, Hour	Load, Tons	Ambient Dry Bulb, °F	Ambient Wet Bulb, °F
6	2.2	70	70
16	0.5	89	74

Over the five-day period, the total amount of heat collected is calculated at 3,102,000 Btu for an average of 517,000 Btu/day. Maximum and minimum daily values are 805,000 Btu/day and 150,000 Btu/day respectively. The average air conditioning load over the 5 days is 1.3 tons with a maximum of 2.5 and a minimum of 0 tons.

# TYPICAL DAILY VARIATION OF PARAMETERS (NASA SOLAR HOUSE, AUGUST 20, 1974)



S-99820

## TRADEOFF AND EVALUATION PARAMETERS

The feasibility of using solar thermal energy for air conditioning will depend primarily on the economic benefits of such systems. To permit evaluation, the air conditioner must be defined in terms of parameters that can be translated directly into the cost factors to be considered in the overall system analysis. Thus, it has been necessary to develop models for estimating the installed cost of the air conditioning system itself. In addition, the following air conditioner characteristics were determined to provide the basic data necessary for estimating overall system fixed and operating costs:

- (1) System thermal COP--This parameter is basic to the definition of the solar collector/thermal energy storage unit size and cost.
- (2) Parasitic power--The power necessary to drive the system fans, pumps, and controls can present a significant factor in determining operating cost.
- (3) Auxiliary power--During periods of insufficient insolation, the system will require augmentation either through the use of electrical power or by combustion of natural gas or fuel oil.

For the largest part of the high population density areas of the United States, the major benefits of solar thermal energy will be for heating. Only the added portion (or quality) of the solar collection system necessary to support the cooling function should be considered as a penalty to the air conditioner.

# TRADEOFF AND EVALUATION PARAMETERS

- EQUIPMENT COST
  - AIR CONDITIONER
  - SOLAR COLLECTOR
  - THERMAL ENERGY STORAGE
- OPERATING COST
  - PARASITIC POWER
  - AUXILIARY POWER
  - MAINTENANCE



2-99817

### ECONOMIC ANALYSIS (TASK 3)

The following pages summarize the results of investigations aimed at the development of cost models to be used in the economic assessment of Rankine-powered air conditioning system for residential application.

The rationale used in the development of the cost models was to: (1) collect cost data on complete systems and on the major equipment used in these systems; (2) reduce these data and establish relationships between cost and other engineering parameters such as weight, size, power level, etc; and (3) derive simple correlations from which system cost-to-the-user can be calculated from performance requirements.

These cost models were used for the purpose of system evaluation and tradeoff studies. The final cost of the selected system can be determined more accurately through detailed cost estimates using equipment sizes and problem statements derived from the system and component analyses and detailed packaging studies that are beyond the scope of the current studies.

# ECONOMIC ANALYSIS (TASK 3)



S-94772-A

Air conditioners are marketed through distributors and dealers to the consumer. List prices represent the approximate price that the user will ultimately pay. Normally, the manufacturer price to the distributor is about 35 percent of the list price. The distributor markup is roughly 30 percent. The contractor will procure the equipment from the distributor and realize a profit on the equipment; in addition, installation charges will be passed on to the user. As a rule, the total contractor contribution to cost-to-user will be about 55 percent of the list price. On this basis, the installed cost of a system can be computed by

$$\text{Installed cost} = 2.86 \text{ (factory sell price)}$$

Historical data obtained from a large manufacturer of residential air conditioners indicate that as a first approximation the factory direct cost of such equipment can be expressed in terms of the original equipment manufacturer (OEM) cost of the major component by the relation

$$\text{System factory direct cost} = \frac{1}{0.7} (\Sigma \text{ component OEM costs})$$

This equation states that the fraction of the cost attributable to structures, enclosures, and assembly operations accounts for 30 percent of the total. Thus, if the component cost can be determined from engineering parameters, then factory direct cost can be estimated. The factory direct cost is defined as the expenditures directly related to the fabrication of a system, including materials, shop labor, and shop setup for a production run; it does not include shop overhead, administrative expenses, or profit. The factory sell price can be determined from the factory direct cost by including overhead and markup. The following factors were used in establishing factory sell price:

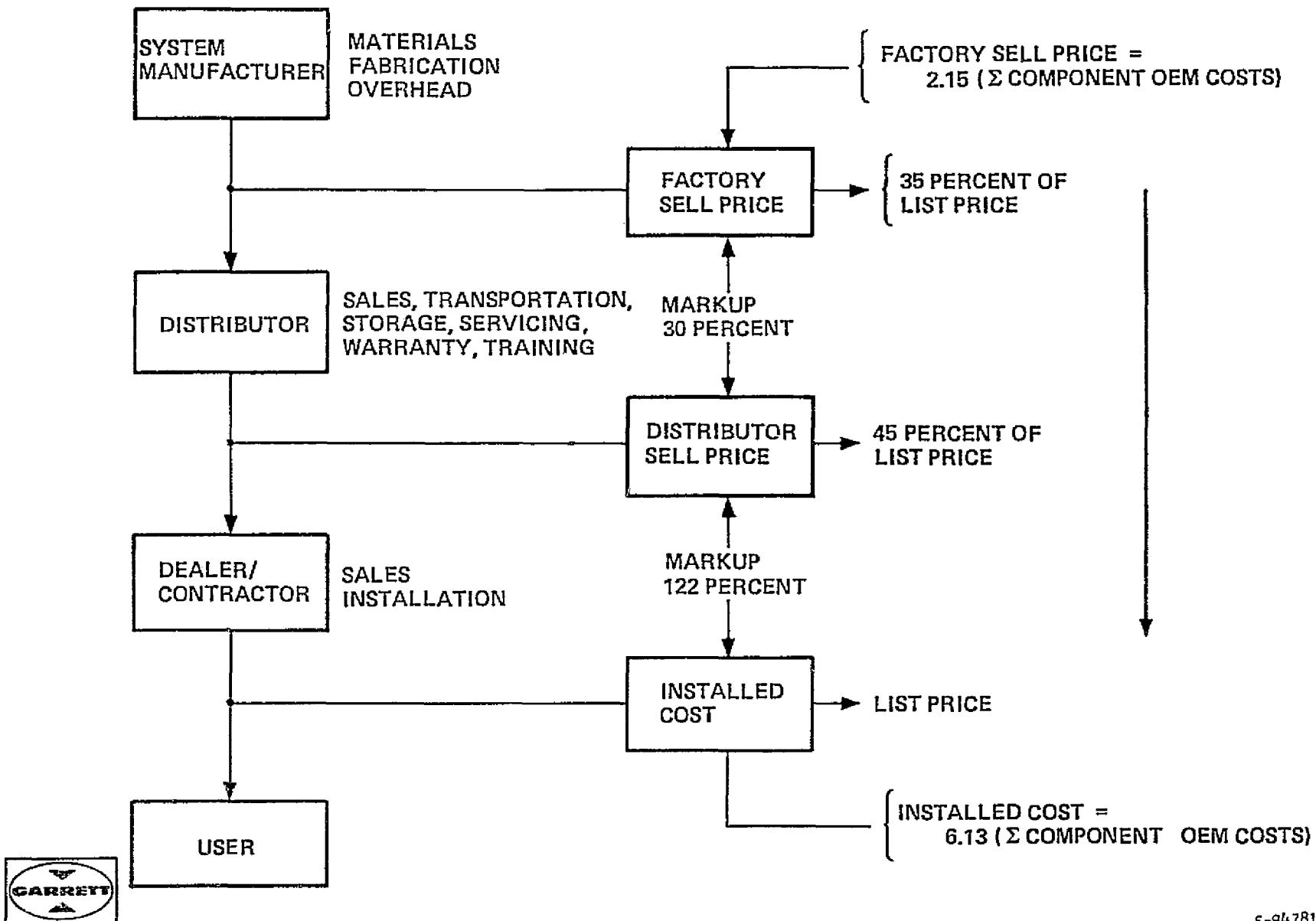
- (a) Labor constitutes about 11 percent of the total system cost for systems in the 2- to 3-ton capacity range
- (b) Shop overhead is estimated at 2.5 times direct labor cost; shop overhead can be approximated by

$$\text{Shop overhead} = 0.275 \text{ (factory direct cost)}$$

- (c) A 30 percent markup was used to cover administrative expenses, engineering, accounting, profit margin, etc. Using these factors, the factory sell price of a system can be calculated from

$$\text{Factory sell price} = 2.3 (\Sigma \text{ component OEM costs})$$

# SYSTEM COST MODEL



System costs can be estimated if the cost of the major system components is known. The cost of components is determined primarily by the materials and the labor costs as affected by the design sophistication of the equipment. Equipment costs furnished by various equipment manufacturers were analyzed and correlated to engineering parameters. Component cost relationships are listed below.

Heat Exchangers--Two types of heat exchangers will be used in Rankine-powered air conditioners: finned tube units for the evaporator and certain condensers, and tubular units for other condenser types and the boiler. The finned tube units consist of copper tubes staggered in the direction of the air flow with wavy aluminum fins mechanically bonded to the tubes on the airside. Material cost for such units is about \$0.84/lb. The cost of the tubular units is estimated in a fashion similar to that of the finned tube units. Factory cost is about \$1.53/lb for an all-copper alloy unit.

Refrigerant Pumps--Estimates of the refrigerant pump requirements indicate that small vane pumps in the size of interest are available commercially. The OEM price on this type of pump is about \$20 for the pump element itself. The entire assembly cost, including a canned motor drive, is estimated at about \$40.

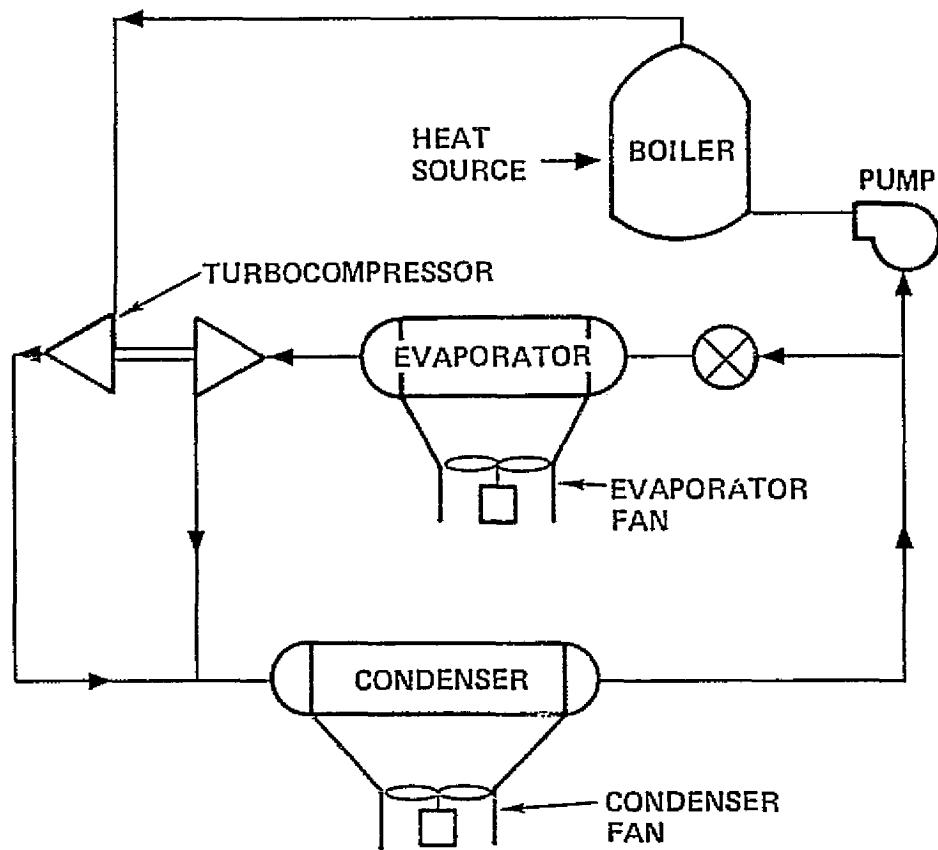
Turbocompressors--Cost data on turbocompressors of considerable aerodynamic sophistication were obtained from the AiResearch Industrial Division (AID) of The Garrett Corporation. Here, material cost is a small portion of the total cost of the unit. For the purpose of system comparison, a unit price of \$100 per unit was used.

Fans and Blowers--Manufacturers' catalogues and price lists were used to determine fan cost. The OEM cost of these fans (steel units) can be approximated at \$0.88/lb. This value was used in estimating fan cost from parametric weight data.

Electric Motors--Electric motors will be designed for operation from a single-phase, 60-Hz, 115/230-v source. Detail matching of the system fan and motor is beyond the scope of the current work. For the purpose of system selection and parametric analysis, belt-drive motors will be assumed to permit fan operation at the optimum speed.

System Controls--At this time, controls constitute a gray area. For a first approximation, the cost of controls and wiring harnesses will be taken as 10 percent of the total system cost.

# COMPONENT COST MODELS



EQUIPMENT	COST MODEL, DOLLARS
<u>HEAT EXCHANGERS</u>	
● EVAPORATOR	0.84 (HEAT EXCHANGER CORE WEIGHT, LB)
● CONDENSER	1.53 (HEAT EXCHANGER WEIGHT, LB)
● BOILER	
<u>REFRIGERANT PUMP</u>	
REFRIGERANT PUMP	40
<u>TURBOCOMPRESSOR</u>	
TURBOCOMPRESSOR	100
<u>FANS/BLOWERS</u>	
FANS/BLOWERS	0.88 (FAN WEIGHT, LB)
<u>ELECTRIC MOTORS + DRIVES</u>	
ELECTRIC MOTORS + DRIVES	10 + 32.5 (KW)
<u>CONTROLS AND WIRING</u>	
CONTROLS AND WIRING	10 PERCENT OF TOTAL



Screening analyses were aimed at the definition of the baseline configuration of a Rankine-cycle solar-powered air conditioner. These studies were concerned with the development of a system optimized on the basis of a fixed design point defined by the standard ARI rating conditions (ambient: 95°F db, 75°F wb; house return: 80°F db, 67°F wb) and by a hot water temperature from the solar collector thermal storage of 200°F. A number of topics were considered in the development of the baseline schematic, including

- (1) Fluid selection
- (2) The use of recuperators
- (3) Selection of a condenser concept
- (4) Definition of optimum internal parameters
- (5) Baseline system controls definition

The performance of these studies required the development of component and system models to permit mechanization of the design calculations so that the system characteristics (COP, parasitic power, and cost) could be calculated over a wide range of ambient conditions and air conditioner internal parameters.

Other system configuration trade studies were performed as part of Task 5 after the definition of the baseline schematic. These latter studies were concerned with the selection of an optimum augmentation scheme. This involved prediction of the performance of a particular system (defined as baseline) under ambient conditions prevalent over a typical time period. The results of these investigations are presented later.

The following discussions describe the activities that led to the definition of the baseline system schematic.

# SCREENING ANALYSIS (TASK 4)



## FLUID COMPARISON

Parametric cycle performance data have been generated covering a range of operating conditions defined by the following:

4 refrigerants: R-11, R-12, R-113, and R-114

3 capacity sizes: 1-1/2, 3, and 5 tons

Condensing temperature from 85 to 110°F

Boiling temperature from 170 to 190°F

All computations were done for an evaporating temperature of 45°F.

Plotted below are the data obtained for an air conditioning load of 1-1/2 tons and a boiling inlet temperature of 180°F. These plots are typical of the other capacities and boiler temperatures investigated. The thermal COP is defined as follows:

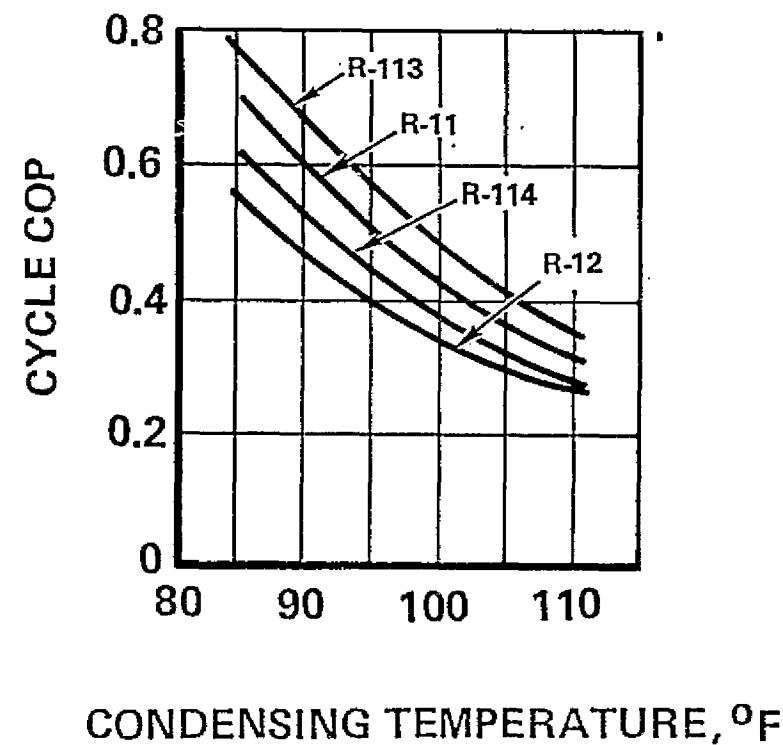
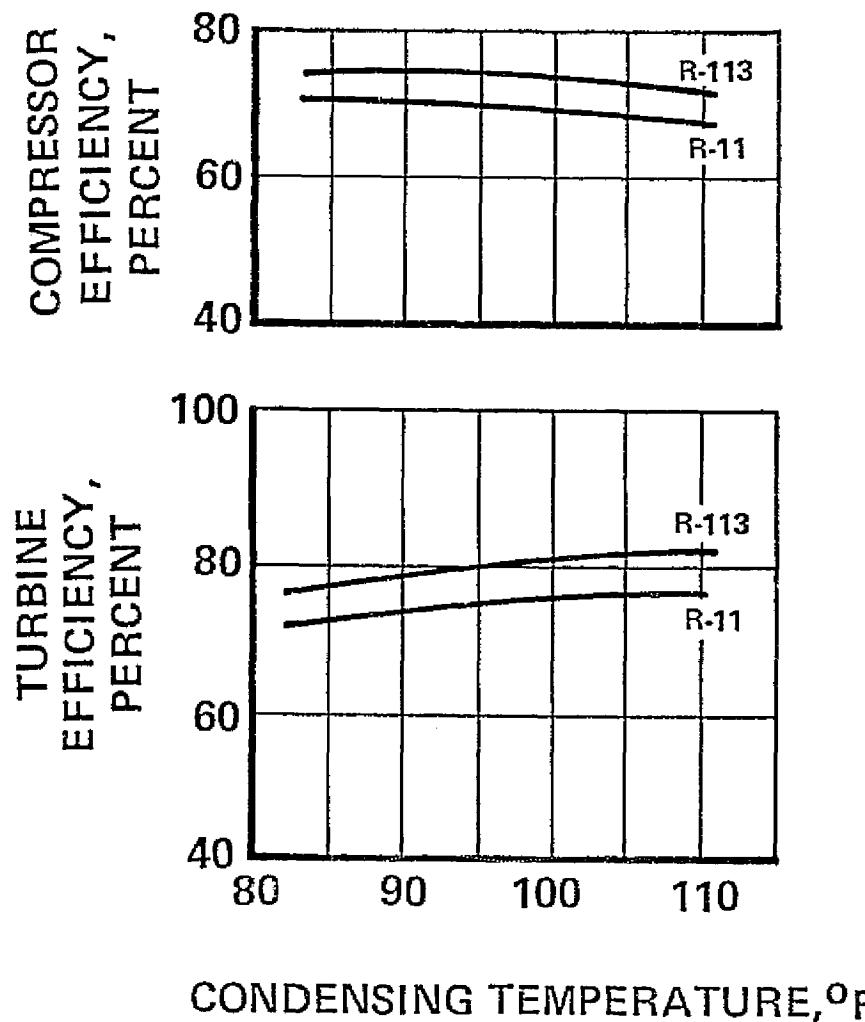
$$\text{Thermal COP} = \frac{\text{evaporator load}}{\text{boiler heat input}}$$

The data show R-113 as the best refrigerant in terms of COP, because of the higher compressor/turbine efficiencies attainable with this low-density refrigerant over the capacity range investigated.

R-22 is essentially a high-pressure refrigerant developed specifically for commercial reciprocating compressors. The use of this refrigerant in small-tonnage systems employing turbomachinery results in relatively low compressor and turbine efficiencies because of the low volumetric flow rate. For this reason, this refrigerant was not considered under this program. R-22 has not been considered because it is a higher pressure fluid than R-12; the characteristic COP with R-22 in a turbomachine system would be lower than that of R-12.

# FLUID COMPARISON

BOILING TEMPERATURE: 180°F  
AIR CONDITIONING LOAD: 1-½ TONS



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## FLUID SELECTION

Refrigerants for conventional residential-commercial air conditioners are selected on the basis of other considerations in addition to cycle COP. One of the most important factors is the density of the refrigerant at compressor inlet; higher densities will result in smaller and less expensive reciprocating compressors, heat exchangers, and interconnecting lines. For this reason R-22 is commonly used. Other considerations influencing selection include:

Compressor discharge temperature limitations

Fluid stability at maximum cycle temperature

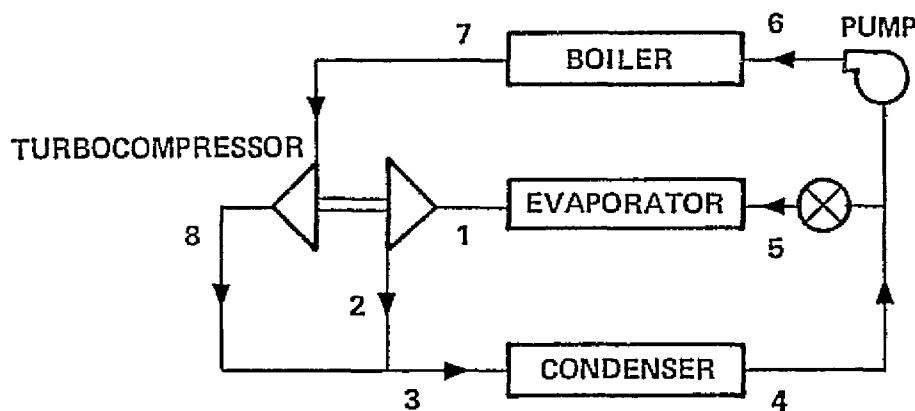
Susceptibility to decomposition under exposure to low concentrations of water vapor

Susceptibility to leakage

In the type of systems considered here, the maximum cycle temperatures will be determined by the temperature of the heat source. No fluid stability problems are anticipated at the maximum temperature (240°F) expected from the solar collector/thermal storage unit.

Although R-113 offers a slight COP advantage over R-11, the low pressures throughout the system make this fluid undesirable for long-term system operation with minimum maintenance. The pressures at various state points in the system are listed below for the R-113 and R-11 refrigerants. Leakage of minute quantities of air into the system over long periods will deteriorate system performance and require recharging. By comparison, system pressures with R-11 are positive (relative to ambient), except between the expansion valve and the compressor. This constitutes only a small portion of the system and appears acceptable in view of the COP advantages offered by R-11 in comparison with R-12 and R-22. Thus, R-11 was selected for further system studies.

# FLUID SELECTION



STATION	REFRIGERANT PRESSURE, PSIA	
	R-113	R-11
1*	3.04	8.0
2	13.4	29.8
3	13.4	29.8
4**	12.8	28.4
5	3.19	8.4
6	43.3	84.2
7***	41.2	80.2
8	13.4	29.8

\*EVAPORATING TEMPERATURE: 45°F

\*\*CONDENSING TEMPERATURE: 110°F

\*\*\*BOILING TEMPERATURE: 180°F

R-11 SELECTED ON BASIS OF NEAR MAXIMUM COP AND ACCEPTABLE SYSTEM PRESSURES.



S-94813-A

## TYPICAL HEAT RECOVERY APPROACHES

A number of Rankine system arrangements were investigated to determine what thermodynamic benefits could be achieved through the use of recuperators. The performance of each of the various approaches considered was calculated using the following design point conditions and assumptions.

System capacity: 3 tons  
Refrigerant: R-11 common refrigerant for the power and refrigeration loops  
Evaporator outlet: 45°F saturated  
Condenser outlet: 90°F saturated  
Boiler outlet: 185°F saturated  
Compressor efficiency: 0.736 (same as baseline system; see data presented later)  
Turbine efficiency: 0.771 (same as baseline system; see data presented later)  
Boiler, evaporator, condenser  $\Delta P$ : 5 percent of inlet pressure  
Recuperator, subcooler/superheater  $\Delta P$ : 2 percent of inlet pressure on vapor side; on liquid side,  $\Delta P$  negligible  
Recuperator, subcooler/superheater effectiveness: 0.85 max.

The results of these investigations are summarized below, where three of the seven approaches considered are compared to the baseline system. In all cases, the quantities of heat recovered are relatively small in comparison to the total heat input to the boiler. The recuperator and subcooler/superheater loads represent less than 5 percent of the boiler load. However, higher compressor work is necessary to overcome the additional heat exchanger pressure drops; simultaneously, the available power at the turbine decreases for the same reason. The net effect is a small change in overall system thermal COP.

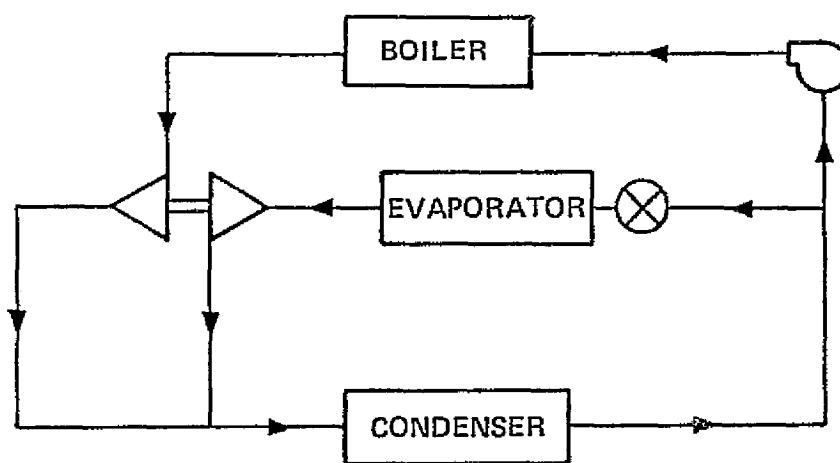
No significant improvement in COP can be realized through the use of recuperators and/or subcooler/superheaters. The very small gain (0.02) shown below is at best marginal and does not warrant the addition of two heat exchangers and associated lines to the baseline system.



# TYPICAL HEAT RECOVERY APPROACHES

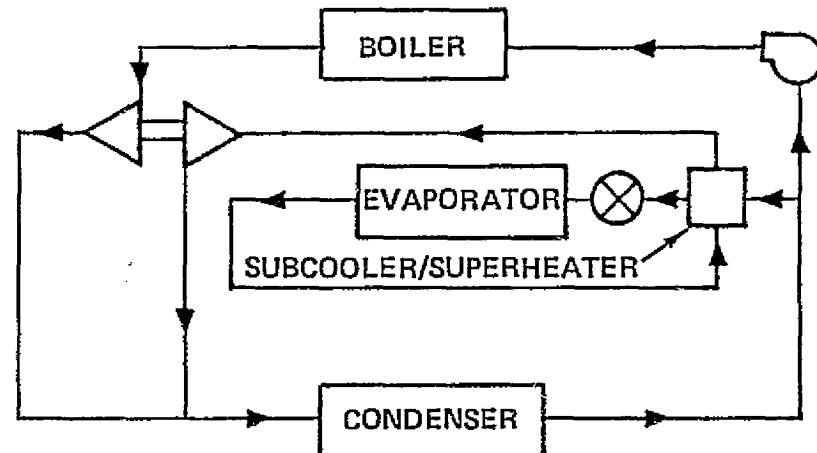
**BASELINE**

COP: 0.69



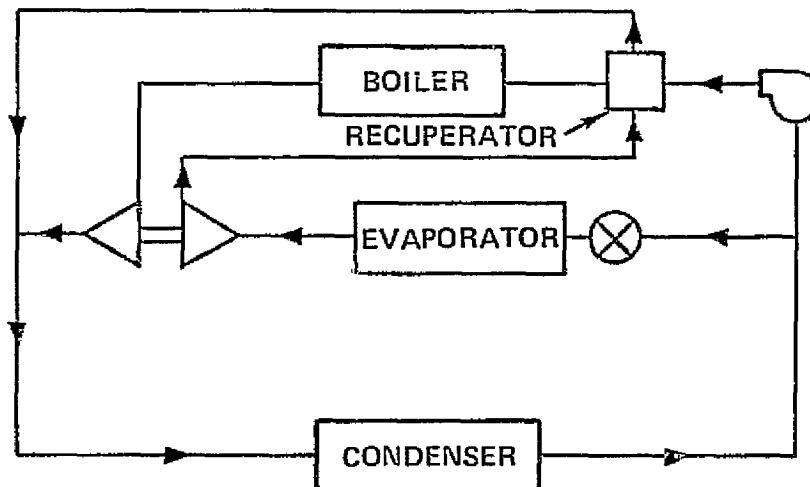
**SUBCOOLER/ SUPERHEATER**

COP: 0.69



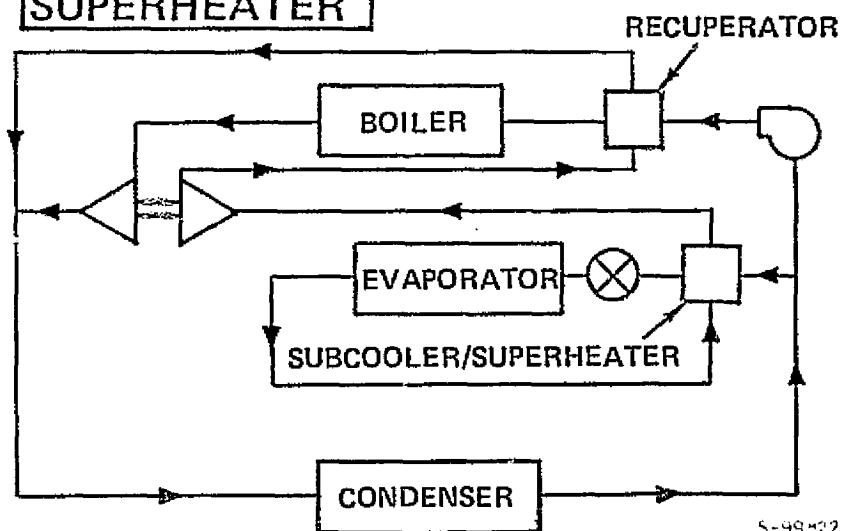
**COMPRESSOR RECUPERATOR**

COP: 0.70



**RECUPERATOR- SUBCOOLER/ SUPERHEATER**

COP: 0.71



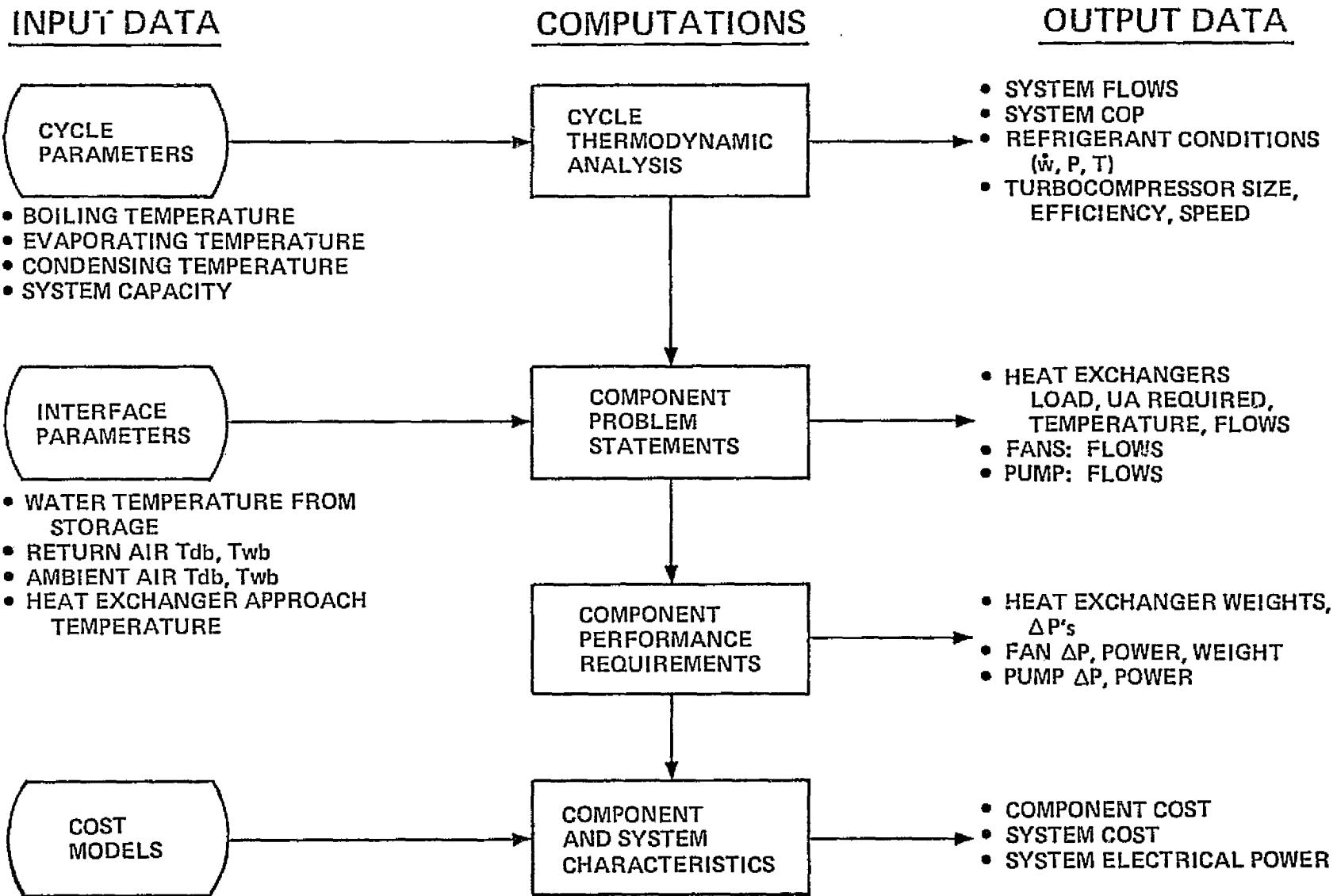
The logic used by the computer program is illustrated below. Basically, the computation of component and system characteristics is performed as follows. Cycle parameters defining the conditions of the refrigerant within the power and refrigeration loop heat exchangers are used to perform thermodynamic analyses. In these computations, it is essential that the efficiencies of the turbine and compressor be estimated accurately, and that the speed of these two components be matched to provide realistic refrigerant conditions through the loop and to assure design feasibility for the turbomachinery.

Through an iterative procedure designed to match the compressor and turbine speed and power, the computer program determines the system flows, R-11 conditions, and system COP. These cycle data are then used with specified interface and heat exchanger approach temperatures to generate problem statements for the heat exchangers, fans, and pumps. The characteristics of these components are then determined in terms of parameters that can be related to cost. Finally, the cost models described previously are used to determine component and overall system cost.

The output data include:

- (a) R-11 temperature, pressure, enthalpy, flow rate, and density at all stations
- (b) Heat exchanger flows, temperatures, heat loads, UA requirement, weight, and cost
- (c) Fan characteristics including flow, pressure rise, and power
- (d) Wet bulb temperature of the air at inlet and outlet of the evaporator and condenser
- (e) Cycle characteristics: power loop efficiency, refrigeration loop COP, and overall system COP
- (f) Turbine and compressor characteristics: efficiency, impeller diameter, and speed
- (g) Electric power requirements for the fans and pumps
- (h) System cost data

# DESIGN COMPUTER PROGRAM LOGIC



COMPONENT DESIGN DATA--COMPRESSOR PERFORMANCE

The efficiency of a single-stage centrifugal compressor can be determined by analytical and experimental data correlated in terms of adiabatic head rise and the four dimensionless parameters listed below:

- |                                |                     |
|--------------------------------|---------------------|
| (a) Adiabatic head coefficient | (c) Tip Mach number |
| (b) Specific speed             | (d) Reynolds number |

The plot below shows the achievable efficiency of centrifugal compressors plotted as a function of specific speed and tip Mach number. This plot is based on actual test data obtained over a range of specific speeds and Mach number; the data cover specific speeds as low as 0.02. The data are representative of recent machines, fabricated using modern fabrication techniques to assure dimensional accuracy and smooth surface finishes. The efficiency plot corresponds to compressor impeller diameters larger than 4.0 in. and Reynolds numbers higher than  $10^6$ . For smaller compressor sizes and lower Reynolds numbers, the efficiency obtained from the non-dimensional plot must be corrected to account for additional losses. The size correction factor (also the result of empirical correlation) is shown in the right side of the figure below. The Reynolds number correction factor can be computed by

$$\frac{1-\eta^*}{1-\eta_0} \left( \frac{10^6}{Re} \right)^{0.1} \quad \text{Where } \eta^* \text{ is the corrected efficiency and } \eta_0 \text{ is the efficiency determined for } Re > 10^6$$

Over the range of specific speeds and tip Mach number anticipated, with R-11 as the working fluid and operating conditions corresponding to the entire range of system interface data, it is estimated that compressor efficiency will vary from a maximum of 80 percent for the larger tonnage machines to a minimum of 60 percent for the smaller units. Here, specific speed is defined as

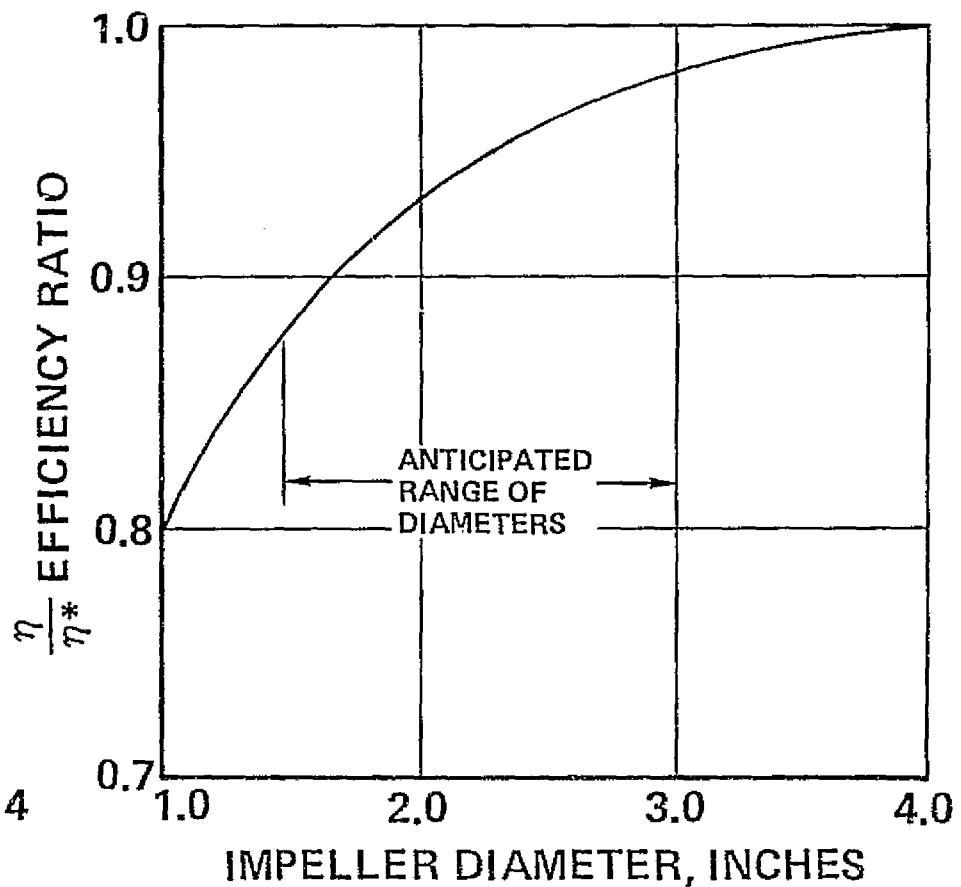
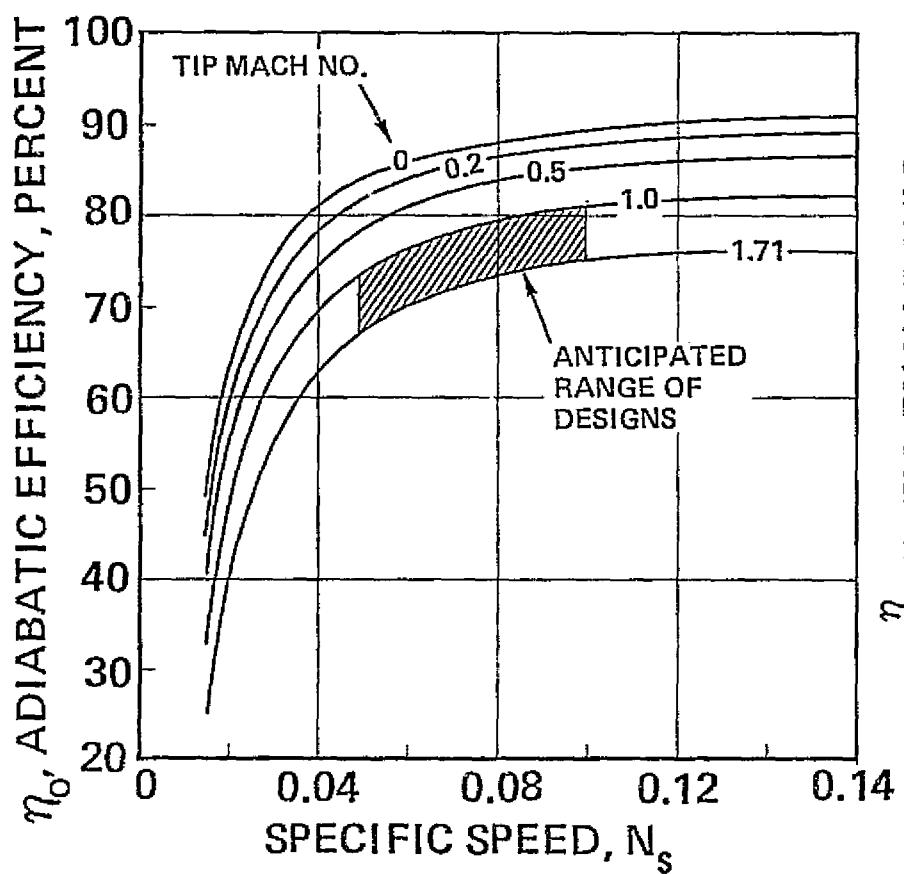
$$N_S = \frac{NV}{60(g_c H_{ad})}^{0.5}$$

where  $\left\{ \begin{array}{l} H_{ad} = \text{adiabatic head, ft-lb}_f/\text{lb}_m \\ V_{in} = \text{volumetric flow, ft}^3/\text{sec} \\ N = \text{rotational speed, rpm} \\ g_c = 32.174 (\text{lb}_m/\text{lb}_f) (\text{ft/sec}^2) \end{array} \right.$

# COMPONENT DESIGN DATA COMPRESSOR PERFORMANCE

$\eta$  = CORRECTED EFFICIENCY

$\eta^*$  = EFFICIENCY FOR IMPELLER  
DIAMETER  $\geq$  4.0 INCHES



As for the centrifugal compressor, radial reaction turbine data have been collected from published literature and from numerous units developed and tested at AiResearch. These experimental data were correlated in the form of efficiency as a function of specific speed and size. Based on these data, the efficiency plot shown below is representative of modern high-performance, high-speed machines.

The data below 50 percent efficiency represent extrapolations of the plots supported by actual test data. This was done to obviate problems with computerized calculation methods. The anticipated range of designs is well above the 50 percent efficiency level.

The efficiency data presented apply to machines with Reynolds numbers larger than 200,000. For lower Reynolds numbers, a correction factor (determined experimentally) must be applied:

$$\frac{1-\eta}{1-\eta_0} = 0.4 + 0.6 \left( \frac{Re}{200,000} \right)^{-0.2} \quad \text{where } \eta \text{ is the corrected efficiency and } \eta_0 \text{ is the efficiency obtained from the efficiency plot.}$$

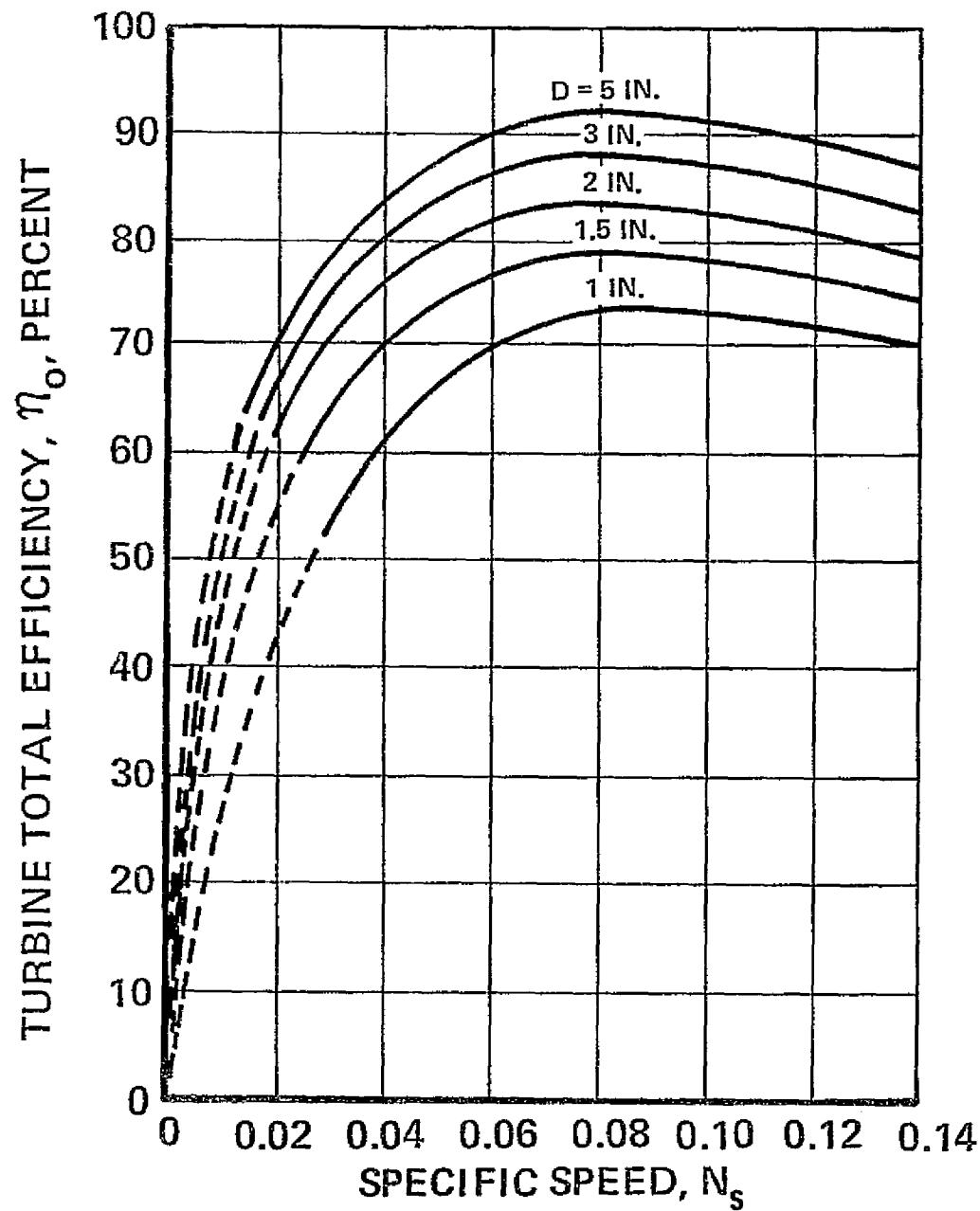
In the plot below, specific speed is defined as

$$N_S = \frac{N \sqrt{Q}}{(g_c H_{ad})^{3/4}}$$

where  $\left\{ \begin{array}{l} N = \text{rev/sec} \\ Q = \text{exit volume flow, ft/sec} \\ H_{ad} = \text{adiabatic head, ft} \end{array} \right.$

Examination of the data indicates a potential for very high efficiencies. The relative aerodynamic sophistication of the turbines and compressors considered here does not preclude their fabrication using mass production technology. For example, turbochargers manufactured by AiResearch Industrial Division (AID) for diesel engines have compressor and turbine efficiencies as high as 74 percent. These machines rotate at speeds up to 135,000 rpm and develop pressure ratios as high as 3:1. The factory sell price (OEM price) for this type of unit is estimated at \$100 per unit for production quantities of 150,000 per year. This is the type of turbomachinery considered for the Rankine air conditioners.

# COMPONENT DESIGN DATA TURBINE PERFORMANCE



## COMPONENT DESIGN DATA--FAN CHARACTERISTICS

Several types of fans or blowers can be used to provide air circulation through air conditioner evaporators and condensers. The particular type used will depend primarily on cost and installation constraints; efficiency is a secondary consideration for low-tonnage systems. Low capacity squirrel cage blowers are generally used in residential systems. These small units have relatively low efficiency--between 20 and 40 percent. Efficiencies as high as 65 percent are typical of larger capacity squirrel cage blowers used primarily in commercial and industrial applications.

Vane axial fans are often selected in the larger sizes where efficiency is more important. However, this type of fan is relatively more expensive than squirrel cage units.

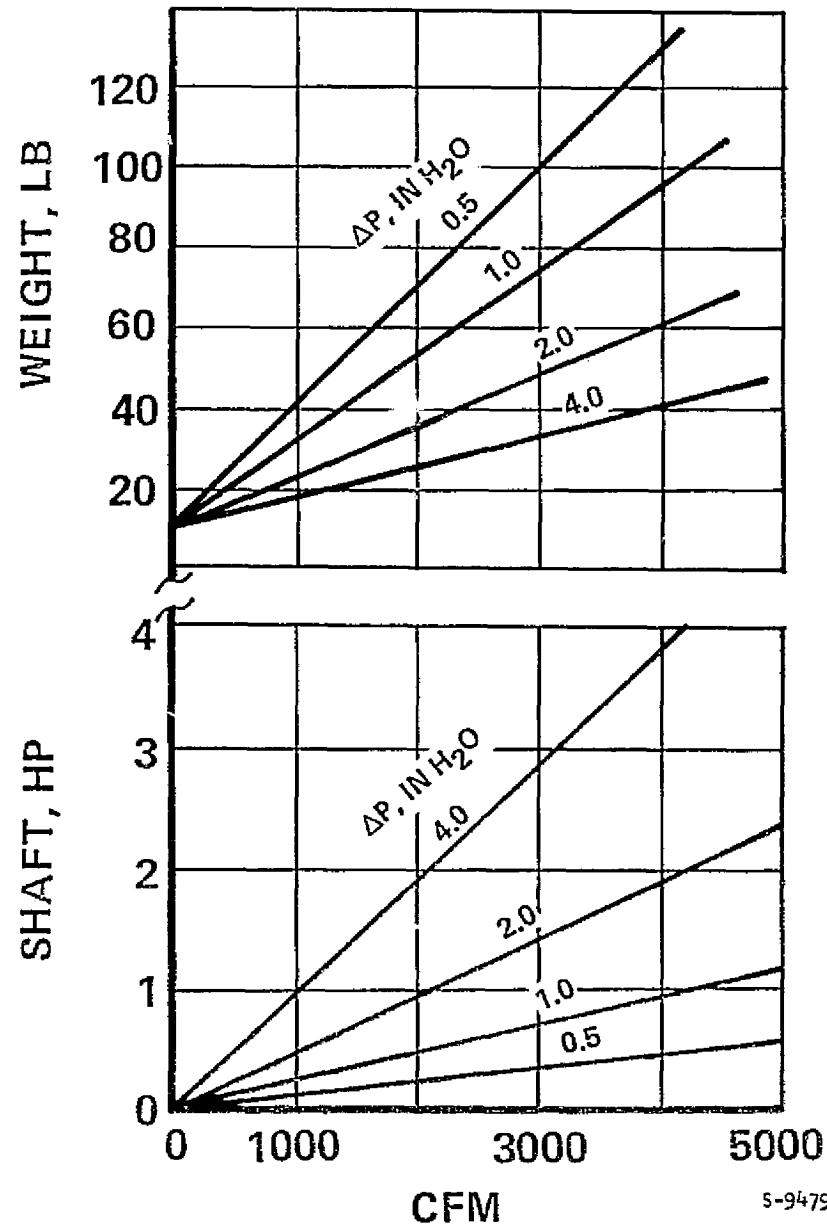
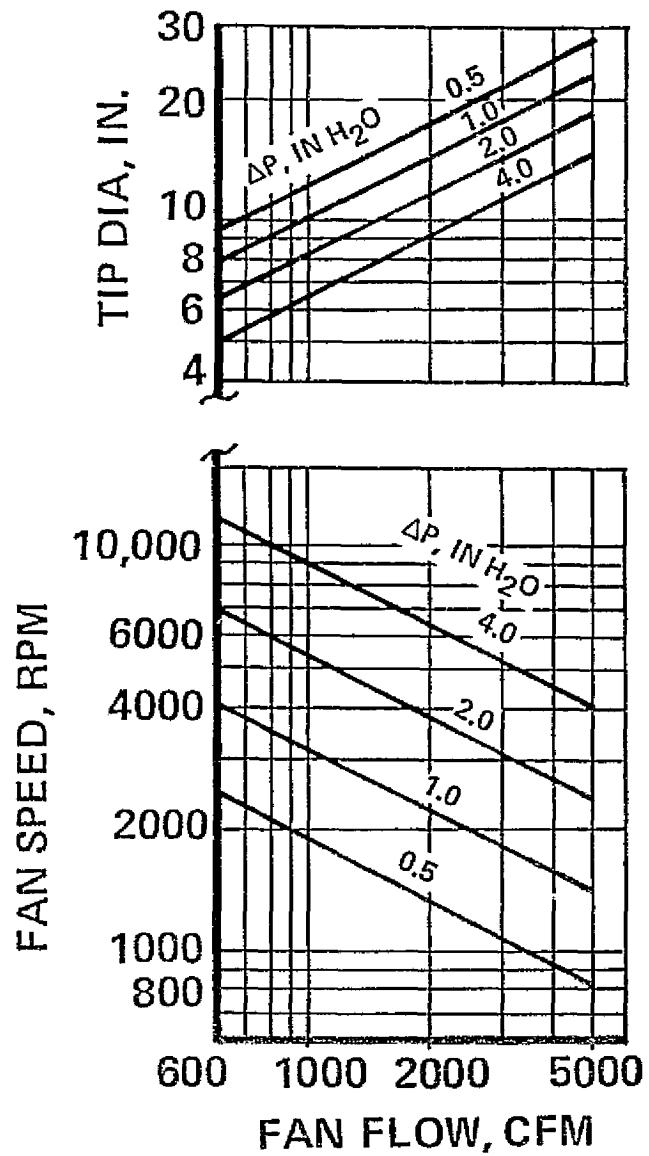
For the purpose of estimating the performance and cost of the Rankine system fans, models were developed to characterize vane axial fans in terms of the flow and pressure rise requirements. This type of fan will give an efficiency of 70 percent over the range of flow rate and pressure rise under consideration. Speed matching between the motor and the fan is assumed to be near optimum so that this high efficiency can be achieved. In practice, this can be accomplished by providing the speed flexibility afforded by a belt drive.

Fan data used in the development of the system computer model are shown in the attached figure. The weights do not include the motor and were estimated using steel as the material of construction. This weight agrees fairly well with catalogue data in the range of flows and  $\Delta P$ 's shown.

In the actual design of a system, it may be necessary to limit fan diameter because of packaging considerations. Such size constraints can only be approximated at this time by assuming that fan diameters larger than 20 in. cannot be accommodated. In cases where the capacity of one fan is exceeded, two fans will be used.

# COMPONENT DESIGN DATA

## FAN CHARACTERISTICS



5-94792-A

COMPONENT DESIGN DATA--HEAT EXCHANGERS

Finned tube heat exchangers constructed of copper tubing to which aluminum wavy fins are mechanically bonded will be used as typical to model the system evaporator and air-cooled condenser. This type of heat exchanger can be fabricated very cheaply, primarily because of the mechanical bonds between the fins and the tubes. As mentioned previously, the factory cost for these units is about \$0.84/lb. The heat transfer surface selected to model the evaporator and condenser in the system computer program is defined by its geometry and heat transfer characteristics. Commercial units are usually designed with air velocities of about 500 ft/min. At this velocity, the heat transfer conductances ( $hA$ , Btu/hr°F) on the air and refrigerant sides are about the same (235 Btu/hr°F).

The evaporative condenser performance was determined from catalog data on similar units. The capacity of this unit is expressed as a function of the total heat content potential of the air. A conservative value of  $200 \text{ Btu/hr ft}^2 \text{ }^{\circ}\text{F}$  was used here as the condensing side heat transfer coefficient.

The liquid condenser is a conventional shell-and-tube unit, with the refrigerant condensing within the shell while the cooling water flows through the copper tubes. The same condensing heat transfer coefficient as above was used in estimating the size and weight of this unit.

The boiler is a tubular unit operated as a pool boiler with the refrigerant outside the tubes. A liquid separator section followed by a superheater section will be necessary to prevent liquid carryover to the turbine and condensation in the turbine expansion nozzles. At the operating temperature range of interest, it is estimated that about  $10^{\circ}\text{F}$  superheat will be necessary.

The characteristics of the cooling tower were derived from data on commercial units where the water temperature can be reduced by evaporation to a value approaching the wet bulb temperature of the air by  $4^{\circ}\text{F}$ . Similar data were used in estimating the performance of the humidifier. In this case the temperature effectiveness of this unit was taken as 0.9.

# COMPONENT DESIGN DATA: HEAT EXCHANGERS

## DRY AIR CONDENSER

- COPPER TUBES WITH WAVY ALUMINUM FINS
- HEAT TRANSFER CONDUCTANCE: 118 BTU/HR/°F/TUBE ROW/SQ FT FACE AREA
- $\Delta P = 0.088$  (NO ROWS)  $0.746$  IN. H<sub>2</sub>O

## EVAPORATOR

- SAME AS ABOVE
- WET HEAT TRANSFER CONDUCTANCE: 330 BTU/HR/°F/TUBE ROW/SQ FT FACE AREA

## EVAPORATIVE CONDENSER

- COPPER TUBES WITH INTERNAL EXTENDED SURFACE
- HEAT TRANSFER CAPACITY:  $q = 373 A \Delta h$ , BTU/HR
- CONDENSING HEAT TRANSFER COEFFICIENT: 300 BTU/HR/SQ FT °F
- $\Delta P = 0.5$  IN. H<sub>2</sub>O

## LIQUID-COOLED CONDENSER

- COPPER TUBES
- CONDENSING HEAT TRANSFER COEFFICIENT CONTROLLING

## BOILER

- COPPER TUBES WITH INTERNAL EXTENDED SURFACE
- BOILING SIDE HEAT TRANSFER COEFFICIENT CONTROLLING (200 BTU/HR/SQ FT °F)

## COOLING TOWER

- WATER TEMPERATURE APPROACHING AMBIENT WET BULB
- FAN POWER:  $2.96 \times 10^{-3} Q$  WATTS (75°F WB)

## HUMIDIFIER

- EFFECTIVENESS OF 0.9 ASSUMED  $(\epsilon = \frac{T_{DB_{IN}} - T_{DB_{OUT}}}{T_{DB_{IN}} - T_{WB}})$

S-99913



## CANDIDATE CONDENSER APPROACHES

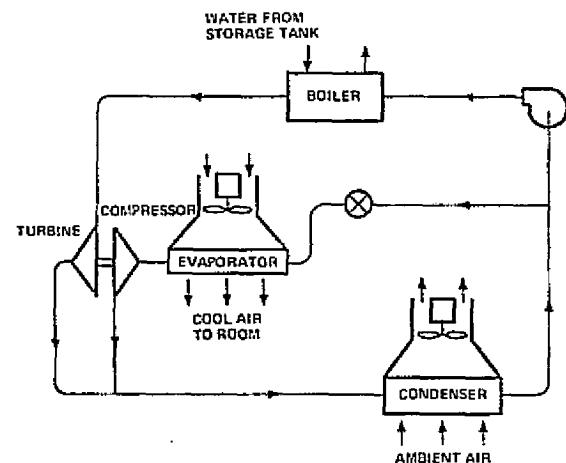
Initial studies revealed that system performance and cost were extremely sensitive to condensing temperature and to the type of condenser used in the system. Consequently, a major portion of the screening analyses was concerned with the generation of parametric design data for the four different condenser approaches defined below.

All systems considered feature a high-performance turbocompressor and a single refrigerant (R-11) for the power and refrigeration loops. The data summarized in the following pages were obtained by computerized methods developed to permit system characterization over a broad range of operating and design conditions. The criteria used for comparison of the candidate system approaches are listed below.

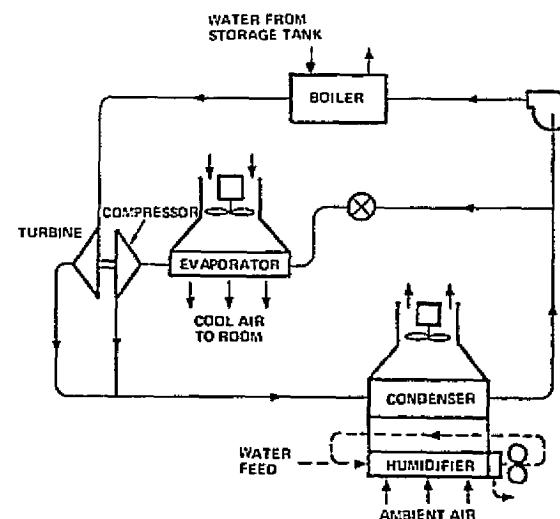
- (a) Overall system COP (refrigeration effect/solar heat input)
- (b) Parasitic electric power for fans and pumps
- (c) System installed cost or cost to the user

# CANDIDATE CONDENSER APPROACHES

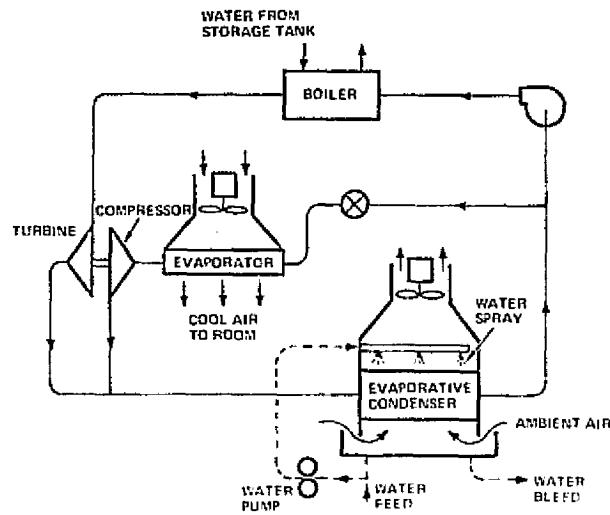
## DRY CONDENSER



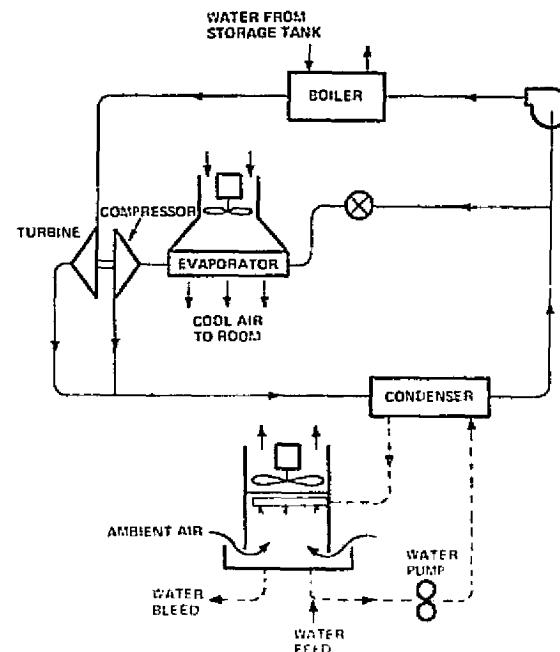
## HUMIDIFIER/CONDENSER



## EVAPORATIVE CONDENSER



## COOLING TOWER/CONDENSER



TYPICAL DESIGN DATA  
(DRY CONDENSER)

The system characteristics obtained by computer analysis were plotted in terms of the significant design parameters--boiling and condensing temperature and heat exchanger approach temperatures. The conditions identified previously were used for purposes of comparison. In all cases, the refrigerant temperature was taken as 45°F--consistent with the requirements for latent heat control. Typical data are shown for the dry condenser concept.

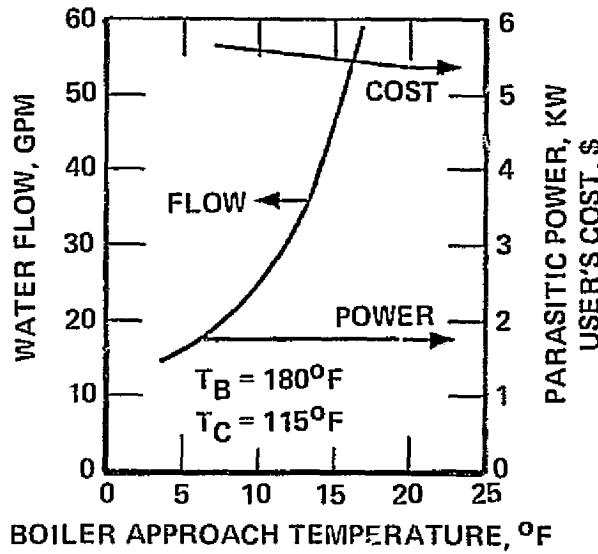
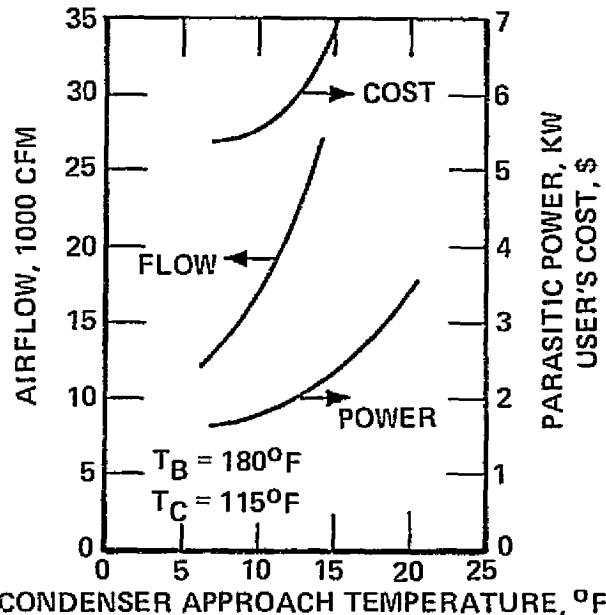
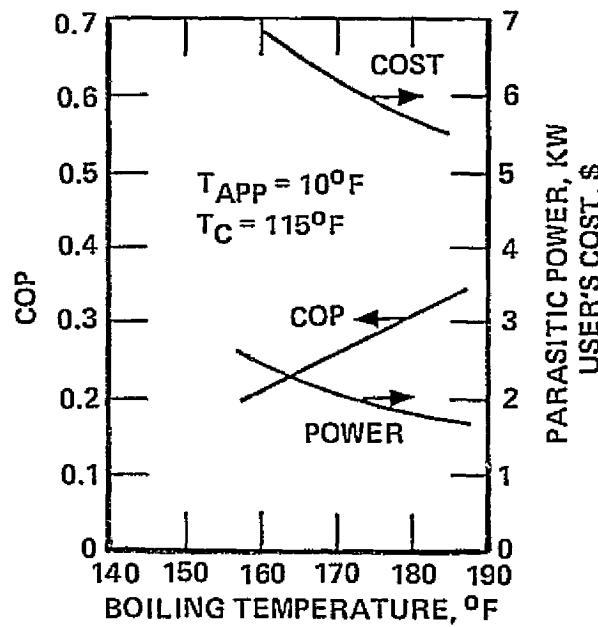
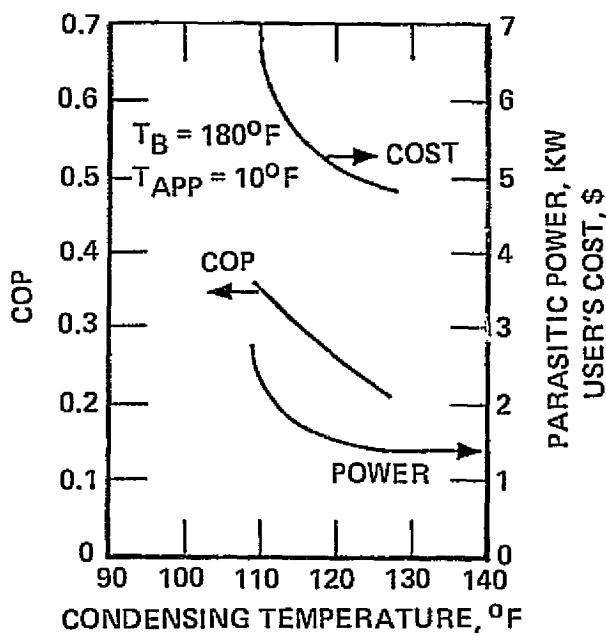
In all cases, and particularly for the dry condenser, the condenser and its fan constitute the most sensitive equipment as they affect overall system COP, cost, and parasitic power. As shown in the upper left figure, lower condensing temperatures yield higher system COP, thus producing lower boiler and condenser heat loads; however, the lower  $\Delta T$  (condensing temperature - ambient air temperature) potential available for heat transfer overshadows this effect. Note the very low COP's (0.2 to 0.35) attainable in the range considered. A condensing temperature of 115°F appears to be a reasonable design point for this dry condenser concept.

For fixed values of condensing and ambient air temperatures, the cooling air flow through the condenser increases rapidly as the approach temperature in the heat exchanger increases. This effect is the primary factor in determining overall system cost and parasitic power as illustrated in the lower left figure.

Similar data for the boiler (shown in the upper right figure) indicate the much lower influence of this heat exchanger on overall system characteristics. Here a high boiling temperature is highly desirable. Boiler design considerations, particularly the requirement for superheating the vapor, limit the boiling temperature to about 185°F, with a hot water source of 200°F at the boiler inlet. For the conditions noted, near the selected boiling temperature an approach temperature of 7.5°F appears reasonable with respect to water flow requirements (see the lower right figure); parasitic power and cost are relatively insensitive to boiler approach temperature.

Similar data for the other three condenser approaches considered were generated and analyzed; these data can be found in the screening analysis task report. Design points were selected for all condenser concepts and are presented and discussed in the following pages.

# TYPICAL DESIGN DATA (DRY CONDENSER)



## COMPARISON OF CONDENSER APPROACHES

Pertinent characteristics of four Rankine air conditioners featuring the condenser concepts considered are summarized below.

The dry condenser yields excessive system costs and low COP's. The high condensing temperatures (115°F) characteristic of this approach result in a COP of 0.33 and excessive condenser size and ambient cooling airflows. As a result of the low COP and high condenser heat loads and very high airflows, the cost of the system is prohibitive.

The addition of a humidifier upstream of the condenser reduces the dry bulb temperature of the ambient air and thus provides an effectively lower temperature heat sink. With this approach, condensing temperatures of 100°F can be achieved without excessively penalizing the system. This reduction in condensing temperature improves cycle COP significantly (from 0.33 to 0.52). As a result, the condenser heat load is reduced considerably, the ambient airflow necessary for cooling also is reduced, and finally, system cost becomes more attractive.

The installed cost of the evaporative condenser system is comparable to that of the Arkla LiBr/H<sub>2</sub>O used as a baseline. The condensing temperature can be reduced to 90°F with an ambient wet bulb temperature corresponding to ARI conditions of 75°F. The COP of this concept is 0.69, which is slightly higher than that of the Arkla unit (0.65). The power requirement is estimated at 1.15 kw (including evaporator fan power); this is slightly higher than that of the Arkla unit (1.06 kw including evaporator fan). This concept represents a significant improvement over the other two concepts and is considered competitive with the Arkla system. Off-design performance must be determined to fully evaluate the advantages of this approach in comparison with the Arkla unit.

In the cooling tower concept, cold water is used as the air conditioner heat sink. The performance of this concept as expressed by COP is the same as for the evaporative condenser--0.69. The installed cost is higher, primarily because of the added cooling tower; the power requirement, however, is substantially lower.

Detailed investigations of the evaporative condenser and cooling tower concepts are warranted to verify by detailed analysis the system cost obtained using the cost model developed earlier in this program. Tentatively, the evaporative condenser approach is selected.

# COMPARISON OF CONDENSER APPROACHES

CAPACITY: 3 TONS

HOT WATER SUPPLY TEMPERATURES: 200°F

AMBIENT AIR TEMPERATURES: 95°F db, 75°F wb

CONDITIONED AIR TEMPERATURES: 80°F db, 67°F wb

PARAMETER	DRY CONDENSER	HUMIDIFIER	EVAPORATIVE CONDENSER	COOLING TOWER
CONDENSING TEMPERATURE, °F	115	100	90	90
CONDENSER (COOLING TOWER) AIRFLOW, CFM	14,050	7780	4050	5000
BOILER WATER FLOW, GPM	29.3	18.4	13.8	13.8
EVAPORATOR AIRFLOW, CFM	850	850	850	850
COP	0.326	0.519	0.691	0.691
ELECTRICAL POWER REQUIREMENTS, KW	1.72	1.46	1.15	0.65
USER'S COST, DOLLARS	5630	3970	3055	3730



## CANDIDATE CONTROL APPROACHES

Control of the baseline system (without provision for system augmentation) is necessary for normal operation when system capacity exceeds the demand. This situation will occur when (1) the house loads are reduced, (2) the boiler temperature is high, (3) the condensing temperature drops, or (4) a combination of the above exists. If the demand exceeds the capacity of the system, augmentation will be necessary.

Two types of control schemes were considered: (1) a modulating control that matches system capacity to the demand, and (2) an on-off control; the system will automatically shut off and start as required by the residence thermal transients.

A modulating control would use signals from the residence temperature, the thermostat setting, and the cool air temperature from the evaporator. Anticipator or duty cycle circuitry will be necessary to ensure stability and control functions. A brief discussion of the alternatives considered to modulate system capacity follows.

Hot Water Flow to Boiler--A reduction in the hot water flow to the boiler will result in a drop in R-11 flow rate to the turbine and a reduction in pressure available at the turbine. The net effects will be lower power loop efficiency and inefficient utilization of stored thermal energy.

Control of Turbine Inlet Pressure--A flow control valve could be used to control turbine inlet pressure with the same effects as above.

Control of Condenser Airflow--Condenser airflow can be controlled by flow control vanes or by a variable (or two-speed) fan. In either case, condenser temperature will increase, resulting in significant performance deterioration and waste of stored energy.

Control of Liquid Refrigerant Flow to the Evaporator--A reduction in evaporator flow will result in an increase in the compressor speed and pressure ratio. Assuming constant condenser temperature, this results in a lower refrigeration loop COP with the same overall effect as for the three previous cases.

Other means of capacity control such as compressor bypass have the same degrading effects on system COP and utilization of solar energy. In addition, a modulating control will increase the duty cycle of the system fans and pumps and the parasitic electrical load. Although it minimizes transient effects, a modulating system is not recommended.

# CANDIDATE CONTROL APPROACHES

## DESIRABLE FEATURES

- OPTIMUM USE OF COLLECTED SOLAR ENERGY
- MINIMUM REQUIREMENT FOR AUXILIARY POWER
- MINIMUM PARASITIC ENERGY
- MINIMUM TRANSIENT EFFECTS

## APPROACHES

- MODULATING CONTROL
  - ANTICIPATOR OR DUTY-CYCLE CIRCUITRY REQUIRED
  - LONG-TERM OPERATION AT LOW COP
  - NOT RECOMMENDED
- ON-OFF CONTROL
  - MINIMUM PARASITIC ENERGY
  - TRANSIENTS TO BE EVALUATED EXPERIMENTALLY
  - SELECTED APPROACH



## BASELINE SYSTEM

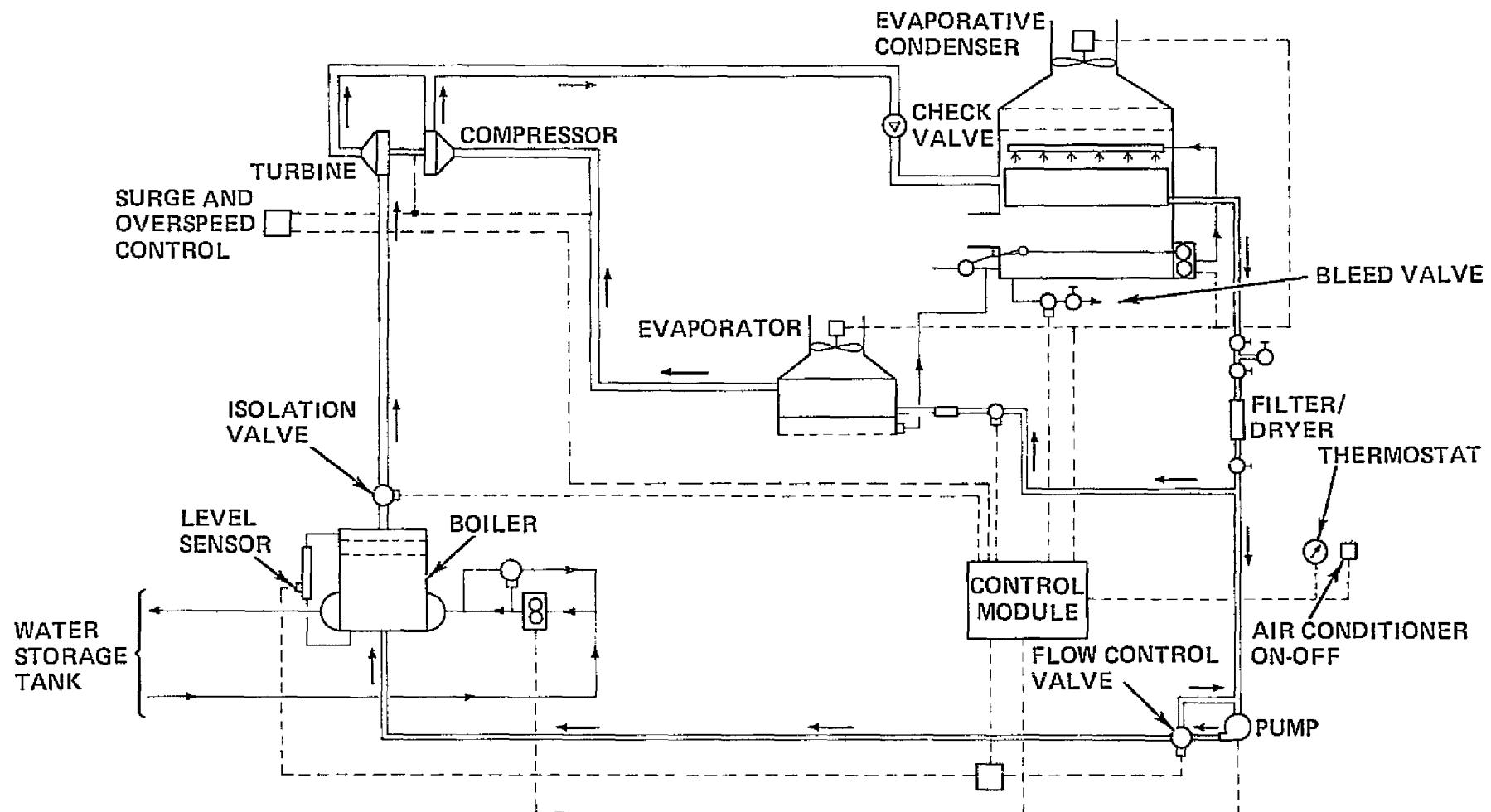
The baseline system control is shown in the schematic below. A thermostat is provided, together with an on-off switch. When the air conditioner is switched on, the control module assumes control for automatic cycling of the system from the thermostat upper and lower set point temperatures. When the residence temperature exceeds the upper thermostat set point, the evaporator and condenser fan and the R-11 boiler feed pump are activated; at the same time, the boiler isolation and evaporator shutoff valves are opened. As the system operates, the residence temperature will drop until the thermostat lower set point is reached. Then the control module will deactivate the fans, refrigerant pump, and solenoid valves, and the system will assume a standby status. With the evaporator shutoff valve opened, the flow of refrigerant to the evaporator is controlled by a capillary tube.

The water level in the sump of the evaporative condenser is controlled by means of a float-actuated water shutoff valve. A fixed bleed is provided to prevent salt accumulation. Water recirculation could be maintained during standby conditions to prevent periodic drying of the water on the surface of the evaporator tubes and to obviate salt deposition and corrosion. In addition, the water flow in the condenser tubes will prevent heating of the condenser during standby. This will provide a significant advantage toward the elimination of startup transients. A check valve in the vapor line to the condenser, together with the refrigerant shutoff valve at the evaporator inlet, will prevent refrigerant transfer to the evaporator during standby and shutdown. Sub-cooled conditions will be preserved in the condenser lower tubes, and a positive head will be available for refrigerant pump startup.

A level sensor on the boiler controls a pump bypass valve and adjusts the refrigerant flow to match the boiling rate. The boiler isolation valve will be closed when the system is on standby to prevent refrigerant migration to the evaporator or condenser. It may be necessary to provide a continuous reduced flow of hot water through the boiler to offset the effects of heat losses and valve leakage during standby. In this manner, the boiler will be maintained at high pressure, and startup transients will be minimized.

Turbocompressor speed is controlled below a maximum value consistent with the aerodynamic and structural characteristics of the turbine and the compressor. A water bypass valve at boiler inlet limits the heat input to the boiler by reducing water flow as the water temperature increases above 210°F.

# BASELINE SYSTEM



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## MAJOR EQUIPMENT SUMMARY

The listing below summarizes the performance characteristics of the major system equipment at design point.

The water flow rate from the solar collector/storage is 13.8 gpm. The temperature drop of the water is estimated at 8.3°F. This flow will be maintained constant over the entire range of operating conditions. A temperature potential of 16°F is available between the hot water stream and the refrigerant vapor for the superheater section of the boiler. This section represents only 3 percent of the total heat transfer conductance (UA) requirement of the entire boiler.

The air exiting the evaporative condenser is saturated with water vapor at a temperature of 80°F. Again, the subcooler section of this unit constitutes only a relatively small portion of the total heat exchanger. Water evaporation rate is estimated as 148.7 lb/hr when the air conditioner operates at the 3-ton level. The average load on the NASA solar house over the period August 18 through 23, 1974, is calculated to be 1.3 tons. This corresponds to a much lower water usage rate.

The evaporator total load is 35,200 Btu/hr. Of this total, about one-third is latent load. The airflow through the evaporator is well below the 1350 cfm maximum value specified by the ARI.

The turbocompressor speed is estimated at 61,316 rpm. Process fluid foil bearings are recommended to obviate lubrication problems. The power developed by the small turbine is estimated at 2.1 hp at design point. Turbine and compressor efficiencies are estimated at 77.1 and 73.6 percent respectively. For large sizes in the 25- to 100-ton capacity, compressor and turbine efficiencies approaching 90 percent could be achieved. A rough estimate indicates that overall COP's as high as 1.0 are attainable for these large systems.

Estimates of the fan and pump power are shown below. These are based on the use of efficient machines designed specifically for energy conservation.



# MAJOR EQUIPMENT SUMMARY

HEAT EXCHANGERS	BOILER	CONDENSER	EVAPORATOR	
HEAT LOAD, BTU/HR	51,172	92,010	35,236	
COLD FLUID	R-11	AIR AND EVAPORATED WATER	R-11	
INLET TEMPERATURE, °F	91.9	95 DB, 75 WB	47.7	
OUTLET TEMPERATURE, °F	183.5	80.2 DB, 80.2 WB	45.6	
FLOW RATE	654 LB/HR —	WATER EVAP.: 148.7 LB/HR AIR: 4050 CFM	494 LB/HR —	
HOT FLUID	WATER	R-11	RETURN AIR	
INLET TEMPERATURE, °F	200	117.5	80 DB, 67 WB	
OUTLET TEMPERATURE, °F	191.7	90.7	55 DB, 53.4 WB	
FLOW RATE	13.8 GPM	1148 LB/HR	850 CFM	
TURBOMACHINES	TURBINE	COMPRESSOR		
FLOW, LB/HR	654	4.94		
INLET PRESSURE, PSIA	84.3	8.1		
PRESSURE RATIO	3.97	2.62		
DIAMETER, IN.	1.77	2.43		
SPEED, RPM	61,316	61,316		
EFFICIENCY, PERCENT	80.1	68.0		
BLOWERS AND PUMPS	CONDENSER BLOWER	EVAPORATOR BLOWER	FREON PUMP	WATER PUMP
FLOW	4050 CFM	850 CFM	654 LB/HR	1000 LB/HR
INLET PRESSURE, PSIA	14.7	14.7	20	14.7
PRESSURE RISE, IN. H <sub>2</sub> O	0.82	0.86	—	—
PRESSURE RATIO	—	—	4.52	2
ELECTRICAL POWER, KW	0.830	0.18	0.10	0.088

## SYSTEM OPTIMIZATION (TASK 5)

The usefulness of the Rankine-cycle solar-powered air conditioner can only be determined through evaluation of its performance over a wide range of interface parameters defined by (1) the temperature of the hot water available to the boiler, (2) the wet and dry bulb temperatures of ambient air, and (3) the wet and dry bulb temperatures of the conditioned space. One of the major characteristics to be used in the evaluation is the energy required for operation in the normal and augmented modes. To this end, candidate augmentation concepts were developed and an optimum approach was selected.

The investigations of Task 5 reported in the following pages covered the following:

- (1) Development of an off-design computer program for system performance prediction (in the normal and augmented modes) over a range of interface parameters including ambient conditions, conditioned space temperature, and heat source water temperature. This program was used to substantiate the characteristics of the system components obtained with the design program.
- (2) Optimization of the turbocompressor design to cover a broad range of conditions and permit operation at low heat source water temperatures.
- (3) Generation of parametric data describing system performance (COP and capacity) over a range of interface parameters.
- (4) Development and evaluation of candidate system augmentation concepts; selection of the optimum approach.
- (5) Generation of auxiliary power requirement data over a range of operating conditions.
- (6) Development of a complete solar collector-thermal storage-air conditioner computer program.
- (7) Evaluation of the baseline Rankine air conditioner over a five-day period simulating the NASA solar house operation.
- (8) Potential evaluation of the air conditioner as a heat pump.

# SYSTEM OPTIMIZATION (TASK 5)



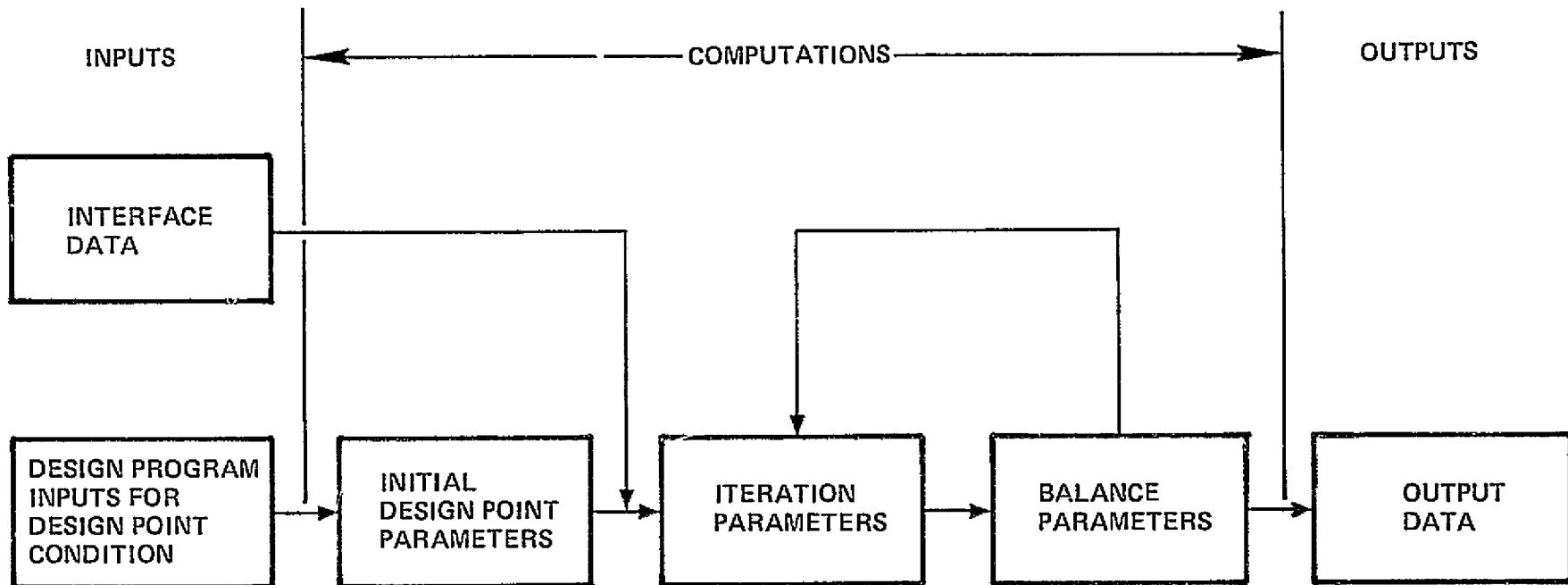
## OFF-DESIGN COMPUTER PROGRAM METHODOLOGY

The off-design computer program was developed to assess the performance of the Rankine-cycle air conditioner over ranges of ambient conditions and solar collector performance. The characteristics of the system major equipment, including all heat exchangers and the turbocompressor, are taken from the design computer program. For heat exchangers, the heat transfer conductance (UA) is assumed constant. This assumption is valid since the refrigerant heat transfer (condensing and boiling) coefficients will remain very nearly the same, and the fluid flow on the other side air or water will be the same. Turbomachine performance maps, presented later, were developed from the design point requirements.

The computations are started by calculating a first set of cycle parameters using the design computer program. Simultaneous iteration of these parameters is performed based on the generalized Newton method of convergence until the flows, temperatures, heat loads, and turbine-compressor power and speed are satisfied. The computer output data include COP and system capacity, as well as turbomachine and system data.

Comparison of the data obtained from the design and off-design programs show excellent agreement. Cycle temperatures are nearly identical; the largest difference is in the boiling temperature, where the value predicted by the off-design program is lower by 1.5°F. Compressor flow is the same in both cases. However, the compressor map used for off-design performance prediction yields a slightly lower efficiency (0.71 vs 0.74). Higher turbine flows were obtained, although the turbine efficiency predicted by the off-design program is higher than that estimated by the procedure used for initial selection of the turbine. This is attributed to minor discrepancies found in the R-11 thermodynamic data. Since more flow is necessary at the turbine, the overall system COP is lower by about 10 percent than predicted by the design computer program. The lower value is believed to be more accurate.

# OFF-DESIGN COMPUTER PROGRAM METHODOLOGY



## COMPRESSOR PERFORMANCE MAP

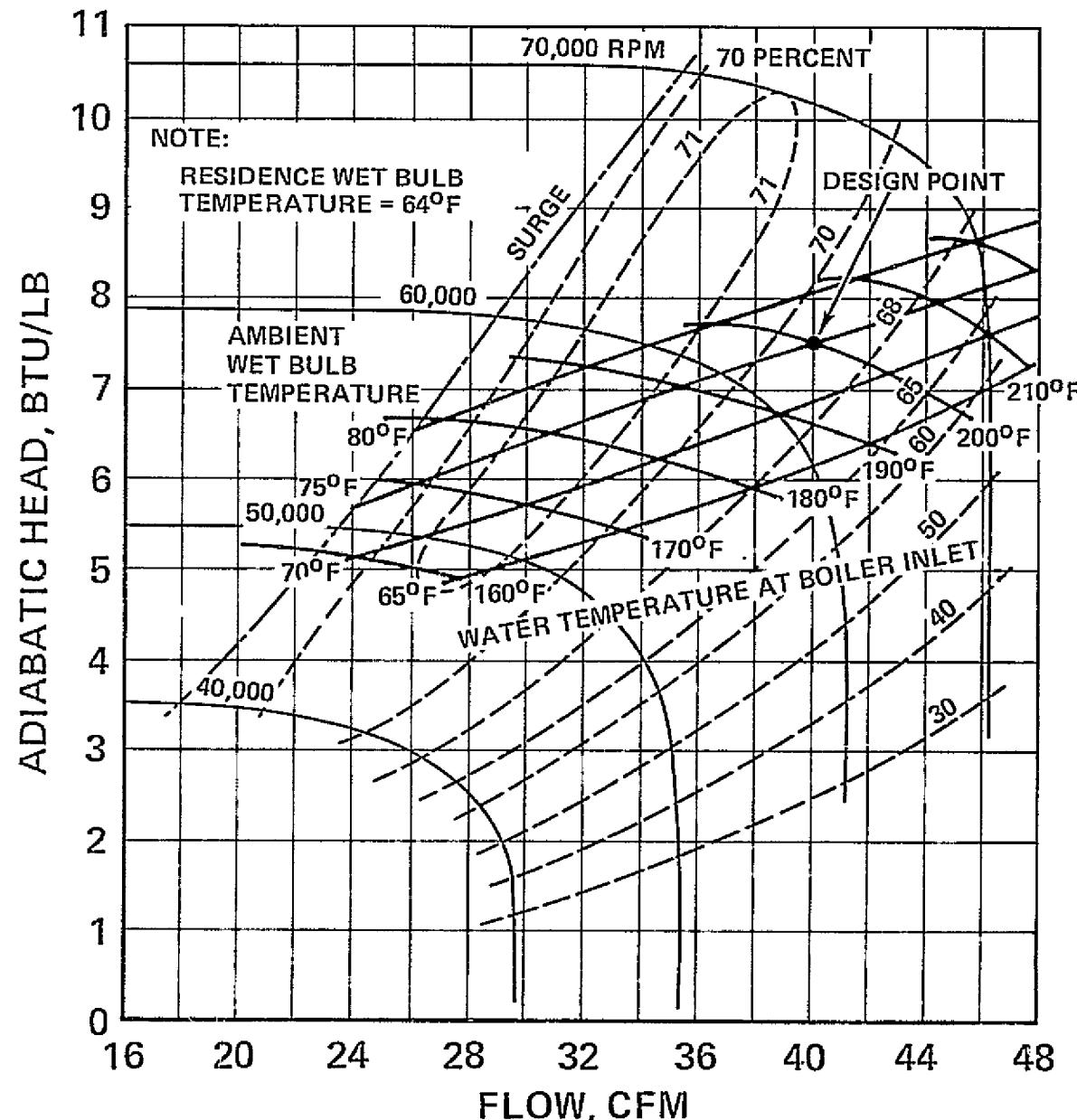
Compressor performance maps were developed using experimental data obtained on similar units. Initially the compressor was designed to yield maximum efficiency at the design point. Preliminary design of the compressor indicated that the efficiency obtained using the procedure cited previously was slightly optimistic. More detailed calculations gave a compressor diameter of 2.43 in. at design speed; design point efficiency is predicted at 71 percent. The efficiency predicted by the design program is 74 percent. These data are considered in good agreement.

System performance prediction with this design-point-optimized compressor indicated that augmentation would be necessary with boiler water temperatures of 190 and 175°F under ambient wet bulb temperatures of 75 and 70°F respectively. This was felt to be too restrictive since ambient wet bulb temperatures between 70°F and 75°F are prevalent over a large portion of the country during the summer months. Consequently, the efficiency of the compressor at design point was compromised to broaden the useful operating range of the system.

The performance map of the broad range compressor is shown below. This map was incorporated in the off-design computer program. Again system operational characteristics were determined; they are shown superimposed on the compressor map. The plot shows that the efficiency of the machine at design point is reduced somewhat; however, operation without augmentation is possible with hot water temperatures as low as 160°F, depending on the wet bulb temperature of ambient air.

Restrictions are imposed by stress limitations in the high water temperature range. Furthermore, the compressor efficiency at higher temperatures is decreased considerably. This does not appear to be a significant disadvantage in the present application, where temperatures much above the boiling point of water are undesirable in view of the water tank pressurization that would be necessary to prevent boiling.

# COMPRESSOR PERFORMANCE MAP



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## TURBINE PERFORMANCE MAP

The turbine was designed to match the speed and power requirements of the compressor at the conditions defined by the design computer program. In the performance of the preliminary design of the turbine, discrepancies were found in the R-11 thermodynamic data used. These discrepancies are related to the values of  $R$  and  $\gamma$  in the region of the vapor dome. The design program uses a value of  $\gamma = 1.11$  published by Allied Chemicals (Genetron 11 thermodynamic properties, 1957). The off-design program calculates the value of  $\gamma$  from the pressure-enthalpy data contained in the same reference. These discrepancies were corrected in final design but were found to have an important effect on turbine performance. As a result, higher turbine flow rates were necessary to furnish the power needed to drive the compressor at the design point.

The turbine is a radial inflow machine with curved blading at the tip to reduce reaction. Turbine diameter is 1.77 in., and turbine efficiency at design point is estimated at 81 percent. This is slightly higher than predicted and offsets the detrimental effect of the lower compressor performance. The overall efficiency of the turbocompressor remains about the same at 0.52 (including a 90 percent mechanical efficiency). Design point flow rate is estimated at 654 lb/hr; this represents a 10 percent increase over the value predicted by the design procedure used previously.

In the turbine performance maps below, the flow factor ( $F_F$ ) and velocity ratio ( $U/C_o$ ) are defined by

$$\frac{W\sqrt{T_o}}{A P_o} = (F_F) \left( \frac{\sqrt{g/R}}{\left(1 + \frac{\gamma-1}{2}\right)^{(\gamma+1)/(2\gamma-2)}} \right)$$

$$C_o = \sqrt{2g H_{ad}}$$

$$U = \frac{DN}{229.2}$$

where  $W$  = flow, lb/sec

$A$  = nozzle area, 0.040 sq. in.

$T_o$  = inlet temperature, R

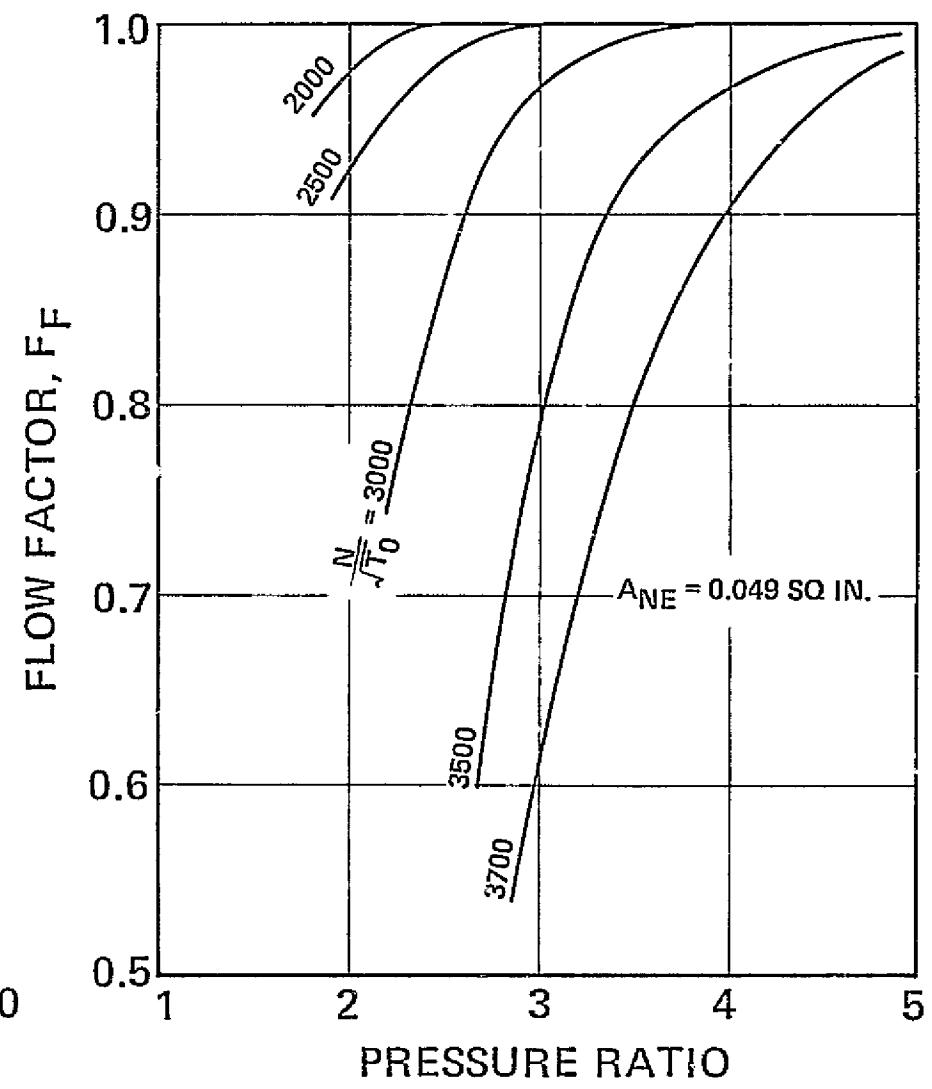
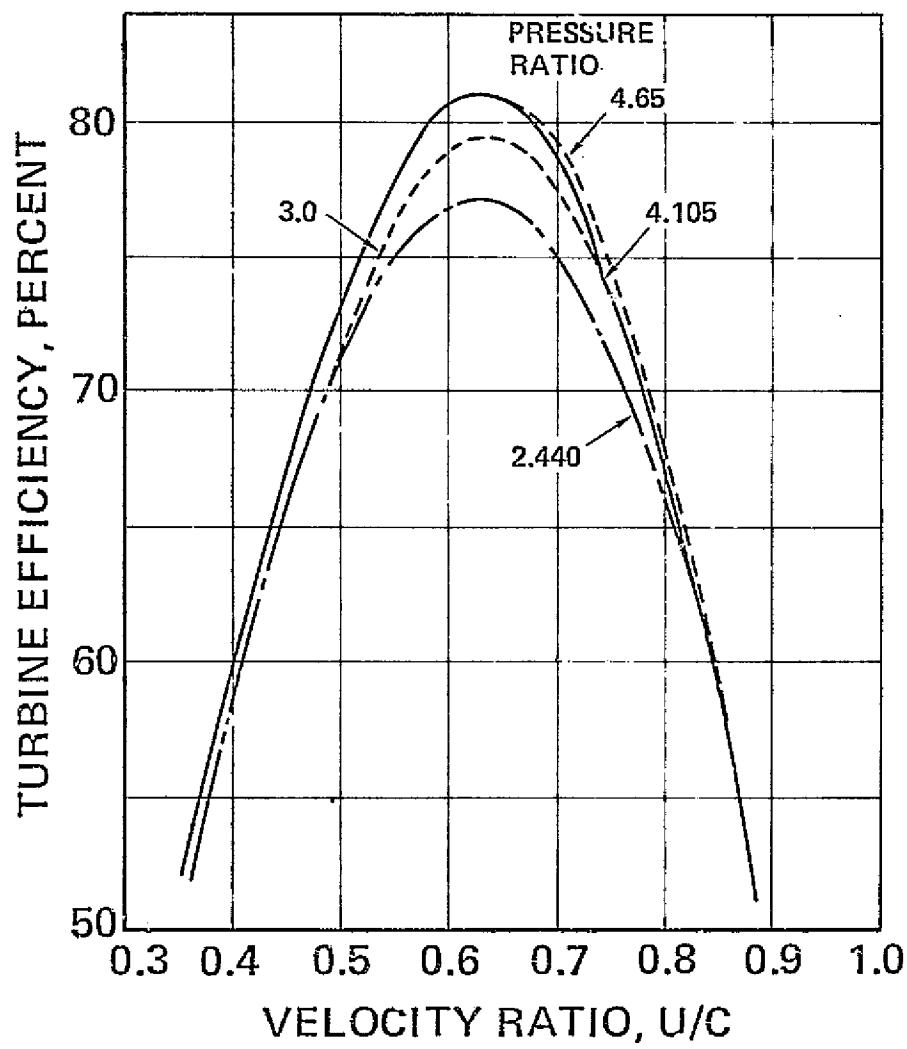
$P_o$  = inlet pressure, psia

$H_{ad}$  = adiabatic head, ft-lb/lb

$D$  = turbine diameter, 1.77 in.

$N$  = turbine speed

# TURBINE PERFORMANCE MAP



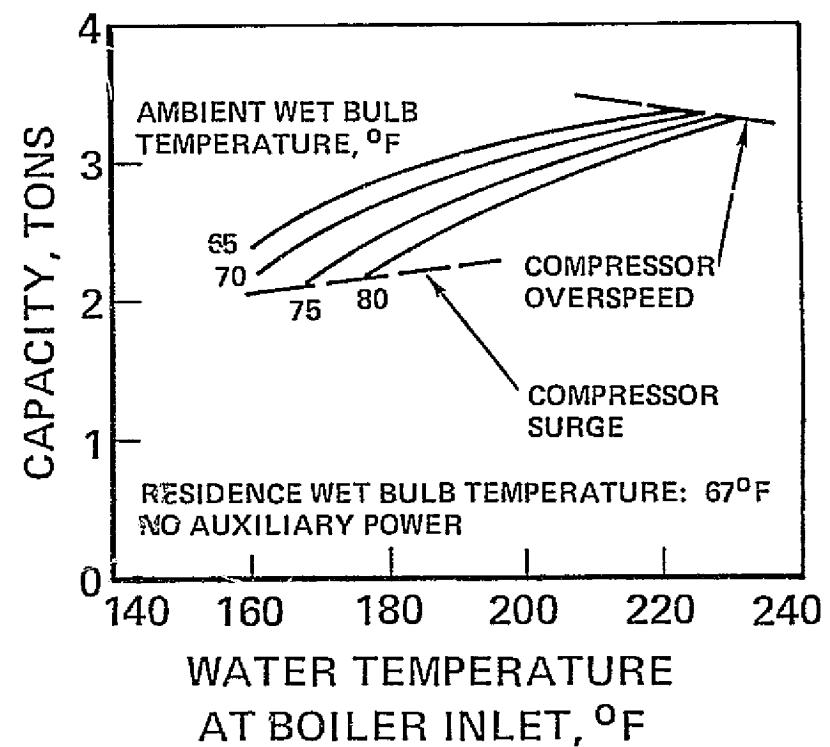
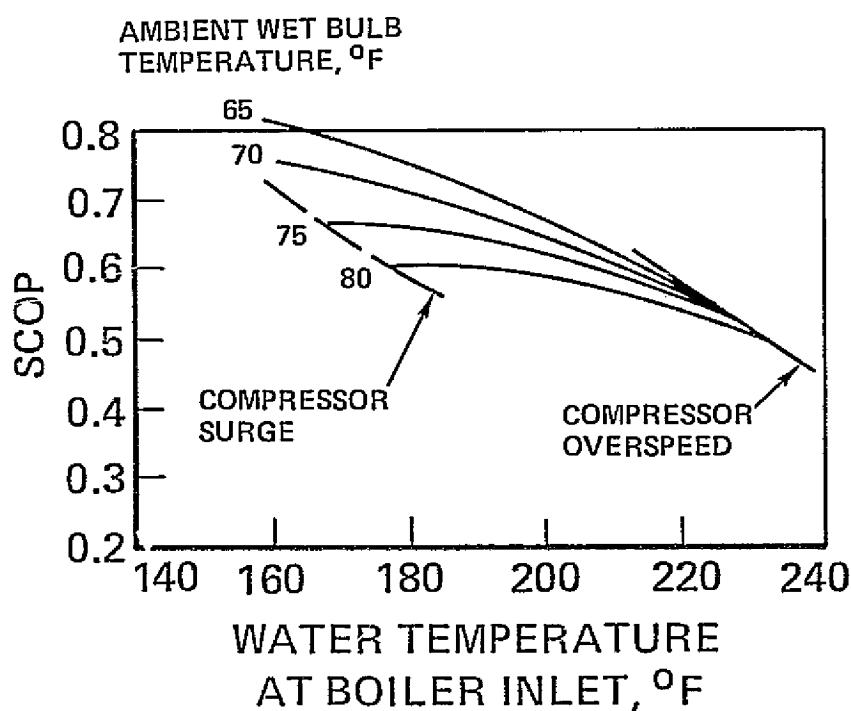
## BASELINE SYSTEM OFF-DESIGN PERFORMANCE

The plot below shows the variation of system capacity and SCOP as a function of the water temperature at boiler inlet for various values of ambient air wet bulb temperature. Operational limitations imposed by compressor surge and turbomachine overspeed are also shown. The data are for a residence wet bulb temperature of 67°F.

Over the range of interface parameters investigated, system capacity will vary from 2 to 3.4 tons. Similarly, the overall SCOP will be between 0.5 and 0.8 with the higher SCOP corresponding to lower ambient wet bulb temperature, higher residence wet bulb temperature, and lower boiler water temperature.

The system is rated at 3 tons under standard ARI conditions (ambient dry bulb and wet bulb temperatures: 95°F and 75°F respectively; return air db and wb temperatures: 80°F and 67°F respectively). Under less severe conditions (for example, at lower ambient wet bulb temperatures), system capacity will be higher than design. Conversely, under less favorable conditions the Rankine air conditioner will, like any other system, degrade in performance and capacity.

# BASELINE SYSTEM OFF-DESIGN PERFORMANCE



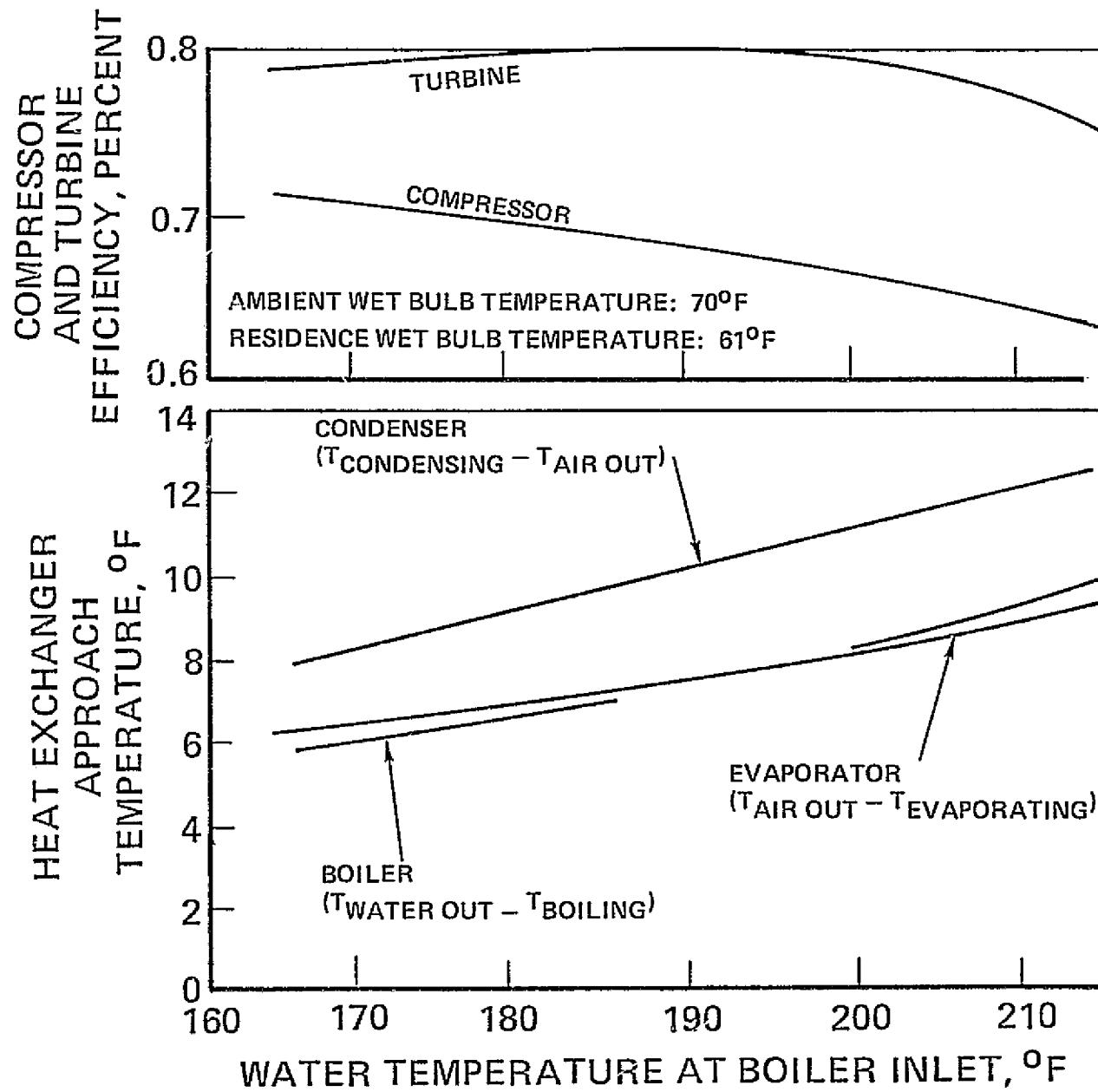
## HEAT EXCHANGERS AND TURBOMACHINE AT OFF-DESIGN

The higher SCOP's achieved at lower boiler water temperatures are due to (1) higher compressor efficiency, and (2) higher heat exchanger effectiveness because of lower loads. The plot below shows turbomachine efficiency and heat exchanger approach temperatures as a function of water temperature for typical operating conditions.

The ambient air wet bulb temperature determines the R-11 condensing pressure. As the ambient wet bulb increases, the system thermodynamic conditions will deteriorate inherently. More power is required from the turbine to match the increasing demand of the compressor. As a result, the capacity and SCOP of the system will drop.

The residence wet bulb temperature has a similar effect on the evaporator. The evaporating temperature of the working fluid drops with the residence wet bulb temperature. Higher compressor lift is necessary to accommodate the lower evaporation pressures.

# HEAT EXCHANGERS AND TURBO-MACHINE AT OFF-DESIGN



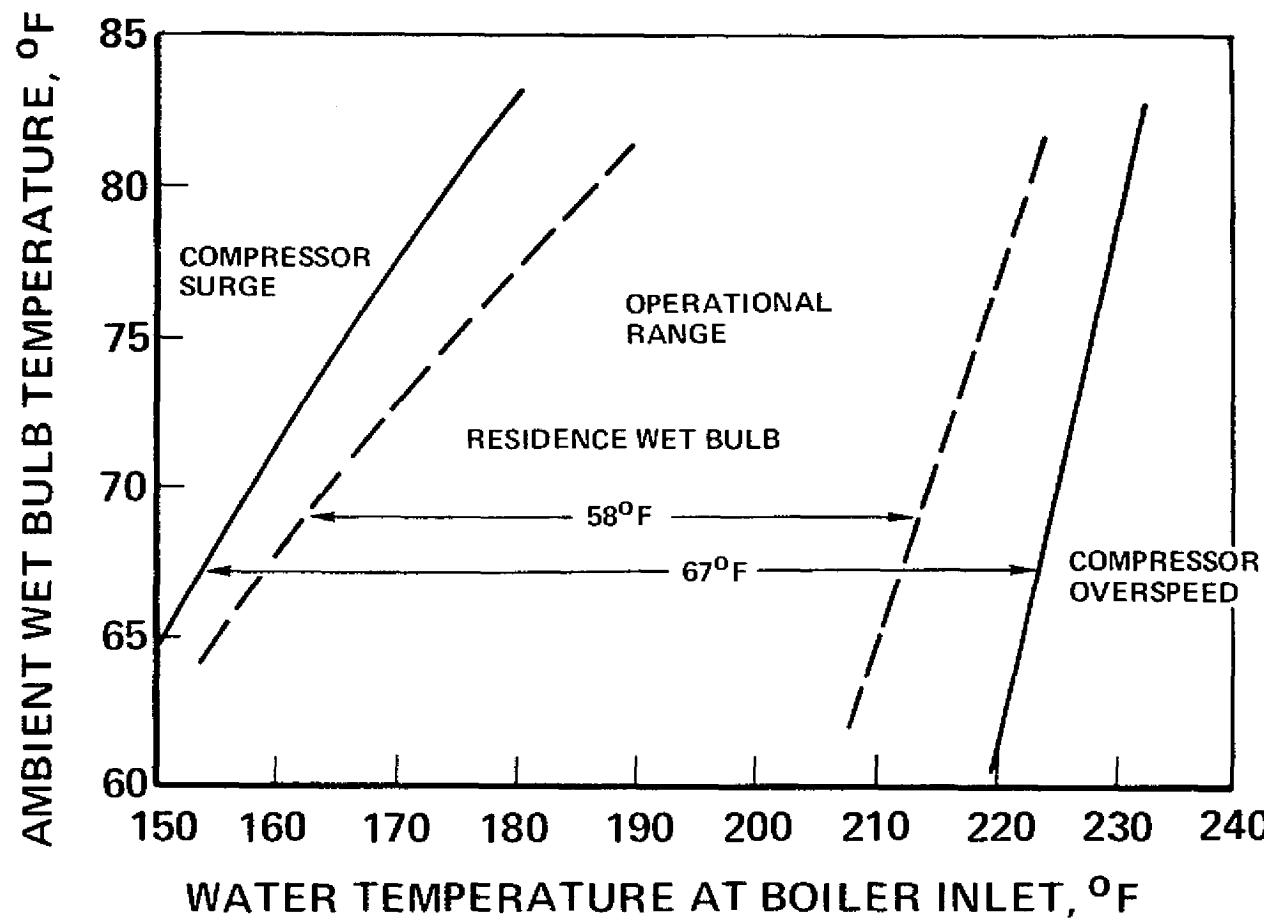
S-490

## OPERATIONAL RANGE WITHOUT AUGMENTATION

Typically, with an ambient wet bulb temperature of 70°F, the air conditioner will run without the need for augmentation with water temperatures at boiler inlet from 160 to 220°F. This represents a very wide range and is attributed to the flexibility of the turbocompressor to find its own operating speed while maintaining high efficiency.

Over the entire range of conditions investigated, the turbine efficiency was found to vary from 75 to 81 percent. By comparison, the compressor efficiency varied from 55 to 71 percent; the lower values occur at high speed corresponding to the upper range of boiler temperatures investigated. This does not appear to be a problem in actual operation since these high temperature levels may never be reached due to solar collector limitations and system heat losses.

# OPERATIONAL RANGE WITHOUT AUGMENTATION



## CANDIDATE AUGMENTATION CONCEPTS

The system must incorporate means of supplementing the solar thermal energy source when necessary. The following approaches were evaluated and compared:

- (a) Thermal energy input into the water loop to the boiler. As an alternate, an electrical heater could be packaged within the boiler itself.
- (b) Auxiliary reciprocating compressor, which assumes the entire load when the Rankine system capacity drops below the demand.
- (c) Electric motor drive integral with the turbocompressor to supplement turbine power and maintain compressor speed at the desired level.

The control module must have sufficient information to (1) determine when to turn the auxiliary power on and off, and (2) control system functions in the augmented mode. A warning light should be installed near the thermostat to indicate when auxiliary power is used. Further means for overriding the automatic mode of operation in the ON or OFF positions should be provided. Such a control is desirable for maximum economy or for system capacity enhancement to meet maximum demand situations which could occur during initial residence cooldown or in extreme climatic conditions.

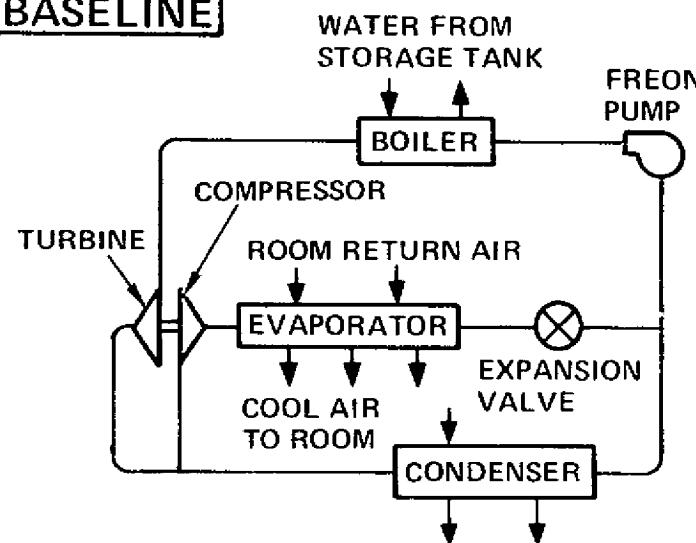
To obviate situations where continuous operation in the augmented mode does not generate sufficient capacity to reduce the residence temperature below the lower thermostat set point, a storage water tank temperature sensor may be desirable. The signal from this sensor would allow activation of the auxiliary only when the water temperature is below a certain value, e.g., 200°F. This signal could also be used to switch off the auxiliary when tank water temperature increases. This will minimize auxiliary energy usage. The auxiliary power and the entire system are switched off when the residence temperature reaches the lower thermostat set point; and the air conditioner control will be reset for baseline operation without the auxiliary power.

A comparison of the three augmentation concepts considered is presented in the following pages.

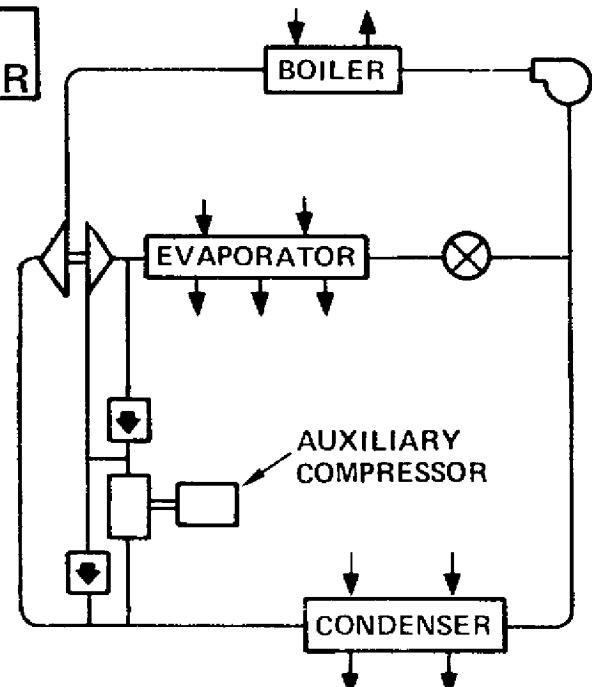


# CANDIDATE AUGMENTATION CONCEPTS

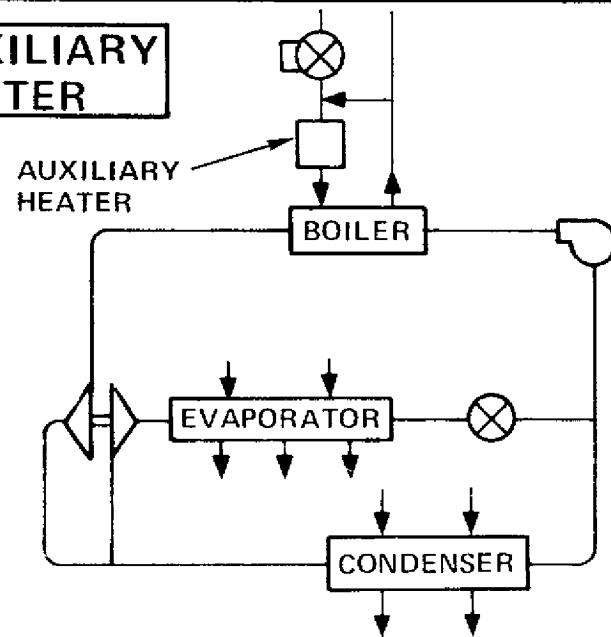
**BASELINE**



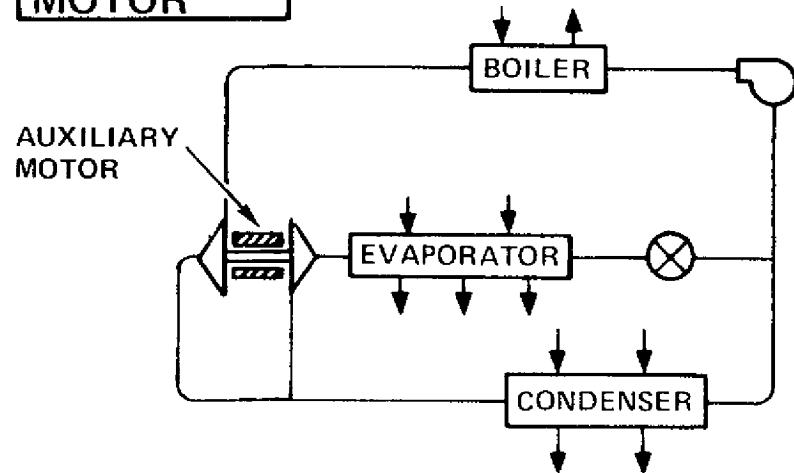
**AUXILIARY COMPRESSOR**



**AUXILIARY HEATER**



**AUXILIARY MOTOR**



## AUGMENTATION BY AUXILIARY HEATER

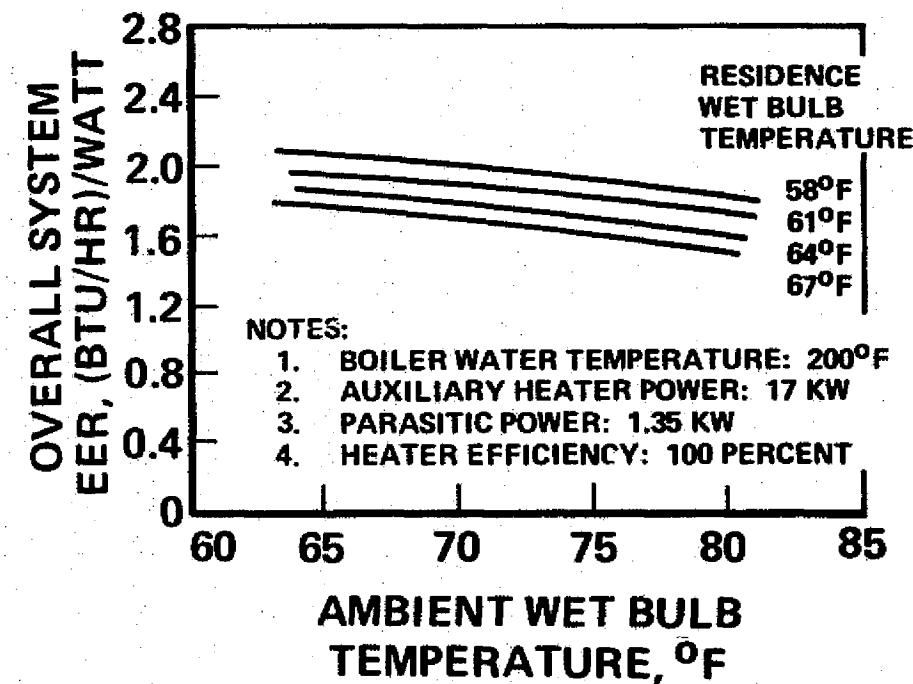
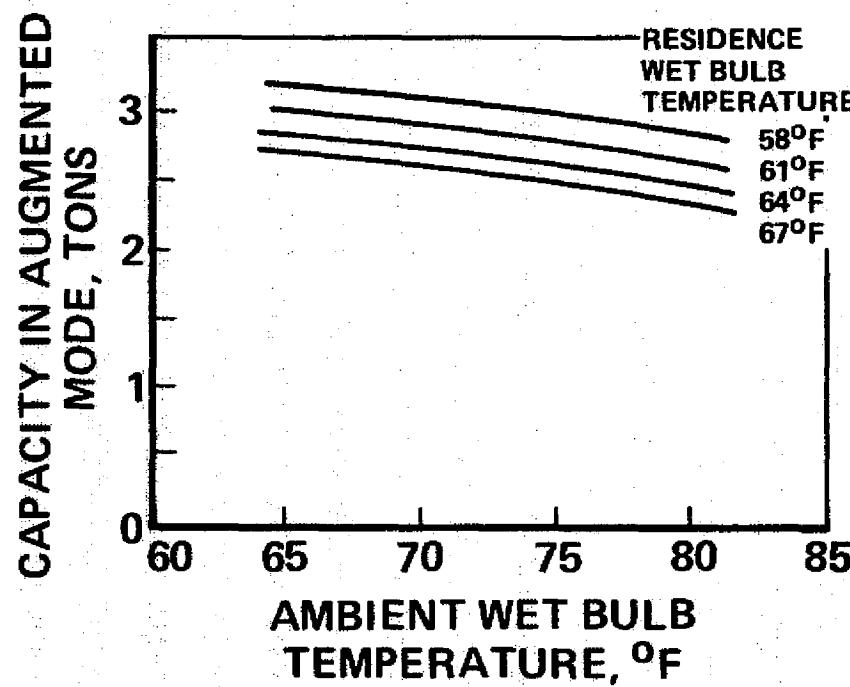
With this method of augmentation, it appears extremely wasteful of energy to activate the auxiliary heater only to enhance system capacity since in the augmented mode all thermal energy necessary for operation is from the auxiliary heater. Auxiliary heater use for capacity increase is therefore not recommended. The signal used by the system for activating the auxiliary heater should be the compressor surge sensor.

The power necessary for system augmentation was taken as constant to obviate the requirements for monitoring and control. Here, a simple on-off control will be adequate. Over the entire range of operating conditions, a 17-kw electrical heater will maintain water temperature at boiler inlet near 200°F. If a gas- or oil-fired heater is used, the system energy requirement should account for the inefficiency of the heater itself. This could be as high as 80 percent through careful design.

The system capacity in the augmented mode is shown below with overall system electrical energy requirement (EER). System EER includes the parasitic power necessary to drive the fans, pumps, and controls and represents the overall COP of the machine in terms of electrical power input.

The power level necessary for operation in the augmented mode far exceeds the power requirements of conventional air conditioners. Commercial units currently marketed have an overall COP (including power for fans and controls) as high as 2.7. This compares to a COP between 0.4 and 0.6 for the Rankine system in the augmented mode with auxiliary heaters.

# AUGMENTATION BY AUXILIARY HEATER



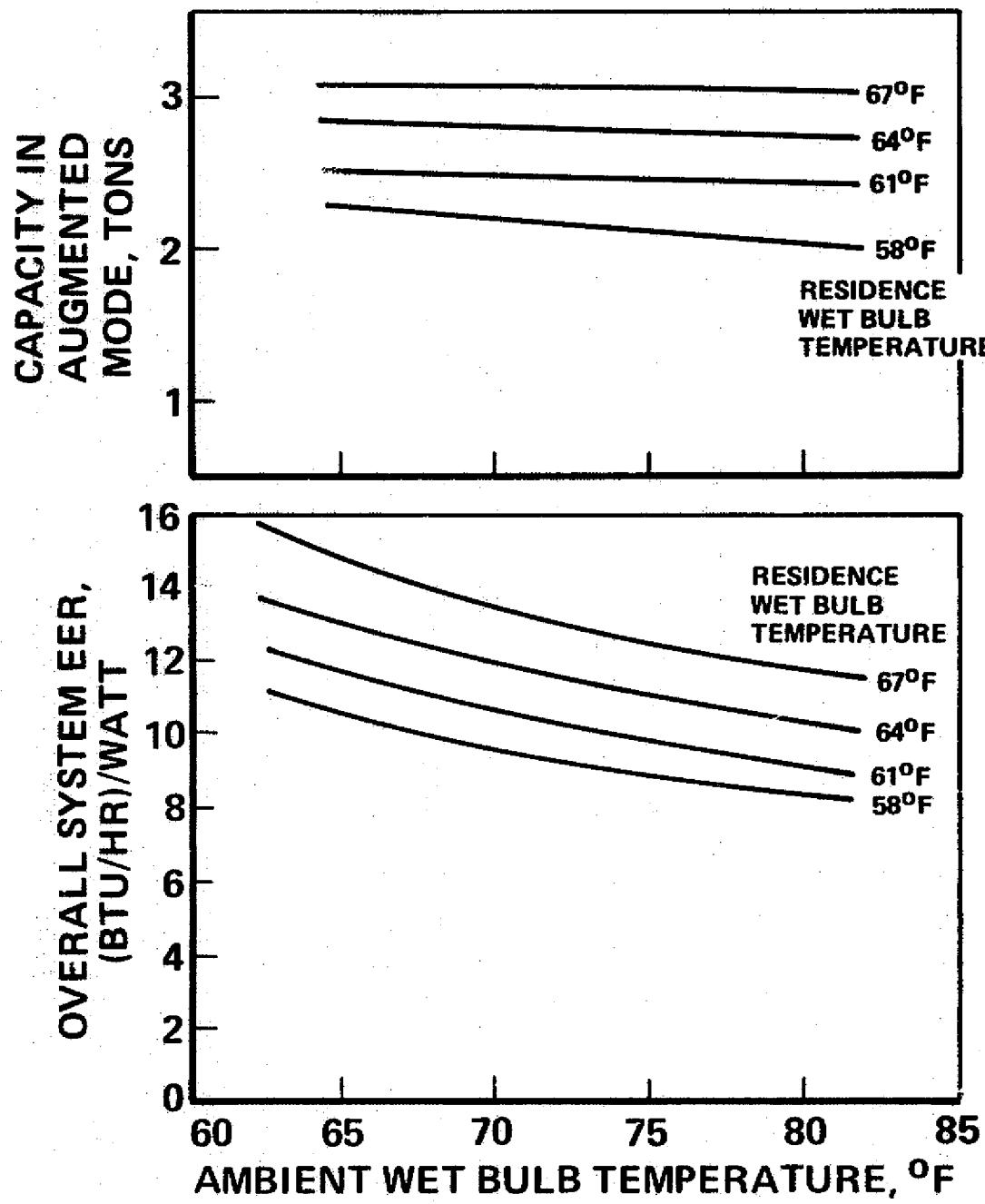
## AUGMENTATION BY AUXILIARY COMPRESSOR

The requirement for auxiliary power is established through monitoring of the parameters described previously. In this case, the Rankine power loop is turned off. This involves stopping the hot water and R-11 pumps and also closing the boiler isolation valve. The entire load is then carried by the auxiliary compressor.

Estimates were made of the auxiliary power necessary in the augmented mode of operation. The design point was selected to give a 3-ton capacity at ARI standard rating conditions. At that point, the isentropic efficiency of the compressor is 80 percent. The motor efficiency was assumed at 70 percent over the entire range of operation.

The system capacity and electrical energy requirement (EER) is plotted below. The EER includes the constant displacement compressor power as well as the parasitic power. The EER shown is 12.8 Btu/hr/watt at standard ARI conditions. This is substantially higher than that of conventional systems (7.5 to 9 Btu/hr/watt), primarily because of the lower condensing and higher evaporating temperatures achieved with the highly efficient heat exchangers used in the system.

# AUGMENTATION BY AUXILIARY COMPRESSOR



S-492



## AUGMENTATION BY AUXILIARY MOTOR

### - TURBOCOMPRESSOR DESIGN -

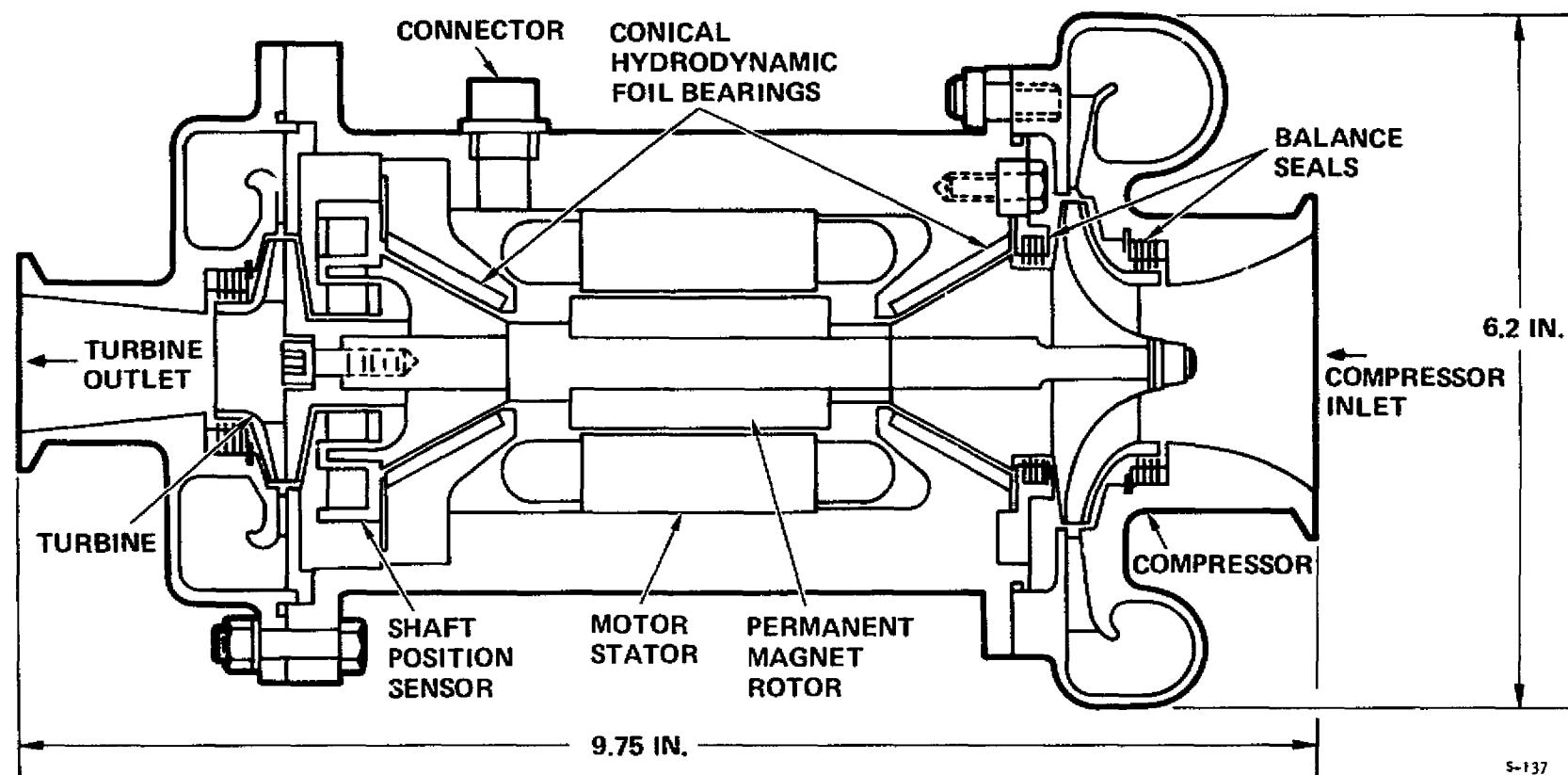
For augmentation by auxiliary motor, a high-frequency motor is packaged as an integral part of the turbocompressor to augment turbine power when necessary. A sketch of the machine, designed to provide 3 tons of air conditioning, is shown below. The compressor and turbine are mounted at either end of the rotor. The motor rotor is on the same shaft between the compressor and turbine. The motor is designed to produce 2.0 kw of power at a speed of 63,000 rpm so that the entire compressor load can be handled by the motor with the turbine windmilling. Electrical power is supplied to the motor from a frequency converter which uses normal house three-wire 230-v, 60-Hz power for conversion to a frequency of 3150 Hz and a three-phase voltage of 120 v. The motor is a six-pole brushless design and uses permanent magnets. In this application, constant speed operation has been selected to simplify the converter circuitry. As discussed later, if the system is used for heating as well as cooling, then a variable frequency converter may be necessary. In this case, motor speed would be adjusted for optimum COP under any heat source temperature by varying the frequency of the power input to the motor.

The motor is highly efficient; testing of similar machines has demonstrated efficiencies higher than 90 percent. Cooling of the motor is by the process fluid.

The rotor assembly is supported by two conical hydrodynamic foil bearings. The use of these bearings minimizes mechanical losses and obviates the requirements for special lubricant. This represents a significant advantage in system design.

Overall dimensions of the unit are shown; the weight of the machine is estimated at 12 lb. The high speed motor is very small, and its cost will be considerably lower than that of a comparable 60-Hz unit. The motor cost savings could be large enough to offset the cost of the frequency converter.

# AUGMENTATION BY AUXILIARY MOTOR — TURBOCOMPRESSOR DESIGN



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AUGMENTATION BY AUXILIARY MOTOR

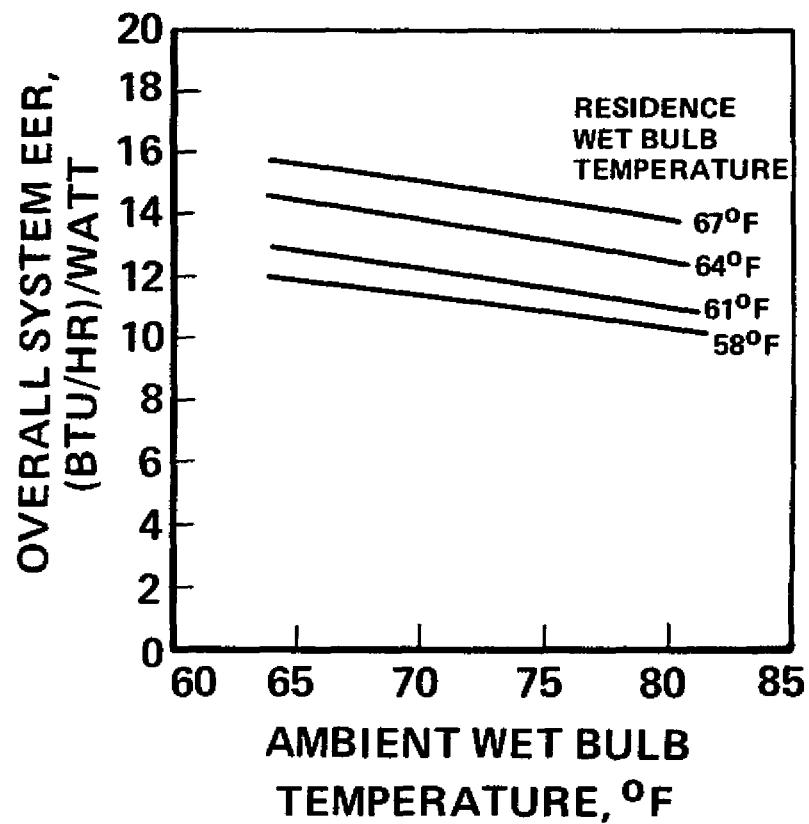
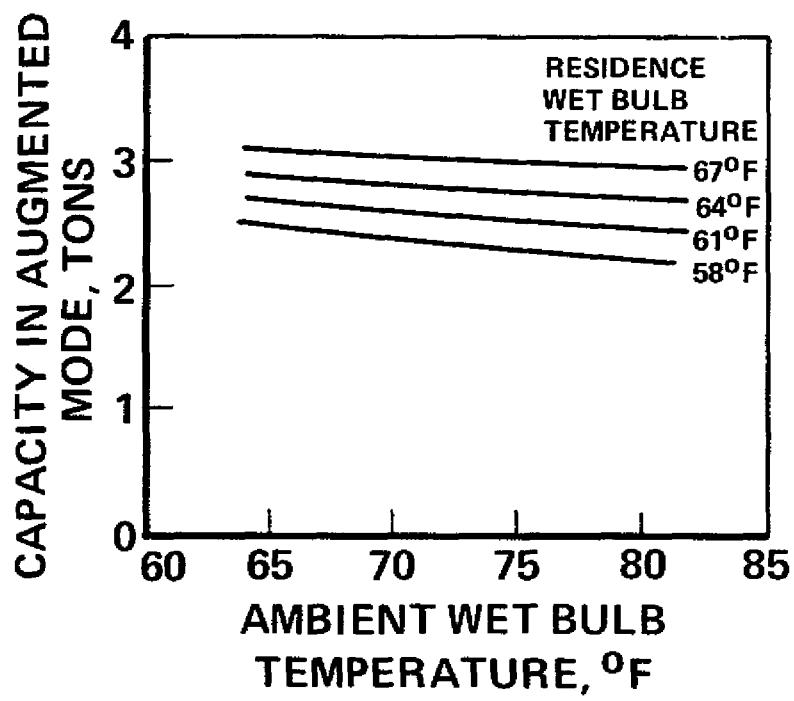
- PERFORMANCE -

System capacity and EER are shown below. Capacity is fairly constant over the entire range of wet bulb temperatures and is more dependent on the wet bulb temperature of the conditioned space. The cooling capacity of the system with the auxiliary motor does not drop as fast as with augmentation with the auxiliary compressor.

As a result, system EER with the auxiliary motor is significantly higher than with the auxiliary compressor at low residence wet bulb temperatures. As mentioned previously, and for the same reasons, the EER in the motor-only mode of operation is considerably higher than for conventional air conditioners.

# AUGMENTATION BY AUXILIARY MOTOR

## — PERFORMANCE —



S-503

## COMPARISON OF AUGMENTATION CONCEPTS

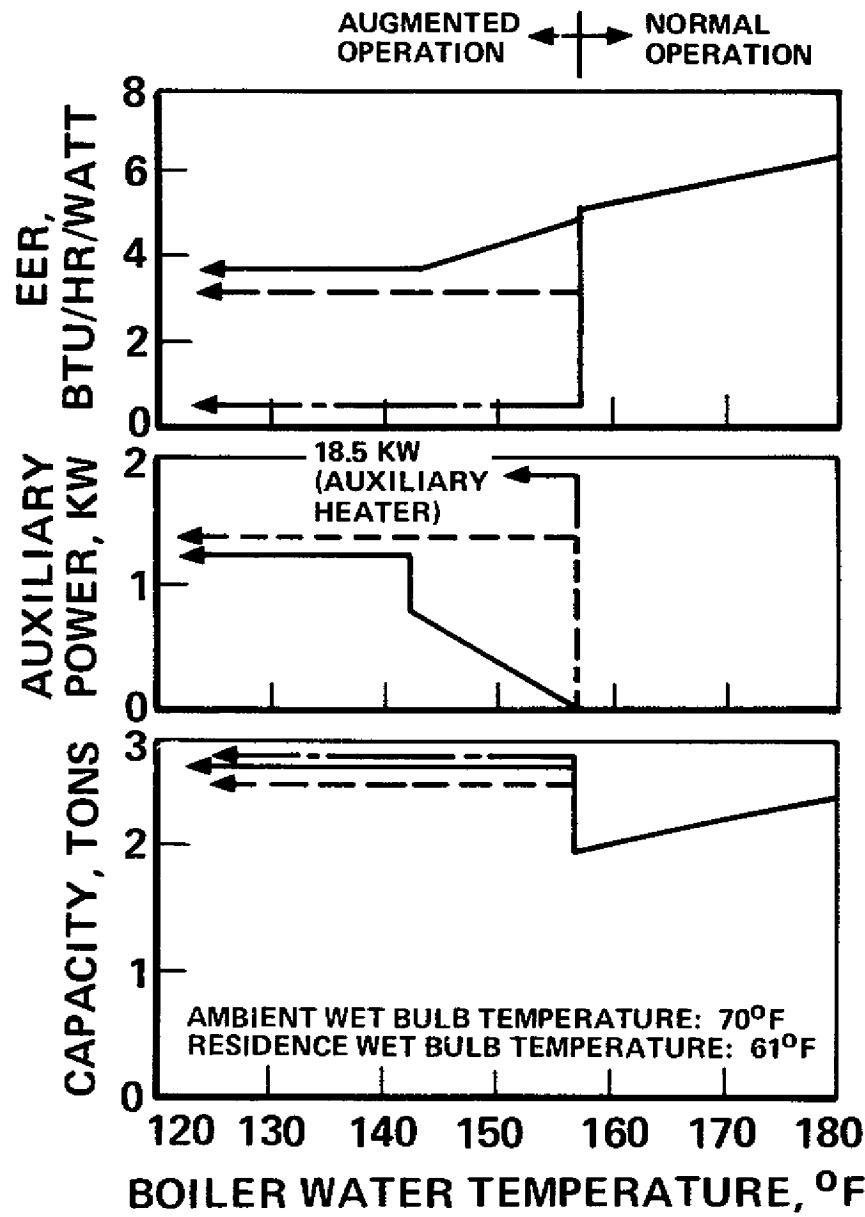
Pertinent system parameters were plotted for the three augmentation concepts corresponding to typical ambient and residence wet bulb temperatures over the range of boiler water temperature of interest.

The auxiliary heater approach is extremely wasteful of energy since the auxiliary heat is used to produce power in a loop which typically has an efficiency of about 10 percent. This is evidenced by the EER characteristic of this approach. Even if gas or fuel oil were used at a unit cost of 25 percent of that for electricity, this approach is not comparable to either of the other two concepts considered. The efficiency of the auxiliary compressor/motor at design point is about the same as that of the converter/motor/centrifugal compressor (0.56). As a result, these two approaches have similar characteristics. At water boiler temperatures below 145°F, the auxiliary motor concept has higher capacity and slightly lower power requirement due to better off-design characteristics. As a result, the auxiliary motor concept has an EER which is 15 percent lower than that of the auxiliary compressor--a significant performance advantage.

Where the auxiliary motor approach is decisively better is in the range of boiler water temperatures from 158 to 145°F. Here, considerable auxiliary power savings can be realized through operation of the turbine-motor combinations. This difference will be larger yet at higher ambient wet bulb temperature.

For this reason, the auxiliary motor concept is selected as optimum.

# COMPARISON OF AUGMENTATION CONCEPTS



## POTENTIAL CAPACITY ENHANCEMENT

In normal operation, without augmentation, the speed of the turbomachine will drop as less power is generated by the turbine at low boiler water temperatures. This will result in a capacity reduction as shown previously. As the boiler water temperature drops below a certain value, the compressor will surge and augmentation will be necessary for operation. At this point, the motor is activated and controls the speed of the turbomachine at a constant preselected value. In terms of turbine operation, this is analogous to a reduction of power requirement. The turbine will still develop some power, although the motor is activated and controls the speed.

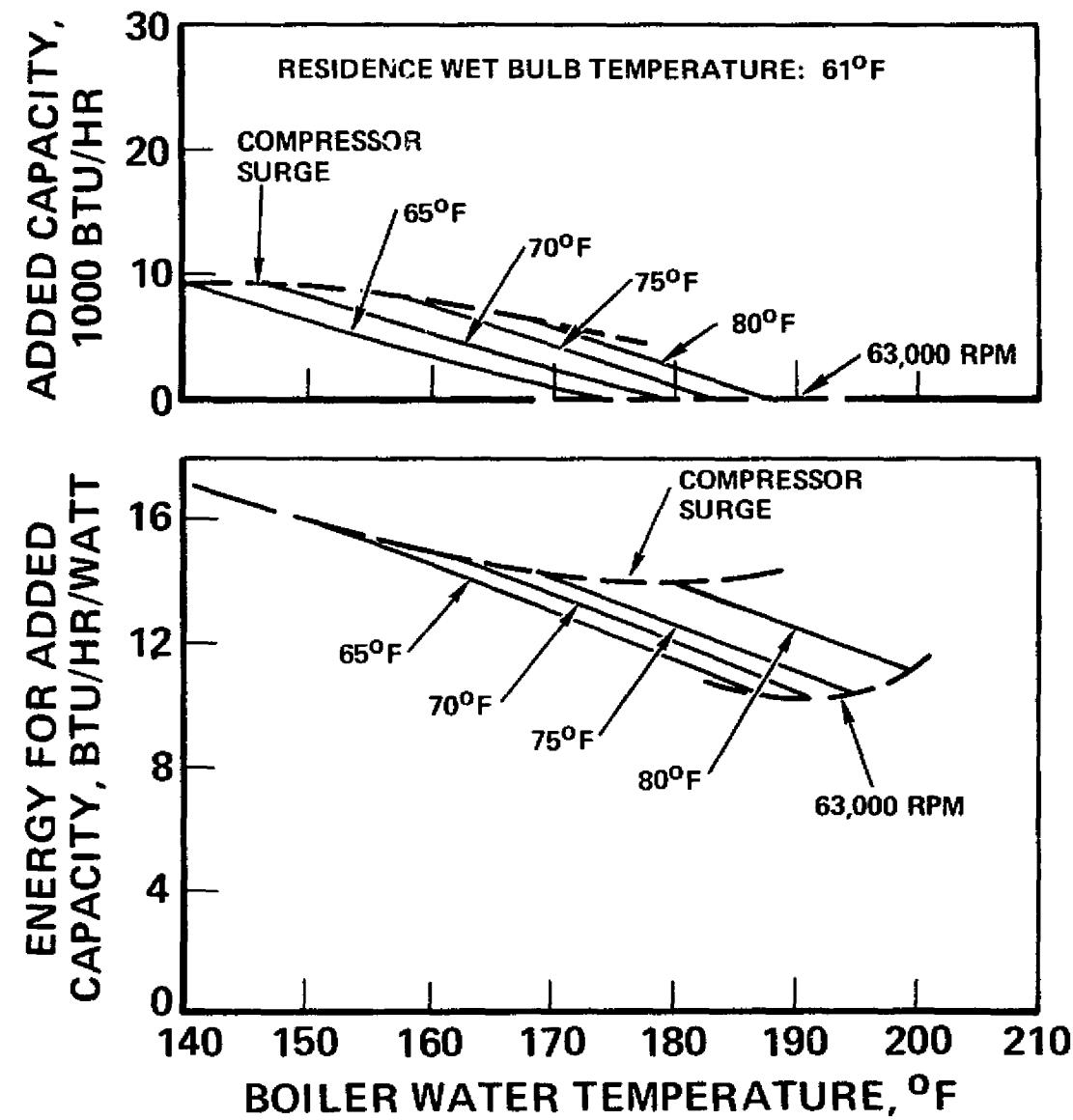
Augmentation by motor activation is possible over the entire range of operation where the conditions are such that the normal speed of the turbocompressor is lower than the selected motor speed. Control in this manner is possible to enhance system capacity even in the range where the system could be operated normally. This would involve starting the motor when the house temperature exceeds the upper thermostat setting by a given preset value, say 2°F.

Shown here is the additional capacity that could be achieved by motor activation in the normal range of operation. The data are presented for a typical value of the residence wet bulb temperature of 61°F. Also shown is the electrical power expended in providing the added system capacity.

The system electrical energy requirement (EER) in the non-augmented mode varies between 18 and 26 Btu/hr/watt over the range of boiler and ambient wet bulb temperatures. By comparison, the EER achieved by augmentation in the normal mode is between 9 and 16 Btu/hr/watt. Thus, the added capacity is about twice as costly in terms of auxiliary energy as the baseline capacity. For this reason, automatic capacity enhancement by auxiliary power was rejected. However, provisions are made for manually activating the motor when desired.

Note that conventional systems do not offer this option and that the capacity of such systems also will degrade under operating conditions (ambient and residence temperatures) less favorable than rated conditions.

# POTENTIAL CAPACITY ENHANCEMENT

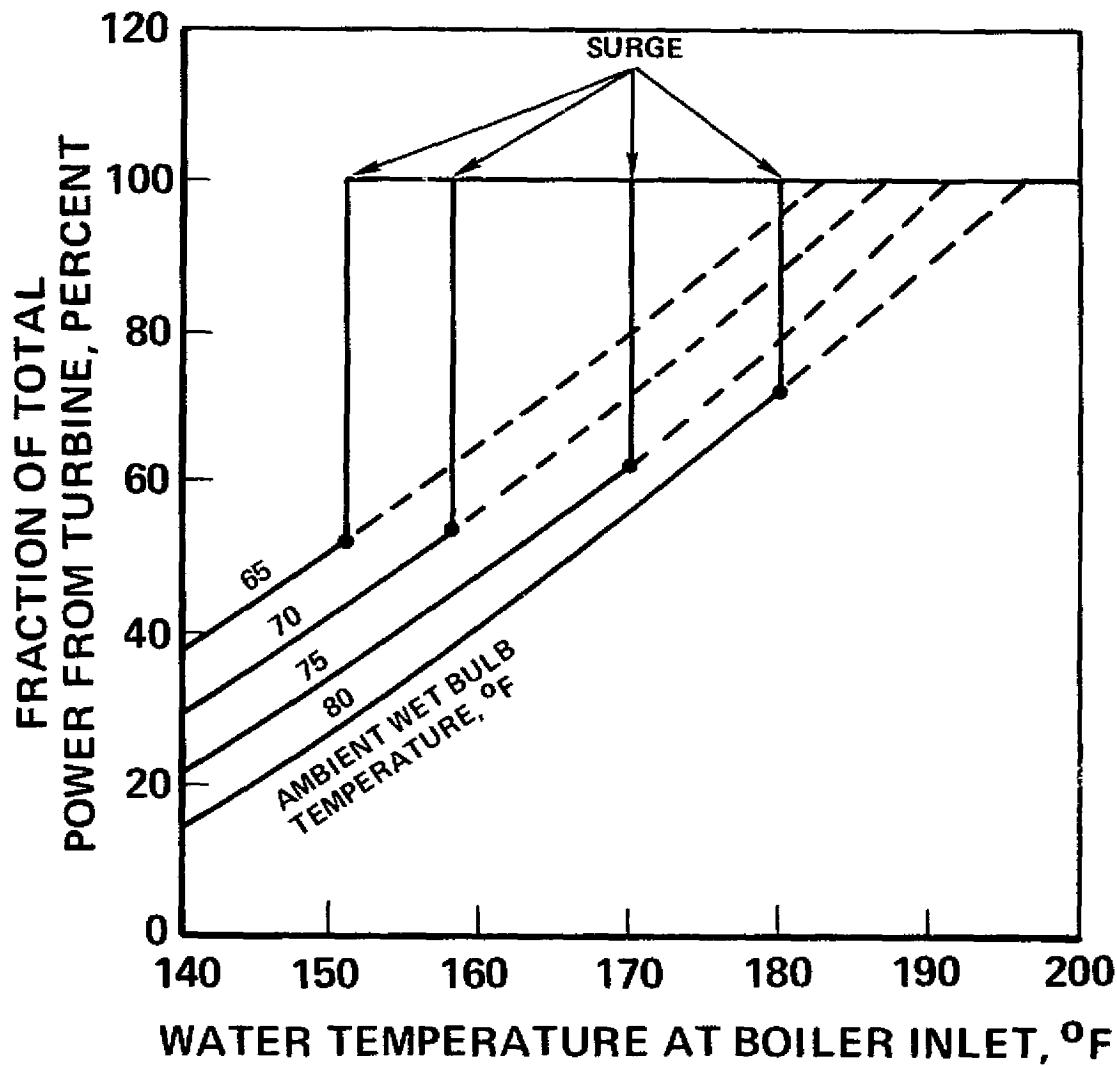


#### TURBINE POWER IN AUGMENTED MODE

When auxiliary motor operation is required to prevent compressor surge, the turbine still has the capability to furnish a relatively large portion of the total power necessary to drive the compressor. This is illustrated for a residence wet bulb temperature of 61°F. Residence wet bulb temperature has only a minor effect on the parameter plotted. The turbine contribution to the total power required is significant at boiler water temperature near that corresponding to compressor surge. As the water temperature drops, the turbine power drops and becomes only a small portion of the total power requirement at high ambient wet bulb temperature. This is due to a rapid deterioration of turbine efficiency at these low temperatures.

It is desirable to disengage the turbine at low efficiency to conserve solar thermal energy, even though it is in a low grade form. Yet it is also desirable to operate the system with the turbine at water boiler temperatures as low as 140°F under conditions of low wet bulb temperatures. A compromise solution, which can be implemented with simple control circuitry, is to disable the turbine when the boiler water temperature drops below 145°F. Under ambient wet bulb temperature conditions representative of average values in a hot humid climate (70°F), it is estimated that 35 percent of the total compressor power is developed by the turbine at a water temperature of 145°F.

# TURBINE POWER IN AUGMENTED MODE



## RECOMMENDED SYSTEM SCHEMATIC

Operation of the system with its control scheme is as follows.

At high boiler water temperature, large quantities of energy are available at the turbine. Turbocompressor speed will be excessive in terms of stress considerations. This condition will occur at boiler water temperatures in excess of 210°F. Turbomachine overspeed protection is provided by a wax element type bypass valve in the hot water line to the boiler. The valve will open at a temperature of 210°F and bypass water around the boiler; the quantity of water bypass will be such as to limit the heat input to the boiler in order to maintain turbocompressor speed below 76,000 rpm.

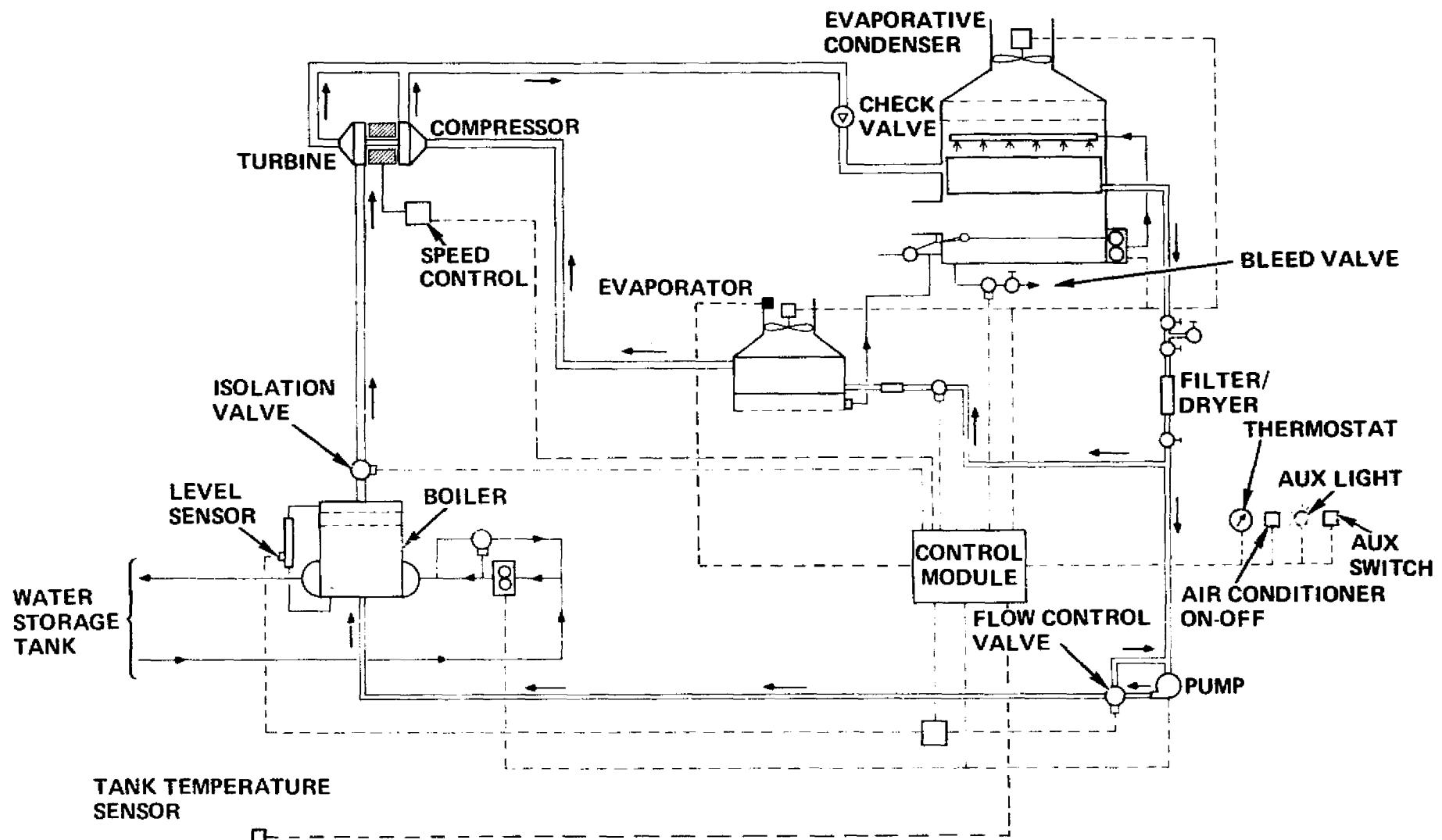
As the boiler temperature drops, turbocompressor speed will also decrease. The capacity of the system and its EER will drop. Under these conditions, the only electric power used by the system is for operation of the fans, pumps, and controls. All turbocompressor power is developed by the turbine. This is the normal mode of operation of the system.

As the turbomachine speed decreases to a value approaching compressor surge speed, a surge sensor will activate the auxiliary motor, which will accelerate the turbomachine to design speed--63,000 rpm. Surge will occur at different speeds and compressor flows depending on the system interfacing parameter. With the auxiliary motor on, system capacity will increase significantly. However, the EER will continue to decrease as the boiler water temperature drops and less power is developed by the turbine and assumed by the auxiliary compressor. Since the turbomachine speed is constant, system capacity is also constant.

A further drop in water boiler temperature will result in system boiler shutdown and operation with the auxiliary motor alone.

The water tank temperature sensor will provide the signal for shutdown of the Rankine power loop when the water temperature drops to 145°F.

# RECOMMENDED SYSTEM SCHEMATIC



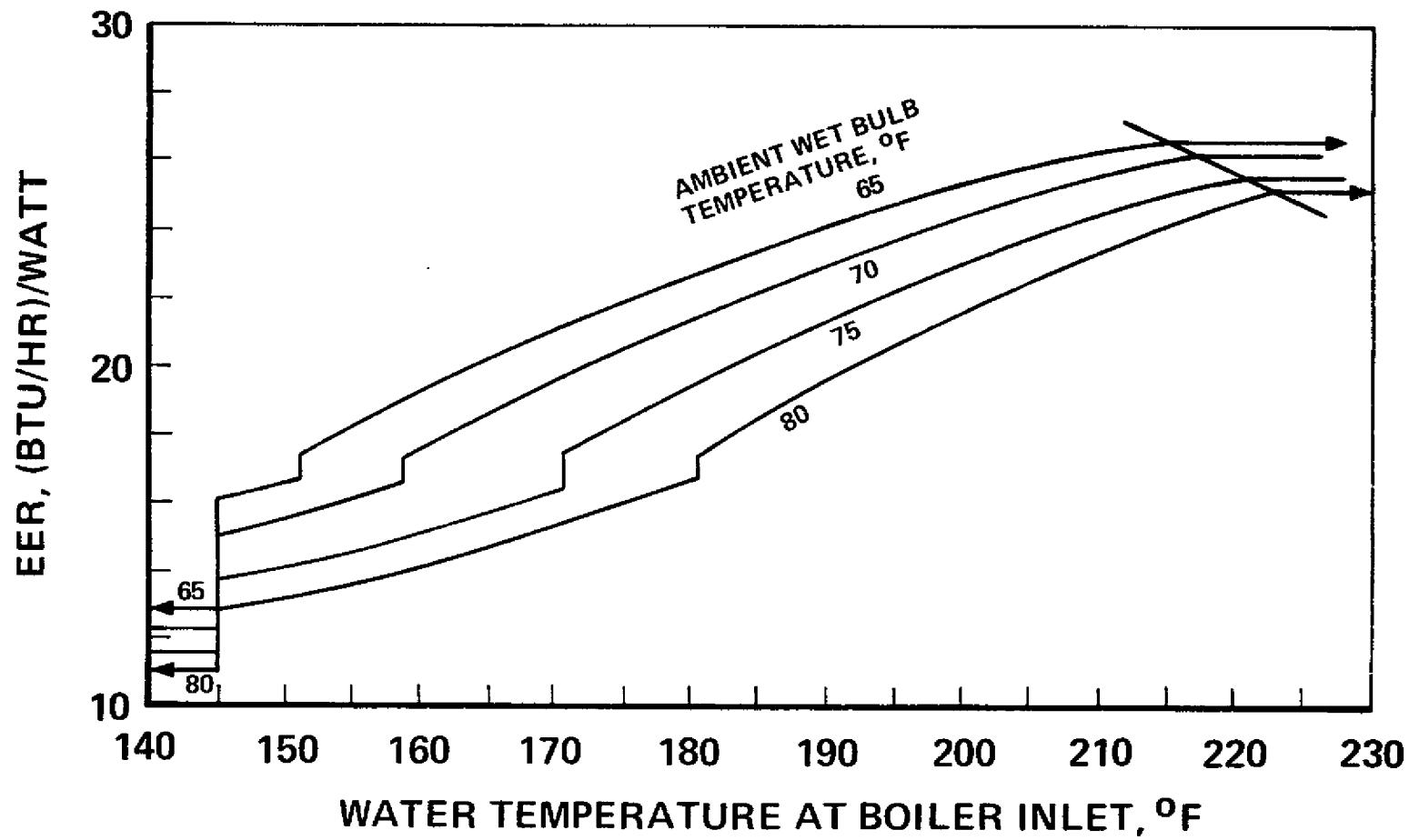
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## RECOMMENDED SYSTEM PERFORMANCE

The overall characteristics of the system in its various operational modes are shown below. Capacity, auxiliary power, and electrical energy requirements (EER) are given over a range of water temperatures at boiler inlet. The data correspond to a residence wet bulb temperature of 61°F. Additional data covering the range of residence wet bulb temperatures from 58 to 67°F can be found in the Task 5 report.

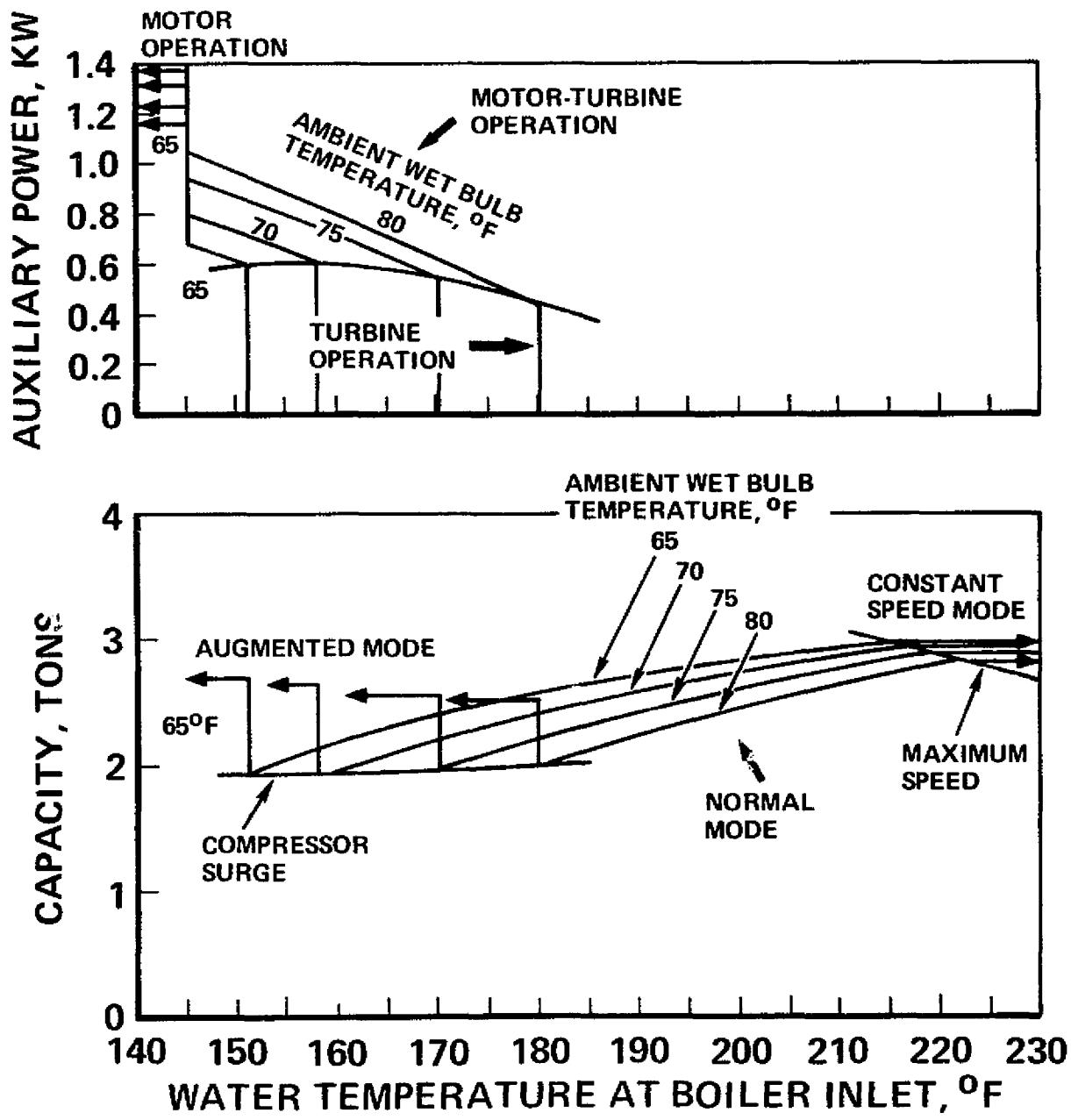
The air conditioner will operate without auxiliary power over a wide range of boiler water temperatures. As a result, system EER is very high--up to 25 to 27 Btu/hr/watt. In the augmented mode with water temperatures above 145°F, the turbine supplies a considerable portion of the power necessary to drive the compressor. However, auxiliary power increases as the temperature drops. Finally, as boiler water temperature drops below 145°F, the boiler is isolated. Under these conditions, the condenser becomes highly effective because of the reduced refrigerant flow; and overall system performance is very high. This is the primary reason for the relatively high EER's achieved in comparison with those of conventional air conditioners. Predicted EER during operation in the all-motor mode at standard ARI conditions is 13.8 Btu/hr/watt; conventional air conditioners under the same conditions have EER's between 7.5 and 9 Btu/hr/watt.

## RECOMMENDED SYSTEM PERFORMANCE



# RECOMMENDED SYSTEM PERFORMANCE

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## COMPARISON OF RANKINE AND ABSORPTION SYSTEM

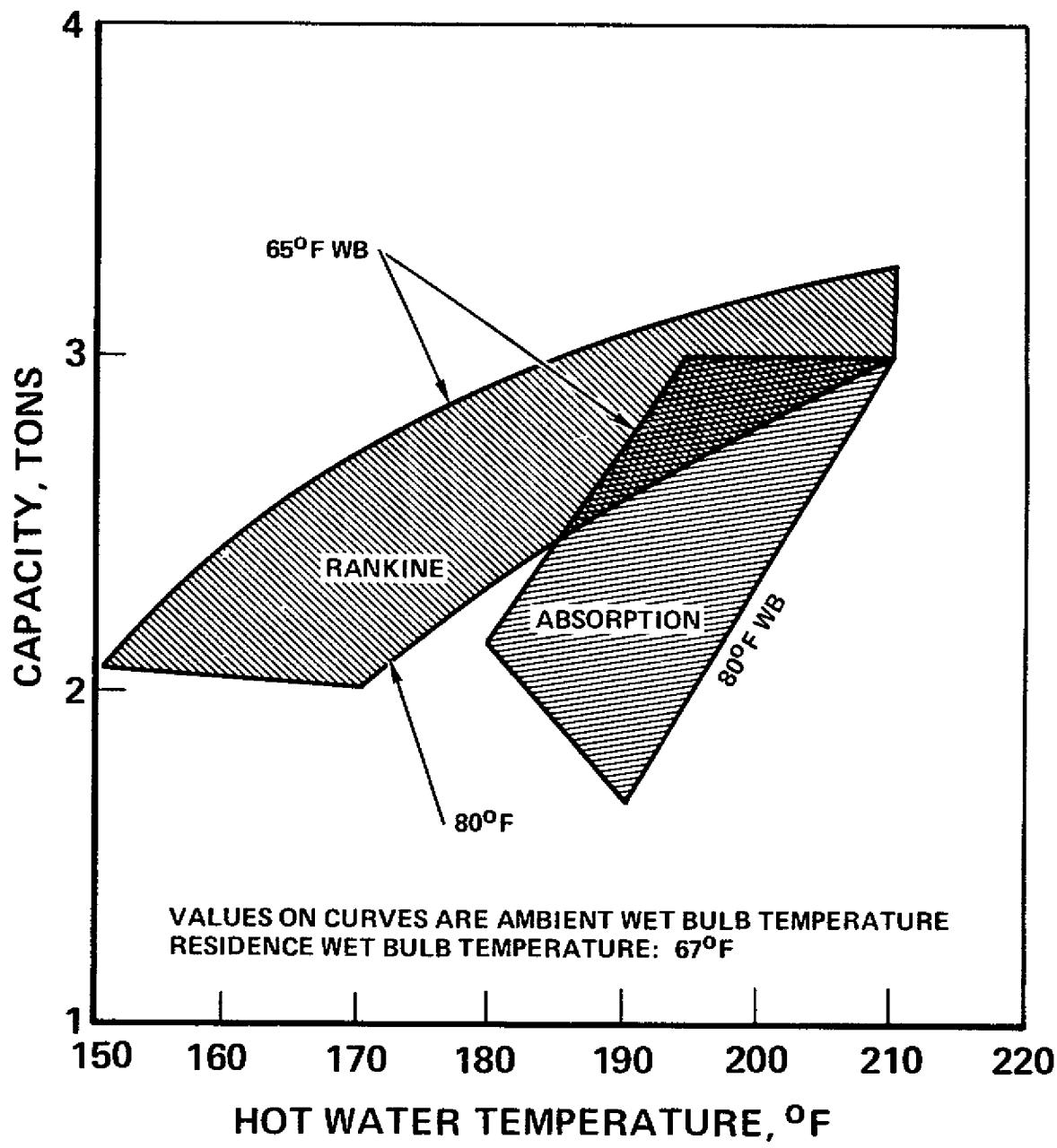
The off-design performance of Arkla Industries' Solaire system, Model 501-WF, was obtained from a recent Arkla brochure (Form No. SP 52T-1). These characteristics are plotted with comparable Rankine system data. The data shown cover the non-augmented mode of operation only. Here it was assumed that the cooling water temperature to the absorption system would be only 5°F higher than the ambient wet bulb temperature. This is representative of a very effective cooling tower.

The operating range of the Solaire system is shown to extend to water temperatures as low as 180°F with a low ambient wet bulb temperature. Capacity drops rapidly with the temperature of the hot water source.

By comparison, the utility of the Rankine system extends to hot water temperatures approaching 150°F, while the capacity remains high over the entire range of water temperature.

In the augmented mode, the absorption system will have an EER comparable to that of the auxiliary heater concept. By comparison, the Rankine system has an EER 6 to 8 times as large. Also, the requirement for auxiliary power with the Rankine system occurs at a hot water temperature about 20°F lower.

# COMPARISON OF RANKINE AND ABSORPTION SYSTEM

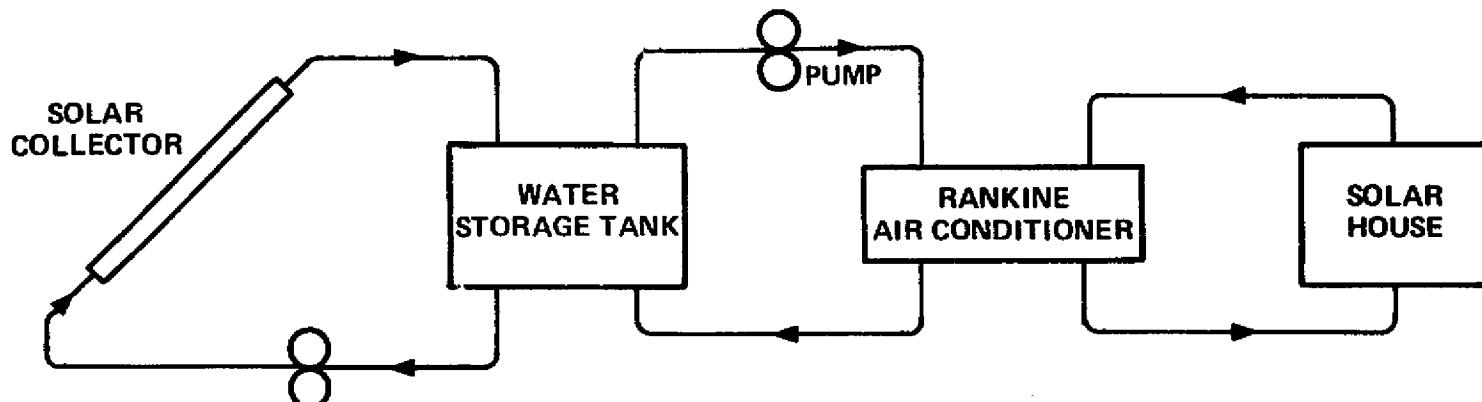


## OVERALL SYSTEM MODEL

The off-design computer program was further developed to model the complete solar system depicted below. The collector, storage tank, and house data obtained by NASA during the period from August 18 through 23, 1975, were reduced to the format necessary for use by the program. In this manner a direct comparison can be made with the LiBr/H<sub>2</sub>O absorption system performance, including COP and auxiliary power usage over the time period considered.

The major assumptions used by the system computer program are listed below. Heat losses in the system pipes and storage tank were estimated from temperature plots furnished by NASA. These losses were then apportioned to yield 200,000 Btu/day, which represents the long-term average obtained.

# OVERALL SYSTEM MODEL



## MAJOR ASSUMPTIONS

- FLAT PLATE SOLAR COLLECTOR: 1300 SQ FT
- WATER STORAGE TANK COMPLETELY MIXED
- WATER TANK CAPACITY: 34,000 LB  $H_2O$
- SYSTEM HEAT LOSSES
  1. TOTAL: 200,000 BTU/DAY
  2. COLLECTOR/TANK PIPES: 40 BTU/HR  $^{\circ}F$  (DURING COLLECTION ONLY)
  3. WATER STORAGE TANK: 25 BTU/HR  $^{\circ}F$
  4. TANK/AIR CONDITIONER PIPES: 30 BTU/HR  $^{\circ}F$
- NASA SOLAR HOUSE DATA USED IN SYSTEM EVALUATION – AUGUST 18-23, 1974
  1. Q COLLECTED
  2. SOLAR HOUSE LOAD
  3. AMBIENT TEMPERATURE AND RH

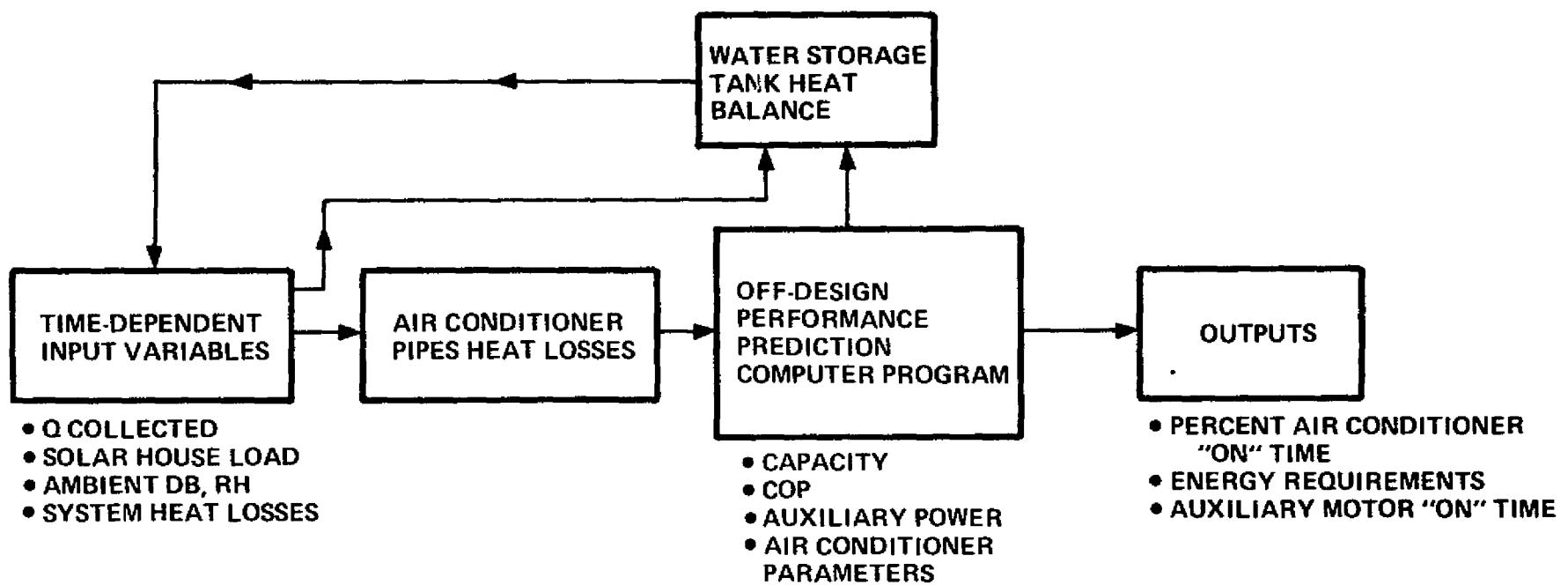


## SYSTEM PROGRAM LOGIC

The methodology used by the computer program is illustrated. The computations are performed as follows. First, system component characteristics are computed using the design computer program. Design point conditions are used for this purpose. Second, a time interval is taken over which the input variables are assumed constant (average value). Then the calculations proceed along the following steps:

- (a) Water temperature at boiler inlet is determined from the water tank temperature and the tank-to-air conditioner heat losses.
- (b) Using the boiler water temperature and ambient and residence wet bulb temperatures, the off-design program determines if augmentation is necessary. System capacity and COP are determined; in the augmented mode auxiliary power is calculated. Water temperature at boiler outlet is also calculated.
- (c) System capacity is compared to the house heat load (input). If the capacity exceeds the load, the fraction of the time period considered during which the air conditioner is "on" is calculated; energy expenditure for the time period is computed. If the capacity is lower than the demand, auxiliary power is used to enhance system capacity.
- (d) The heat used by the air conditioner is calculated.
- (e) The heat losses through the tank-air conditioner pipes, the collector-tank lines (if any), and the water storage tank are calculated. A heat balance is performed on the tank, accounting for the energy collected during the time period considered and the heat used by the air conditioner. Tank water temperature at the end of the interval is determined.
- (f) Summations are made of the energy requirements, and calculations are repeated for the next time interval.

# SYSTEM PROGRAM LOGIC



## SOLAR SYSTEM 5-DAY PERFORMANCE SUMMARY

The performance of the Rankine air conditioner over the five-day period considered is presented here.

The thermal COP of the air conditioner varied from 0.66 to 0.52 during the 5-day period. The low COP's corresponded to the very high wet bulb temperature (80°F) at the end of Day 2 (August 20). The average COP over the entire period was approximately 0.6.

The Rankine air conditioner carried the entire solar house load without auxiliary energy except for very short periods on Day 5 (August 23). On that day, 0.8 kw-hr of auxiliary energy was used. Parasitic power for fans, pumps, and controls is estimated at 1350 watts when the system is in operation. Total electrical energy requirement for the 5-day period is calculated to be 83.7 kw-hr for an average of 16.7 kw-hr per day.

It is interesting to note here that on days when reasonable quantities of thermal energy were collected with the solar collector, the water storage tank temperature from time 0 to 24 hr did not change appreciably or increase (see days 1, 2, and 5). Days 3 and 4 represent worse case situations where the air conditioning load is high and yet little solar energy is available to the system.

The system heat losses (tank and pipes) represent 28 percent of the total energy collected. In system design, careful attention should be paid to this aspect of thermal management to increase the effectiveness of the entire system.

# SOLAR SYSTEM 5-DAY PERFORMANCE SUMMARY

DAY	1	2	3	4	5	TOTAL	DAILY AVERAGE
DATE – AUGUST 1974	19	20	21	22	23		
AIR CONDITIONING LOAD, BTU	348,600	292,200	315,100	425,400	348,400	1,729,700	345,940
Q COLLECTED, BTU	778,000	805,000	478,000	150,000	681,000	2,892,000	578,400
Q LOSSES, BTU	180,800	175,800	179,400	145,900	139,800	821,700	164,300
Q USED BY AIR CONDITIONER, BTU	576,800	512,600	510,400	651,100	541,900	2,792,800	558,600
TANK TEMPERATURE, °F							
AT 0 HR	185.0	185.7	189.0	182.7	164.2		
AT 24 HR	185.7	189.0	182.7	164.2	164.2		
AVERAGE THERMAL COP	0.59	0.58	0.60	0.60	0.60		0.60
ELECTRICAL ENERGY REQUIREMENTS, KW-HR							
AUXILIARY ENERGY	0	0	0	0	0.8	0.8	0.16
PARASITIC ENERGY	15.8	13.2	14.2	21.8	18.7	83.7	16.7
TOTAL	15.8	13.2	14.2	21.8	19.5	84.5	16.9



S-602

## SOLAR SYSTEM PERFORMANCE

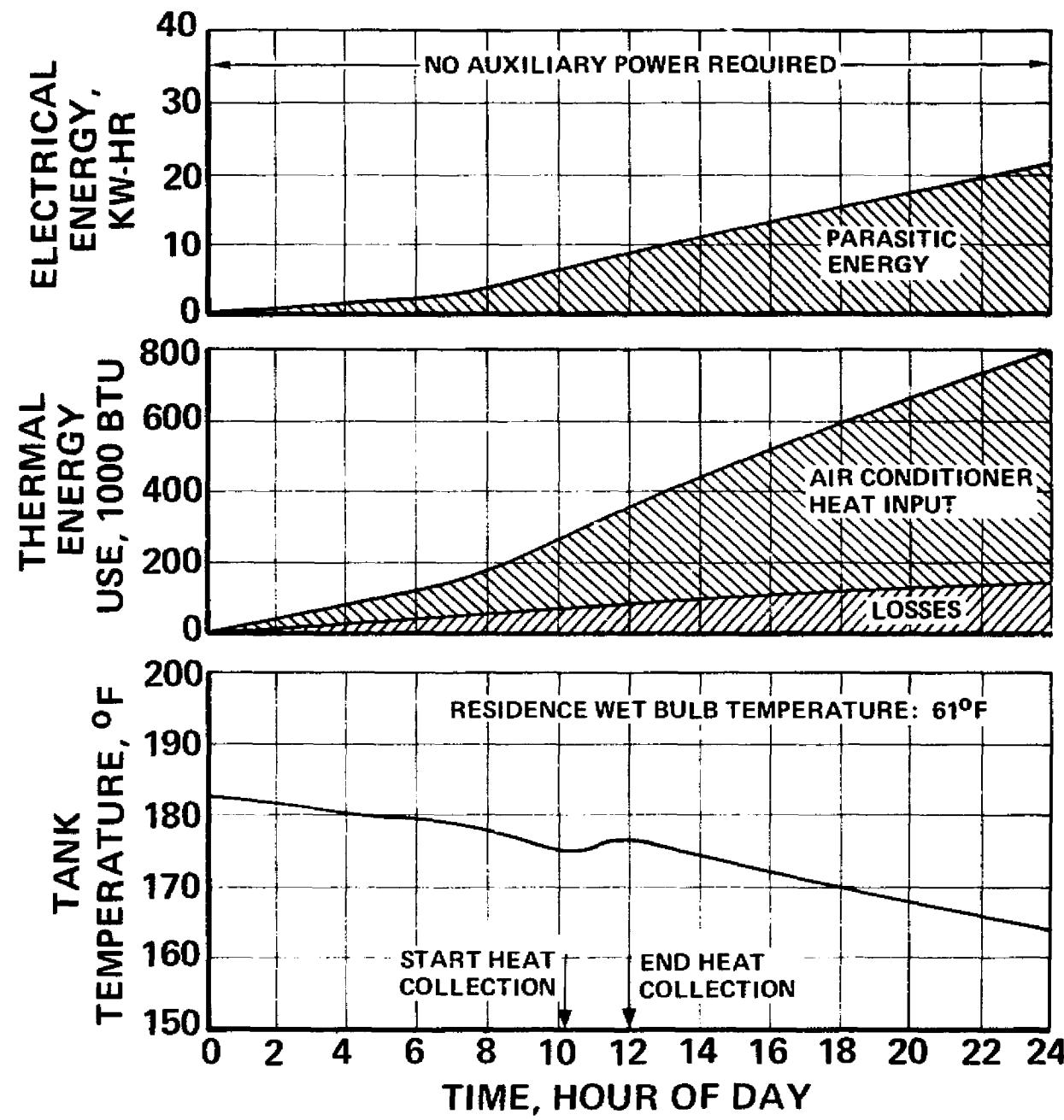
The performance of the Rankine air conditioner under conditions simulating the NASA solar house is presented for a particular day (August 22, 1974) when the thermal energy from the collector was very low (150,000 Btu). Concurrently, during that day the house air conditioning load was very high--425,400 Btu for the day (this load was by far the highest of the 5 days used in system evaluation). This combination of conditions does not represent a likely occurrence and was due to anomalies with the experimental system. This case would be representative of a worse case situation.

As illustrated in the plots below, the Rankine air conditioner operated throughout the day without the requirement for auxiliary power. Water storage tank temperature dropped from 182.7 to 164.2°F providing most of the thermal energy for system operation. This drop in temperature corresponds to a stored energy use of 655,200 Btu.

During that particular day the system thermal losses were low due to (1) the short duration of operation of the solar collector, and (2) the decreasing temperature of the hot water tank.

Parasitic power was 1350 watts while the system operated. Total electrical energy required for the production of 425,400 Btu of cooling is estimated at 18.7 kw-hr for an average EER of 22.7. Average system thermal COP during that day was about 0.6.

# SOLAR SYSTEM PERFORMANCE



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## PERFORMANCE AS A HEAT PUMP

Parametric data were generated to cover the range of conditions defined by the following:

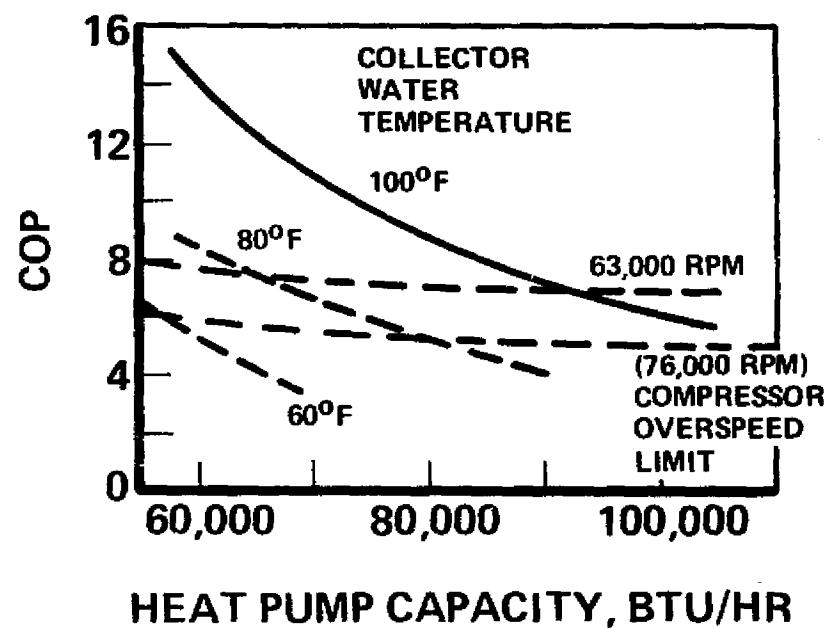
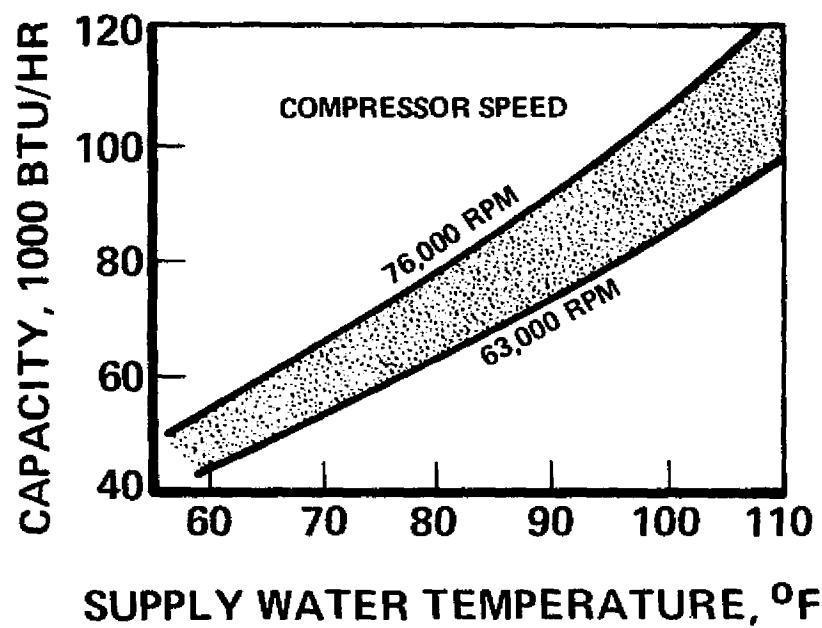
- (1) Heating capacity: 60,000, 80,000, and 100,000 Btu/hr
- (2) Water temperature from the solar heat source: 60, 80 and 100°F
- (3) Residence temperature: 70°F

The heat pump capacity and COP are shown as a function of water source temperature. Operation with low water source temperature is limited by the maximum speed of the compressor selected as 76,000 rpm. A turbocompressor could be designed for operation at higher speeds, thus extending the utility of the system. However, at higher speeds, system COP will drop rapidly due to the low turbine and compressor efficiencies.

Since the system motor is operated at constant speed (63,000 rpm), the system capacity will vary as a function of heat source temperature. To enhance capacity, compressor speed could be increased to 76,000 rpm by providing the necessary electronic circuitry in the frequency converter.

With a machine of this type, minimum power usage will be achieved if the speed of the compressor can be adjusted to provide maximum COP at any operating point. A variable speed motor could be used to control speed using water heat source temperature as the input signal.

## PERFORMANCE AS A HEAT PUMP



## RANKINE SYSTEM AS A HEAT PUMP

Preliminary studies were performed to define the modifications necessary for operation of the system as a heat pump and also to evaluate its performance. In the heating mode, the Rankine power loop is deactivated and the auxiliary motor is used to drive the refrigeration loop compressor.

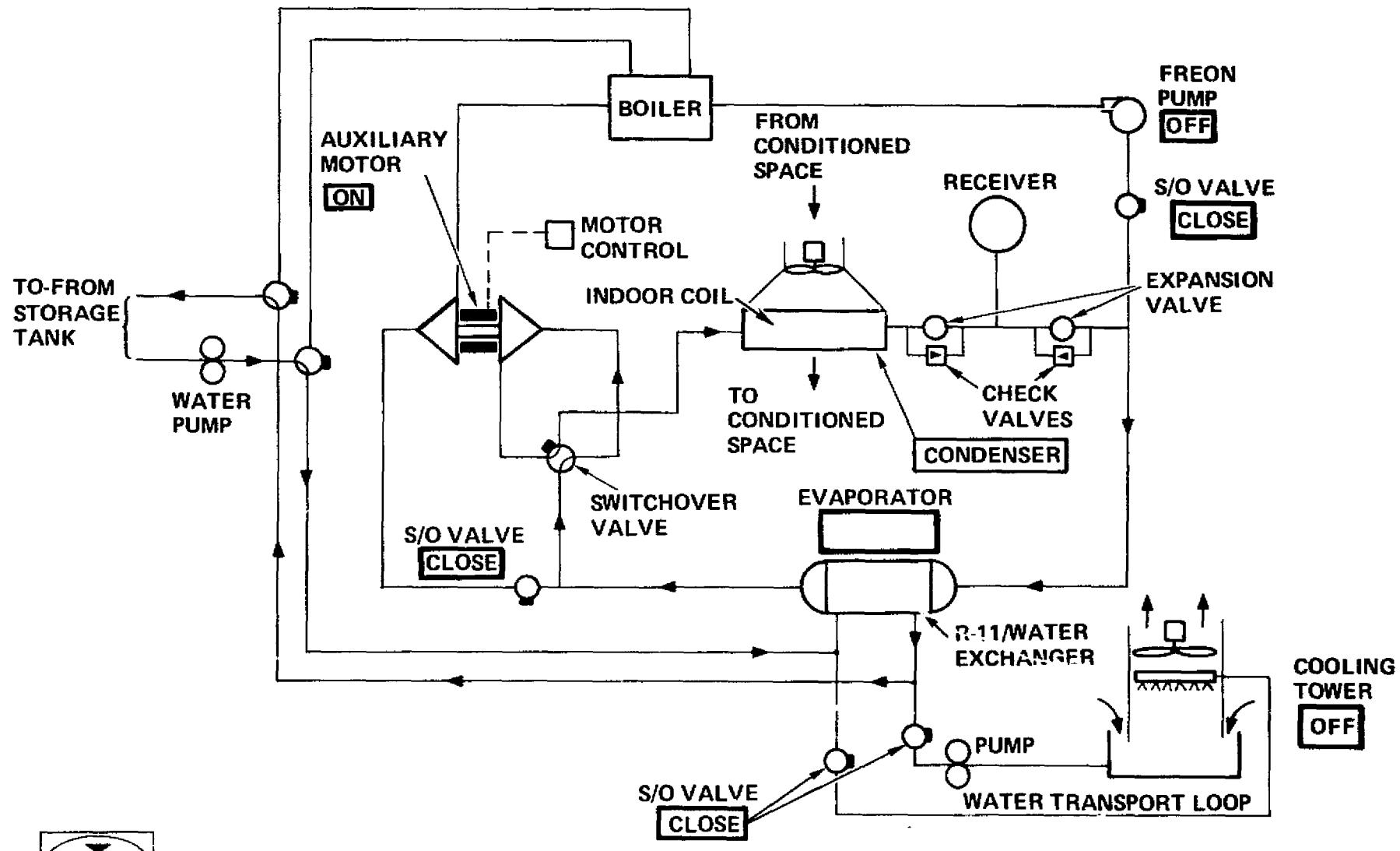
The modified system is shown in the heating mode of operation. The schematic was prepared for the case where a cooling tower is used as the ultimate heat sink in the cooling mode. With an evaporative condenser an R-11-to-water heat exchanger would have to be added to the system in parallel with the evaporative condenser. This heat exchanger would not be used in the cooling mode and would require isolation.

The system modifications necessary for dual mode operation include:

- (a) Addition of selector valves in the water lines from the hot water storage tank. These valves control the flow of water either to the boiler (cooling mode) or to the R-11/water heat exchanger (heating mode).
- (b) Addition of isolation shutoff valves in the Rankine power loop.
- (c) Addition of shutoff valves to isolate the cooling tower in the heating mode.
- (d) Addition of a switchover valve to assure reversal of the refrigerant flow in the compressor circuit.
- (e) Addition of dual expansion valve-check valve in the refrigerant line between the two-loop heat exchanger. These valves are necessary to switch the condenser-evaporator functions.
- (f) Addition of a receiver for fluid inventory control.

Other modifications involve resizing equipment already included in the baseline system, namely the auxiliary motor, the indoor coil size, and the conditioned space recirculation fan.

# RANKINE SYSTEM AS A HEAT PUMP



## PROGRAM RESULTS

### DESICCANT SYSTEM INVESTIGATIONS

A process was evaluated that uses low-temperature solar heat directly to provide air conditioning. In this process, a regenerable desiccant is used to remove water vapor from an airstream; this dry airstream is cooled with ambient air, and the stream is then humidified adiabatically to provide conditioned air. The desiccant is regenerated by heating from a solar heat source to provide a continuous process.

The preliminary analysis conducted was directed toward determining the dynamic heat and mass transfer performance characteristics of a desiccant dryer using ARI standard values for room return and ambient air and a solar collector heat source temperature of 180°F. Preliminary estimates then were made of the performance characteristics and size of the overall system. The evaluation showed that the adsorption process offers significant potential for air conditioning using solar heat.

# **PROGRAM RESULTS DESICCANT SYSTEM INVESTIGATIONS**

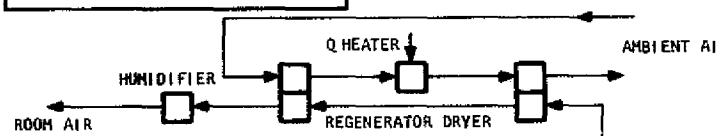


## CANDIDATE CONCEPTS

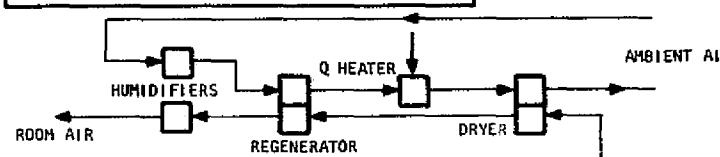
All approaches shown use an air recirculation scheme. The adsorption process can be designed with a 100 percent fresh air supply without recirculation. However, the higher temperature and humidity content of ambient air will result in reduced performance and larger dryer sizes. The performance of the dryer under the conditions imposed by the system arrangements defined as Concepts 1 through 7 was determined by computer analysis. Concepts 3, 4, and 7 offer the greatest potential in terms of bed size and system COP. These three approaches yield the lowest specific humidity for adsorption and desorption.

# CANDIDATE CONCEPTS

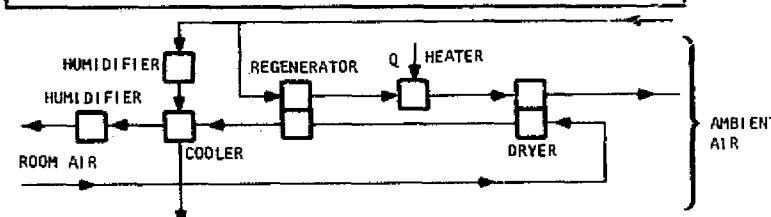
CONCEPT 1--RETURN AIR HUMIDIFIER



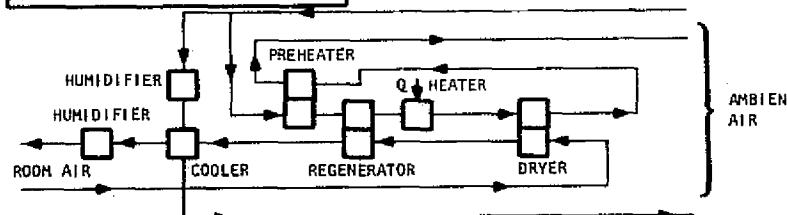
CONCEPT 2--AMBIENT AND RETURN AIR HUMIDIFIERS



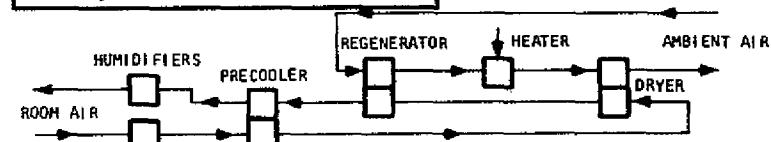
CONCEPT 3--AMBIENT AND RETURN AIR HUMIDIFIERS WITH SEPARATE COOLER



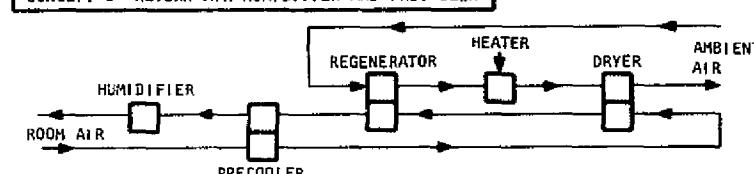
CONCEPT 4--ADDITION OF PREHEATER



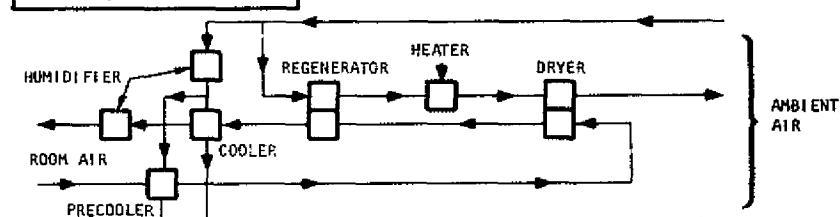
CONCEPT 5--EXIT AND RETURN AIR HUMIDIFIER



CONCEPT 6--RETURN AIR HUMIDIFIER AND PRECOOLER



CONCEPT 7--DRYER PRECOOLER



S-96348



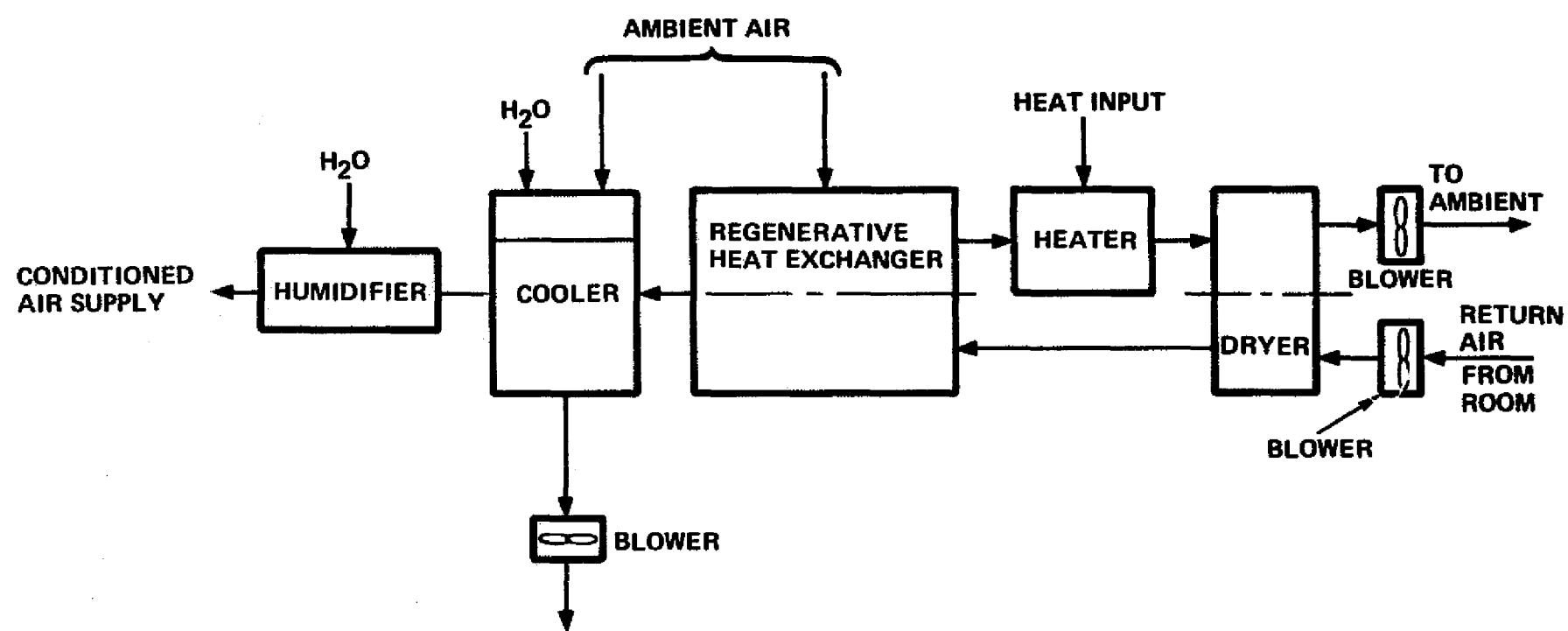
## BASELINE SYSTEM ARRANGEMENT

A schematic of a desiccant air conditioning system is shown. Warm humid air at 80°F dry bulb/67°F wet bulb temperature from the room is directed to the adsorption side of a rotary type regenerable dryer. The dryer contains a desiccant, in this case silica gel. Water is adsorbed from the airstream, and the air leaving the dryer is hot (138°F) and dry (about 0.004 lb water/lb air). The air then passes through a regenerative heat exchanger, where it is cooled to about 100°F, and then to another heat exchanger (cooler) where the temperature is further reduced to 80°F. Finally, the air is humidified adiabatically to a dry bulb temperature of 58°F and is returned to the room.

Ambient air is used to regenerate the dryer, as well as to cool the returning dry air. Part of the ambient air, 95°F db/75°F wb, is humidified and cooled to below 80°F; it then passes through one side of a heat exchanger (cooler) before returning to the ambient environment. This cooled air serves as the final heat sink for the return air. Another portion of the ambient air flows through the regenerative heat exchanger, where it is heated by the hot, dry air from the dryer. The regenerator reduces both the heating load for the heater and the cooling load for the cooler. The ambient air is further heated to 180°F utilizing a solar heat source, and then flows through the rotary dryer. The desiccant is warmed and the adsorbed water is removed by the hot airstream. This regenerates the sorbent bed in the rotary dryer and prepares it for adsorption.

With this process, relatively low-temperature heat is used directly to power the system. Further, the solar collector and thermal storage system can use either air or water. Electrical power is required only for circulating air and for rotating the rotary sorbent bed and regenerative heat exchanger.

# BASELINE SYSTEM ARRANGEMENT



## SORBENT SELECTION

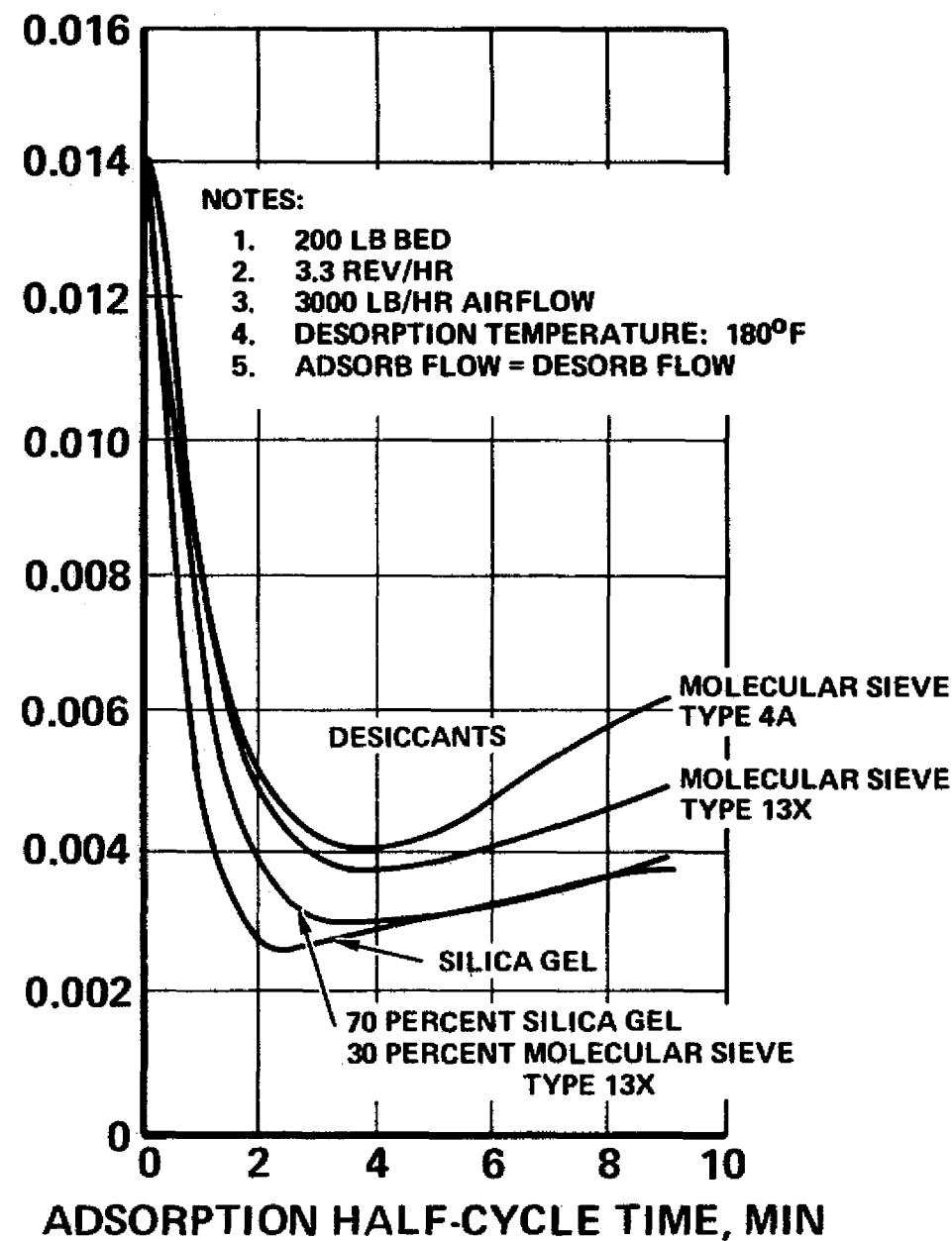
Silica gel was selected as the baseline sorbent because preliminary calculations indicated that its load capability between adsorption and desorption is greater than molecular sieves and the resultant outlet humidity would be lower at the desorption temperature of 180°F considered.

The selection of silica gel as the desiccant material was substantiated by simulating dryer operation using molecular sieve types 13X and 4A. In addition, a computer simulation was run for a composite bed where 30 percent of the desiccant was molecular sieve type 13X and the remainder was silica gel. The 13X was located at the dryer outlet, because under certain conditions 13X has the capability to provide lower outlet specific humidity than silica gel. As shown below, silica gel provides the lowest average outlet humidity, which verifies the initial selection of this sorbent.

Rotary desiccant beds are sometimes made by impregnating the sorbent particle in paper. This results in a significant increase in sorbate flow path to the sorbent as compared to a granular sorbent bed. For comparative purposes, a computer simulation was made of silica gel impregnated paper; and the dryer outlet specific humidity was found to be slightly higher. Thus, desiccant particles rather than desiccant paper were selected as the preferred material for the dryer. Sorbent bed cost considerations also favor the use of the granular material. The cost of silica gel paper obtained from a manufacturer is \$1.60 for a sheet 8-1/2 in. by 3 ft, corresponding to a specific cost of about \$8/lb. For a 200-lb sorbent bed, the material cost excluding fabrication would be \$1600.

# SORBENT SELECTION

SPECIFIC HUMIDITY AT DRYER OUTLET,  
LB WATER VAPOR/LB AIR

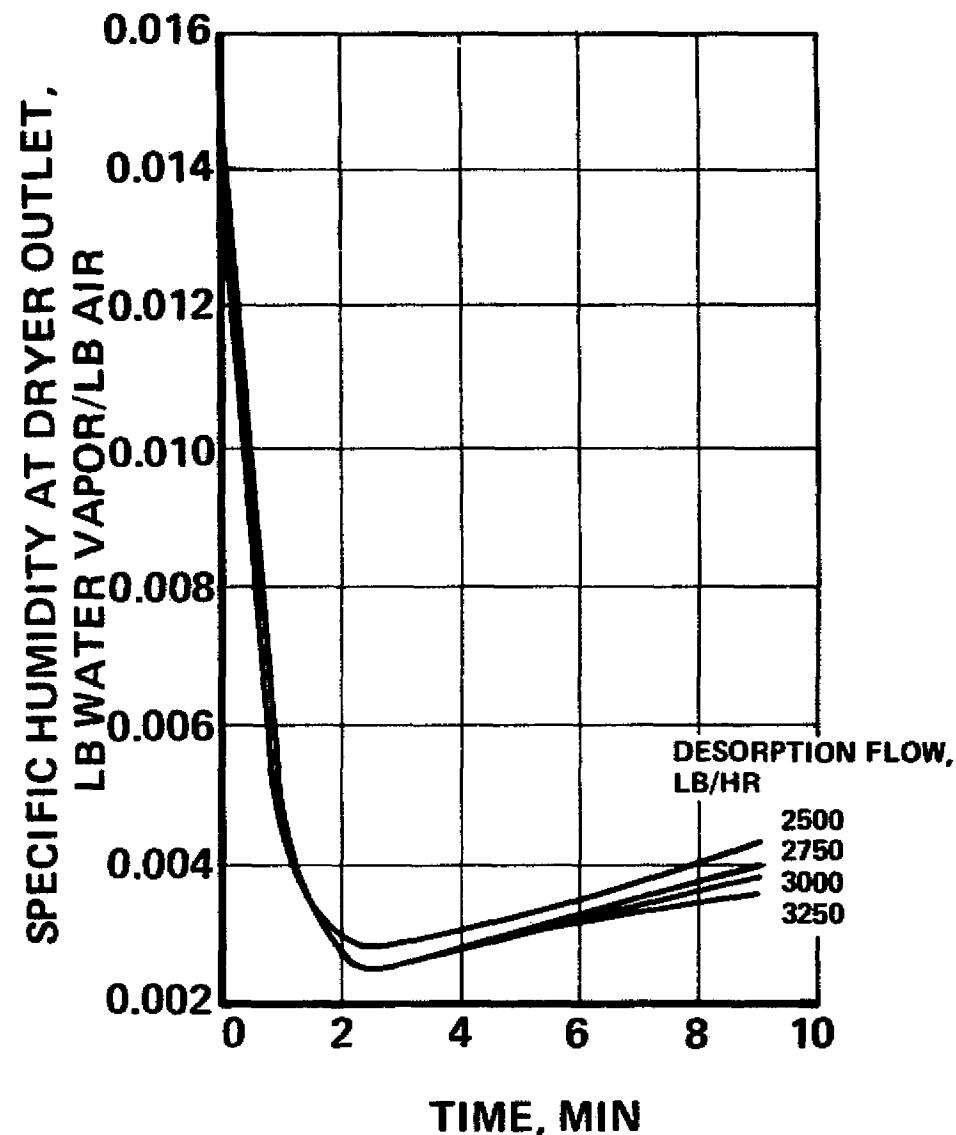
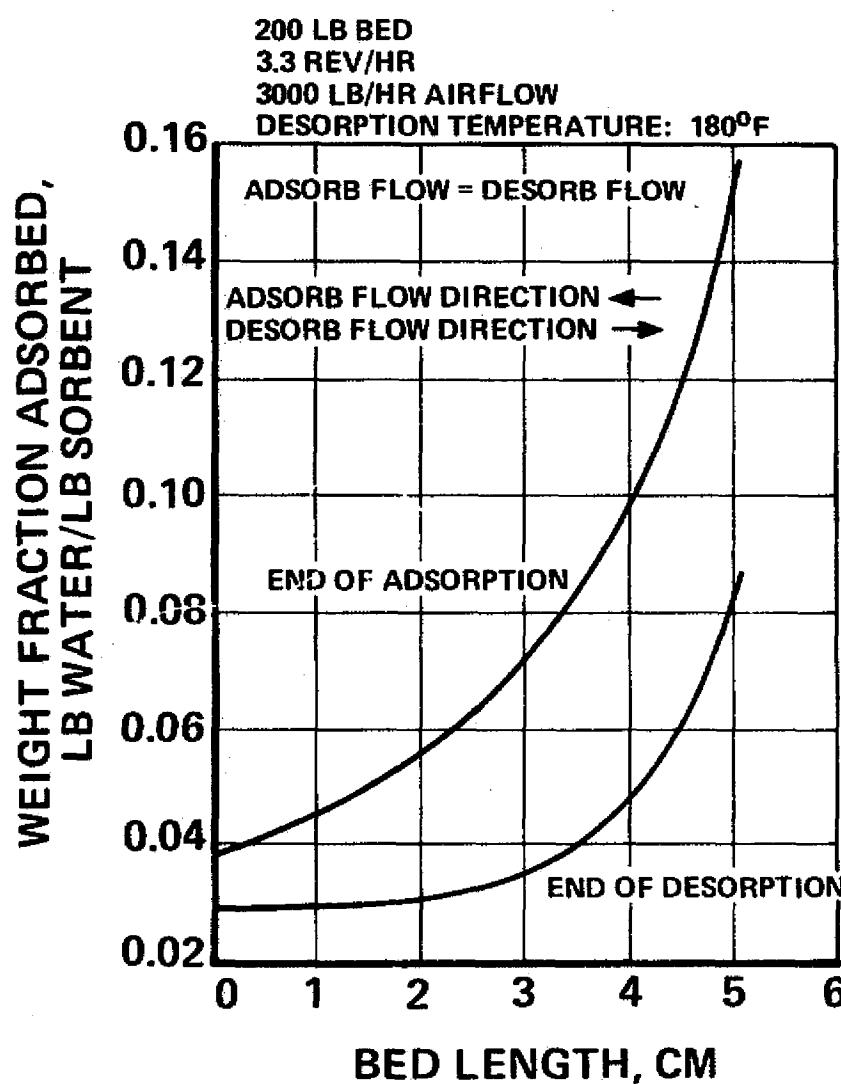


## SORBENT BED CHARACTERISTICS

The weight fraction of water adsorbed on the silica gel desiccant as a function of bed depth is shown below for the conditions noted. The quantity of adsorbed water remaining on the sorbent at the end of desorption is nearly constant for a significant distance into the bed. This indicates that lower desorption airflows could be used rather than equal adsorption and desorption airflow rates.

A computer simulation was run utilizing three desorption airflow rates different from the baseline value of 3000 lb/hr. The results also are shown. Desorption airflows 8.3 percent greater or smaller than the baseline have little effect on the average dryer outlet humidity. A desorption airflow of 2500 lb/hr (16.7 percent less than the baseline) increased the average outlet specific humidity by 7.4 percent. The cooling capacity of the return air was decreased by 3.9 percent; however, the heater load may be decreased by as much as 16.1 percent at this lower airflow. Thus, a net gain in COP is obtained by using a desorption-to-adsorption ratio lower than one. This represents a significant parameter in the optimization of a system for maximum COP.

# SORBENT BED CHARACTERISTICS



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## BASELINE SYSTEM PACKAGE

### - 3-TON UNIT -

Physically, the dryer is a thin cylindrical annulus, which was found to yield acceptable package dimensions. One-half of the dryer was assumed to be in the adsorb mode and the other half in the desorb mode as the cylinder slowly rotates. Seals separate the two sections of the sorbent bed. The pressure difference across the seals will be small, and leakage can be neglected. Utilizing a rotary device rather than a fixed bed results in a constant mixed-air condition during adsorption and desorption. (Use of a fixed sorbent bed would cause temperature and humidity swings in the conditioned space due to the variable outlet temperature and specific humidity.) Conceptually, a rotary bed translates the time-variable output from a fixed bed to a variable output as a function of degree of rotation. However, the average exit conditions from the rotary bed are constant for given operating parameters.

The system regenerator is also configured as a thin cylindrical annulus concentric with the sorbent bed. The package below shows the arrangement of the major system components (excluding the blowers) about the axis of rotation of the dryer-regenerator.

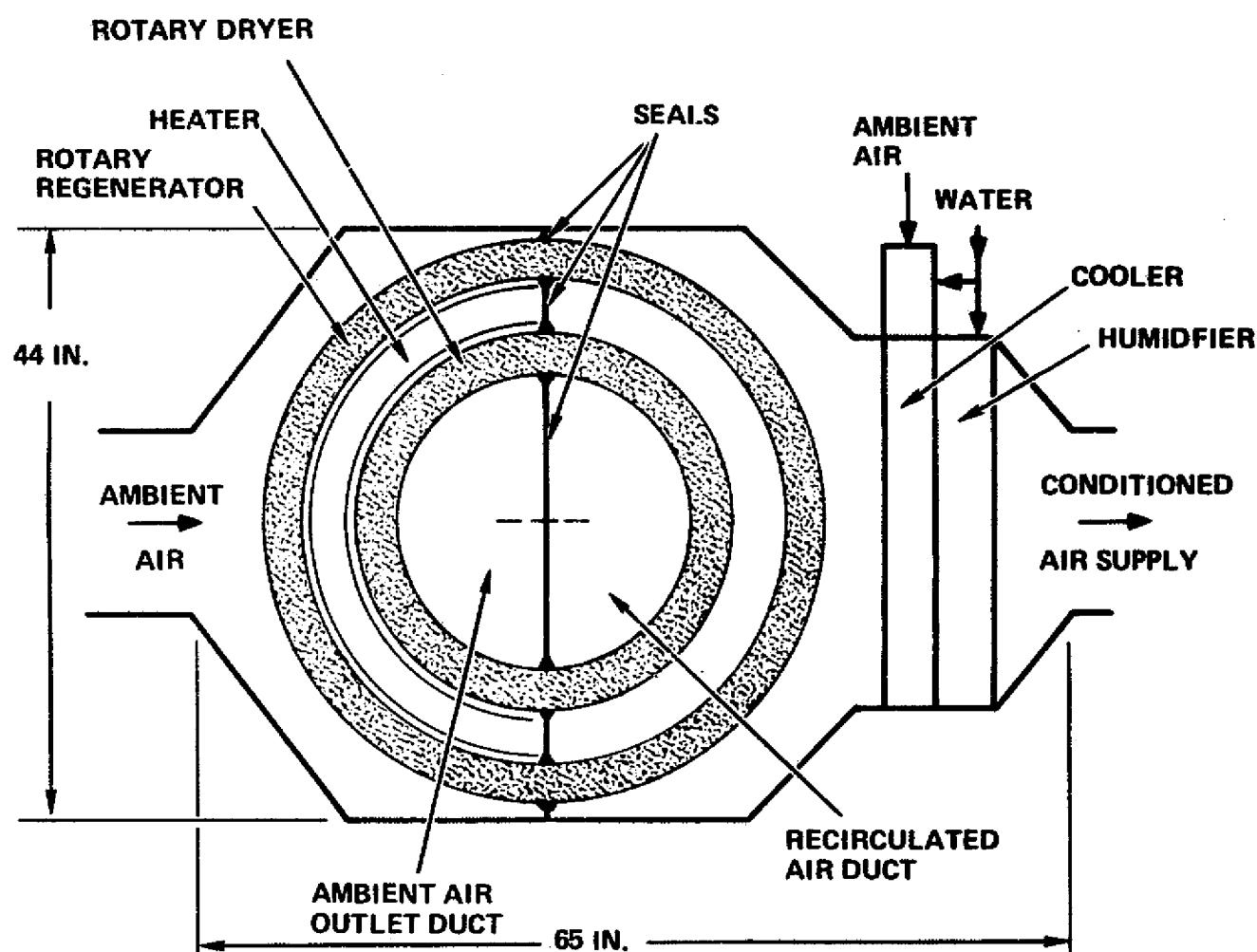
Ambient air enters the air conditioner on the left-hand side and flows through the regenerator, heater, and dryer in series. This air is collected in half the cylindrical volume contained within the dryer and is exhausted from the top of the package. The ambient air blower could be mounted on top of the package or on the left side of the package by rearranging the inlet air manifold.

The room return air enters the unit at the top, flows into the recirculated air duct and through the dryer, regenerator, cooler, and humidifier in series. This arrangement minimizes ducting and turning losses through the unit. The recirculated air fan could be mounted in the ducting interfacing with the air conditioner or in a manner similar to the ambient air blower.

The bearings and drive for the rotating assembly (not shown) will be at the bottom and top of the unit. The overall package dimensions as shown are 44 by 48 by 65 in.

# BASELINE SYSTEM PACKAGE

## — 3-TON UNIT —



#### BASELINE SYSTEM CHARACTERISTICS

The overall characteristics of the system designed to provide 3 tons of refrigeration under standard ARI conditions are summarized below. It is estimated that COP's as high as 0.8 can be achieved through optimization of the system parameters. Off-design data generated indicate that operation at temperatures well below 160°F is feasible; COP's comparable to design point values are anticipated at lower desorption temperatures. Capacity reduction is estimated at 0.67 percent/°F.

## BASELINE SYSTEM CHARACTERISTICS

SYSTEM CAPACITY	3 TONS
AMBIENT TEMPERATURE	95°F DB, 75°F WB
ROOM RETURN AIR TEMPERATURE	80°F DB, 67°F WB
DESORPTION TEMPERATURE	180°F
RECIRCULATED AIRFLOW	1110 CFM
AMBIENT (DESORPTION) AIRFLOW	1000 CFM
AMBIENT (COOLING) AIRFLOW	620 CFM
SORBENT BED (SILICA GEL) WEIGHT	200 LB
ROTATING ASSEMBLY SPEED	3.7 REVOLUTIONS/HR
CONDITIONED AIR SUPPLY	58°F DB, 56°F WB
COP (CAPACITY/SOLAR HEAT INPUT)	0.717
ELECTRICAL POWER USAGE	1.15 KW
OVERALL PACKAGE SIZE	44 BY 48 BY 65 IN.



# CONCLUSIONS

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## CONCLUSIONS--DESICCANT SYSTEM

The analyses conducted have demonstrated the feasibility of using the adsorption process for the development of a solar-powered air conditioner. Compared with competing approaches (absorption and Rankine systems), the process offers several advantages. The data presented in this report show the following:

- (a) An adsorption air conditioner can be designed to operate at a COP higher than 0.7 with a desorption temperature of 180°F. It is anticipated that a COP of 0.8 can be achieved by optimization of the system parameters, primarily the desorption/adsorption flow ratio.
- (b) A 3-ton unit using granular silica gel as the sorbent and operating at a desorption temperature of 180°F has an overall package size of 44 by 48 by 65 in. and electrical power usage of 1.15 kw.
- (c) The system will operate at desorption temperatures lower than 160°F at reduced capacity. Capacity drop as a function of temperature is estimated at 0.67 percent/°F.
- (d) No major process or hardware problems are anticipated in the development of a complete system. The only equipment item that may require design support testing is the evaporative cooler.

By comparison to the Rankine-cycle machine, the desiccant system offers potential COP advantages and hardware simplicity. One of the major drawbacks is the requirement for augmentation by thermal energy; another is the capability of the Rankine-cycle system for operation as a heat pump.

By comparison to the LiBr/H<sub>2</sub>O absorption system the desiccant air conditioner offers the advantages of (1) hardware simplicity, (2) flexibility of operation over a wide range of heat source temperatures, and (3) potentially higher thermal COP. Like the absorption system, it is strictly an air conditioner and cannot be used for heat pumping. System application would be in climatic areas where air conditioning establishes the size of the solar collector/thermal storage unit.

In view of the inherent advantages offered by the adsorption process, it is recommended that the investigations initiated under Task 1 of Contract NAS8-30758 be continued. The first step in these studies should be aimed at the optimization of the design in terms of system performance.

## CONCLUSIONS DESICCANT SYSTEM

- GRANULAR SILICA GEL APPEARS TO BE THE OPTIMUM SORBENT FOR DESORPTION TEMPERATURE COMPATIBLE WITH FLAT PLATE SOLAR COLLECTOR
- THERMAL COP'S HIGHER THAN 0.7 CAN BE ACHIEVED AT A DESORPTION TEMPERATURE OF 180°F
- SYSTEM WILL OPERATE AT REDUCED CAPACITY AT DESORPTION TEMPERATURES AS LOW AS 160°F
- SIMPLE SYSTEM IN TERMS OF CONTROLS AND HARDWARE
- CAN BE DESIGNED TO INTERFACE DIRECTLY WITH AIR TYPE COLLECTOR/THERMAL STORAGE UNIT
- STRICTLY AN AIR CONDITIONER; CANNOT BE USED FOR PUMPING HEAT



S-470

## CONCLUSIONS--RANKINE-CYCLE AIR CONDITIONER

Investigations under this contract were centered on the definition of a Rankine-cycle solar-powered residential air conditioner. The energy source is a water-type flat plate collector with water temperatures below 200°F. The major technical objective was to develop a system that could operate without auxiliary power with relatively high COP at heat source temperatures below 200°F. This objective was achieved.

Early studies revealed that operation of a heat machine using a heat source temperature below 200°F would require minimum heat sink temperatures. Comparative evaluation of various condenser concepts demonstrated that an evaporative-type heat sink (water evaporation) was essential. Either an evaporative condenser or a cooling tower is an acceptable solution.

The system developed uses a turbocompressor for operation in the normal mode. In the augmented mode, when solar energy is not sufficient to power the system, electrical energy is used in a high-speed electrical motor designed as an integral part of the turbocompressor. This provides (1) flexibility of operation at high efficiency over a wide range of heat source temperatures, and (2) the means for minimum use of auxiliary power in the augmented mode. The salient features of the recommended concept are listed below.

The capacity of the system investigated was 3 tons, representative of a residential air conditioner. The turbocompressor designed for that size machine features relatively small compressor and turbine; the design point efficiency of these components is estimated at 0.69 and 0.80 respectively. For larger size units (25 tons), turbocompressor efficiency will be significantly higher; it is estimated that for larger tonnage systems, COP's as high as 0.75 to 1.0 could be achieved with current turbomachine technology.

A significant advantage of the Rankine-cycle system, in comparison with the absorption and desiccant systems, is its potential for utilization as a heat pump. In such operation, the compressor is driven by the motor, and the wintertime low-temperature solar energy is lifted to a temperature level suitable for heating. It is estimated that COP's between 6 and 14 can be achieved depending on the heating load and heat source temperature.

The flexibility of operation and the overall performance of the Rankine-cycle system, featuring a turbocompressor with integral motor, make it an extremely attractive system for use with solar thermal energy. It is recommended that such a system be developed and demonstrated to ascertain its full potential. The technology is available for the development of such a system; and preliminary cost estimates prepared under this contract indicate economic feasibility in view of escalating energy costs.

# CONCLUSIONS

## RANKINE-CYCLE AIR CONDITIONER

- REQUIRES EVAPORATIVE HEAT SINK FOR OPERATION
- THERMAL COP IN NORMAL OPERATION BETWEEN 0.6 AND 0.8 (3-TON SYSTEM)
- THERMAL COP COMPARABLE TO COMPETING CONCEPTS (ABSORPTION SYSTEM, DESICCANT SYSTEM)
- THERMAL COP OF LARGER CAPACITY SYSTEMS (25-TON AND UP) ESTIMATED BETWEEN 0.75 AND 1.0
- NORMAL OPERATING RANGE EXTENDS TO HEAT SOURCE WATER TEMPERATURES AS LOW AS 160°F DEPENDING ON OPERATING CONDITIONS
- TURBOMACHINE WITH INTEGRAL MOTOR PROVIDES FOR EXTENSION OF THERMAL ENERGY UTILIZATION TO TEMPERATURE LEVELS AS LOW AS 145°F
- OPTIMUM SYSTEM IN TERMS OF AUXILIARY POWER IN AUGMENTED MODE
- RANKINE AIR CONDITIONER CAN BE USED AS HEAT PUMP WITH MINOR MODIFICATIONS
- COP IN HEAT PUMP MODE ESTIMATED AT 6 TO 14
- TECHNOLOGY AVAILABLE FOR DEVELOPMENT AT COMPETITIVE COST

