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DEVELOPMENT OF A SOLAR-POWERED
RESIDENTIAL AIR CONDITIONER

CONTRACT NAS8-30758
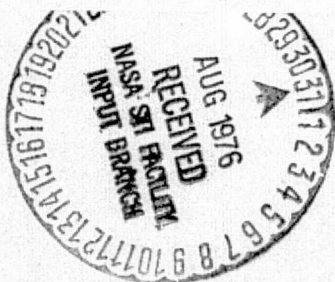
PROGRAM REVIEW

APRIL 8, 1975

74-10996(5)

GARRETT

PRESENTATION OUTLINE

PROGRAM DEFINITION

- OBJECTIVES

- APPROACH

- STATUS

PROGRAM RESULTS

- STATE-OF-THE-ART REVIEW

- REQUIREMENTS ANALYSIS

- ECONOMIC ANALYSIS

- COMPONENT DATA

- SYSTEM DATA

CONCLUSIONS AND FUTURE WORK

PROGRAM OBJECTIVES

Three processes are currently considered to provide residential air conditioning using low temperature thermal energy from a flat plate solar collector: (1) Rankine-cycle-powered mechanical refrigeration system; (2) absorption process; and (3) adsorption process. The efforts under this contract are primarily concerned with the Rankine-powered air conditioner. Only a minor portion of the total effort is devoted to the absorption and adsorption processes.

The temperature level of the energy source available from the solar collector is about $200^{o}F$; that of the heat sink necessary for operation of the heat machine is about $95^{o}F$. Under these conditions, the maximum efficiency attainable as expressed by Carnot cycle efficiency is about 15 percent. Actually, a Carnot cycle efficiency on the order of 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.

Thus, it is apparent that the actual efficiency of the Rankine machine will be low, and large quantities of source heat will be necessary for operation. This in turn translates into large solar collector and thermal energy storage devices.

For this reason, in order to maximize the utility of the system it will be necessary to maximize the effectiveness of all system components and configure the system to minimize cost of ownership. This constitutes the overall goal of the study program. Specific objectives are listed on the opposite page.

The optimization of the air conditioning system in terms of cost and performance will require 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, it is of primary importance here that the Rankine air conditioning system cost data be generated in a parametric fashion, so that these data can be used in the future for overall system optimization over a range of solar collector cost and performance.

Under the present contract, a system will be selected using realistic solar collector performance and cost data and electrical and thermal energy costs representative of the 1980 projected costs.

PROGRAM OBJECTIVES

⊚ OPTIMIZATION OF A RANKINE POWERED AIR CONDITIONER DESIGNED
SPECIFICALLY FOR OPERATION USING THERMAL ENERGY FROM A FLAT
PLATE SOLAR COLLECTOR

⊚ CHARACTERIZATION OF THE OPTIMUM SYSTEM IN TERMS OF COST AND
PERFORMANCE SO AS TO PERMIT OVERALL SYSTEM TRADE STUDIES

⊚ DEVELOPMENT OF A PRELIMINARY SPECIFICATION FOR A FIELD
EVALUATION DEMONSTRATOR

S-94775

PROGRAM LOGIC

The approach used in achieving the program objectives is shown in the accompanying diagram. The six program tasks are identified and the relationship between these tasks is illustrated.

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$_2$O) absorption system was characterized in terms of performance and cost; these data were compiled to establish a baseline for comparison. In the performance of Task 1, data on an adsorption system using silica gel sorbent indicated that such a system may offer advantages. Evaluation of this approach has been included as part of Task 1 to determine feasibility. Data on systems and equipment applicable to the Rankine air conditioner under consideration were collected to establish a data bank for performance characterization of the system component.

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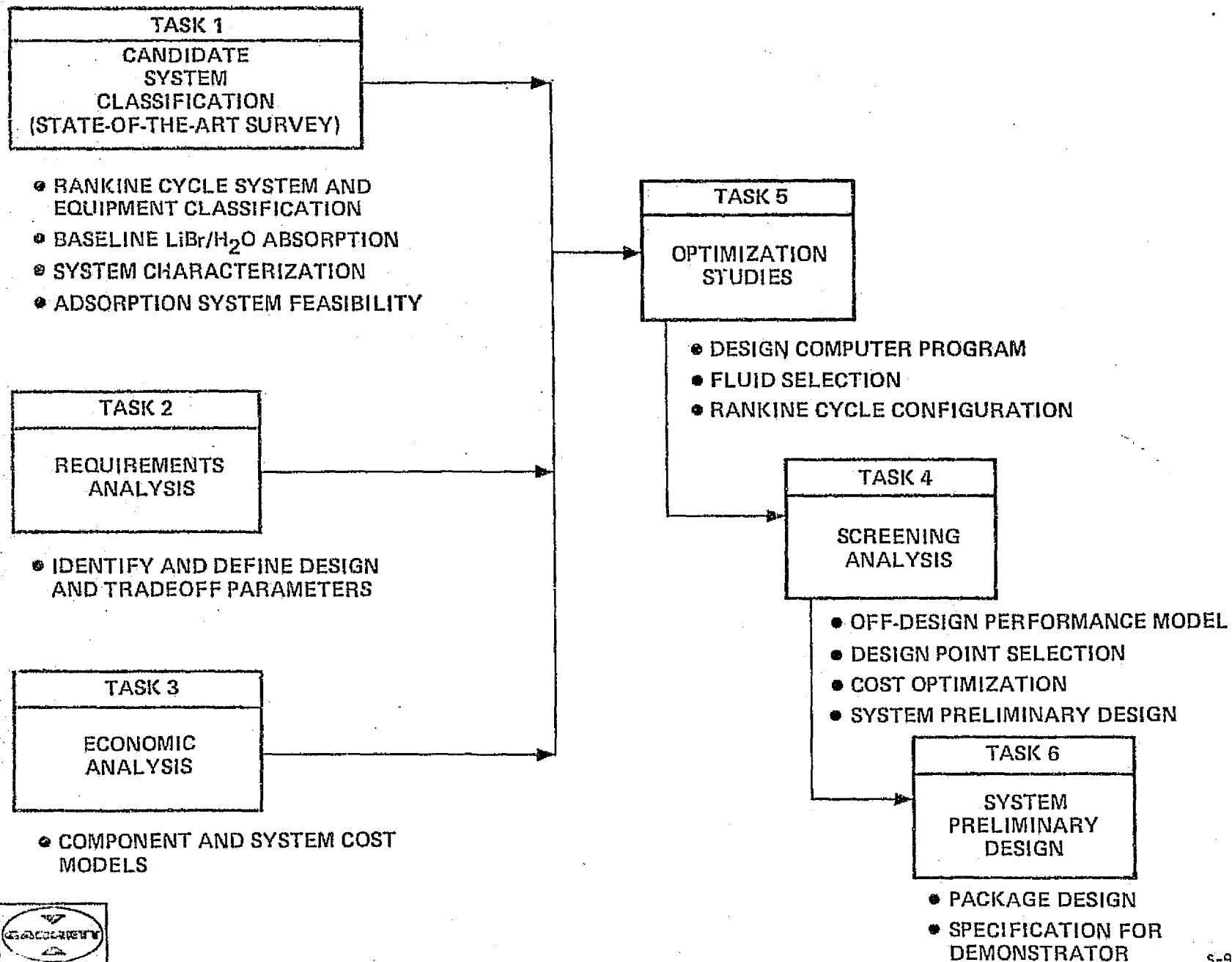
Under Task 3, economic analysis, system and equipment cost data were collected and correlated. Models were developed to permit characterization of candidate air conditioning systems in terms of cost.

The results of Tasks 1 through 3 are currently used in a screening analysis. Task 4 is 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 has been developed to mechanize system design calculations. Fluid selection studies have been completed. The computer program initially developed as a cycle analysis program is being modified into a system design program by incorporating the 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 will require off-design performance evaluation. This will be done under Task 5. A computer program describing the off-design performance of the system will be developed. This program will be used together with a solar collector-thermal storage system to determine overall system performance and cost. In this manner, system design point will be selected corresponding to minimum overall cost or minimum usage of auxiliary power. Note that these two selection criteria may correspond to different system design points.

System design point performance data will be used to generate component problem statements from which equipment preliminary designs will be developed. Preliminary package layouts and a preliminary component and system specification will be prepared. This will be done under Task 6 of the program.

PROGRAM LOGIC

TASK 1

CANDIDATE
SYSTEM
CLASSIFICATION
(STATE-OF-THE-ART SURVEY)

- RANKINE CYCLE SYSTEM AND EQUIPMENT CLASSIFICATION
- BASELINE LiBr/H_2O ABSORPTION
- SYSTEM CHARACTERIZATION
- ADSORPTION SYSTEM FEASIBILITY

TASK 2

REQUIREMENTS
ANALYSIS

- IDENTIFY AND DEFINE DESIGN AND TRADEOFF PARAMETERS

TASK 3

ECONOMIC
ANALYSIS

- COMPONENT AND SYSTEM COST MODELS

TASK 5

OPTIMIZATION
STUDIES

- DESIGN COMPUTER PROGRAM
- FLUID SELECTION
- RANKINE CYCLE CONFIGURATION

TASK 4

SCREENING
ANALYSIS

- OFF-DESIGN PERFORMANCE MODEL
- DESIGN POINT SELECTION
- COST OPTIMIZATION
- SYSTEM PRELIMINARY DESIGN

TASK 6

SYSTEM
PRELIMINARY
DESIGN

- PACKAGE DESIGN
- SPECIFICATION FOR DEMONSTRATOR

S-94779

PROGRAM SCHEDULE AND STATUS

The performance period for the contract is 11 months. Currently Tasks 1 and 4 are active. Task 1 was extended to cover the duration of the program. This was done primarily to permit coverage of current study and hardware programs directly related to the development of solar-powered air conditioners. Investigations of the adsorption process also have been initiated under this task. An interim document summarizing the findings of the literature survey for the first three months of the contract was published in January as AiResearch Report 74-10996(3). Component performance models evolved under Task 1 are currently being incorporated in the system design computer program developed under Task 4.

The results of Task 2, requirements analysis, were compiled and published in November. These data, contained in AiResearch Report 74-10996(2), will be updated as necessary throughout the program.

The economic analysis of Task 3 also has been completed. The cost models developed for parametric characterization of the Rankine system and components are documented in AiResearch Report 74-10996(4).

Currently the major effort is concentrated on the screening analysis of Task 4. A cycle computer program has been developed and is being converted into a system design program by incorporation of component models. The cycle computer program has been used in the evaluation of candidate working fluids over a range of cycle parameters. It is estimated that the system program is 90 percent complete.

Off-design performance maps for the Rankine expansion turbine and the refrigeration compressor are being generated as the basis for the development of the off-design optimization program under Task 5.

Data generated under Tasks 1 through 4 are presented in the remainder of this report.

PROGRAM SCHEDULE AND STATUS

| TASK DESCRIPTION	MONTHS										
	NOV	DEC	JAN	FEB	MAR	APR	MAY	JUNE	JULY	AUG	SEPT
	1	2	3	4	5	6	7	8	9	10	11
*TASK 1. CANDIDATE SYSTEM CLASSIFICATION											△
TASK 2. REQUIREMENTS ANALYSIS											
TASK 3. ECONOMIC ANALYSIS											
*TASK 4. SCREENING ANALYSIS											
*TASK 5. OPTIMIZATION STUDIES											
TASK 6. SELECTED SYSTEM PRELIMINARY DESIGNS											
DOCUMENTATION											
MONTHLY PROGRESS REPORTS	▲										△
PROGRAM REVIEWS					▲				△		△
FINAL REPORT											△

* ACTIVE TASKS

S-94778
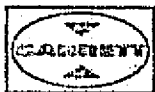
PROGRAM FUNDING

Actual and projected expenditures of program funds are shown below. Program manpower loading was planned at a level of effort corresponding to one man for the first 3-1/2 months of the contract. As shown previously, activities during this period were concerned with the state-of-art survey, establishment of design requirements, and economic analysis. With the screening analysis task, average program loading corresponds to a 2-man level of effort as originally planned. The higher rate of expenditure reflects primarily the activities of aerodynamicists for compressor and turbine mapping, and system analysts for system computer program development.

Referring to the previous figure (program schedule and status), it is apparent that the program is slightly underexpended relative to the work accomplished at this time. For this reason, it has been possible to include the investigation of the adsorption process within the present contract funds.

PROGRAM FUNDING

PROGRAM PERIOD

S-94850
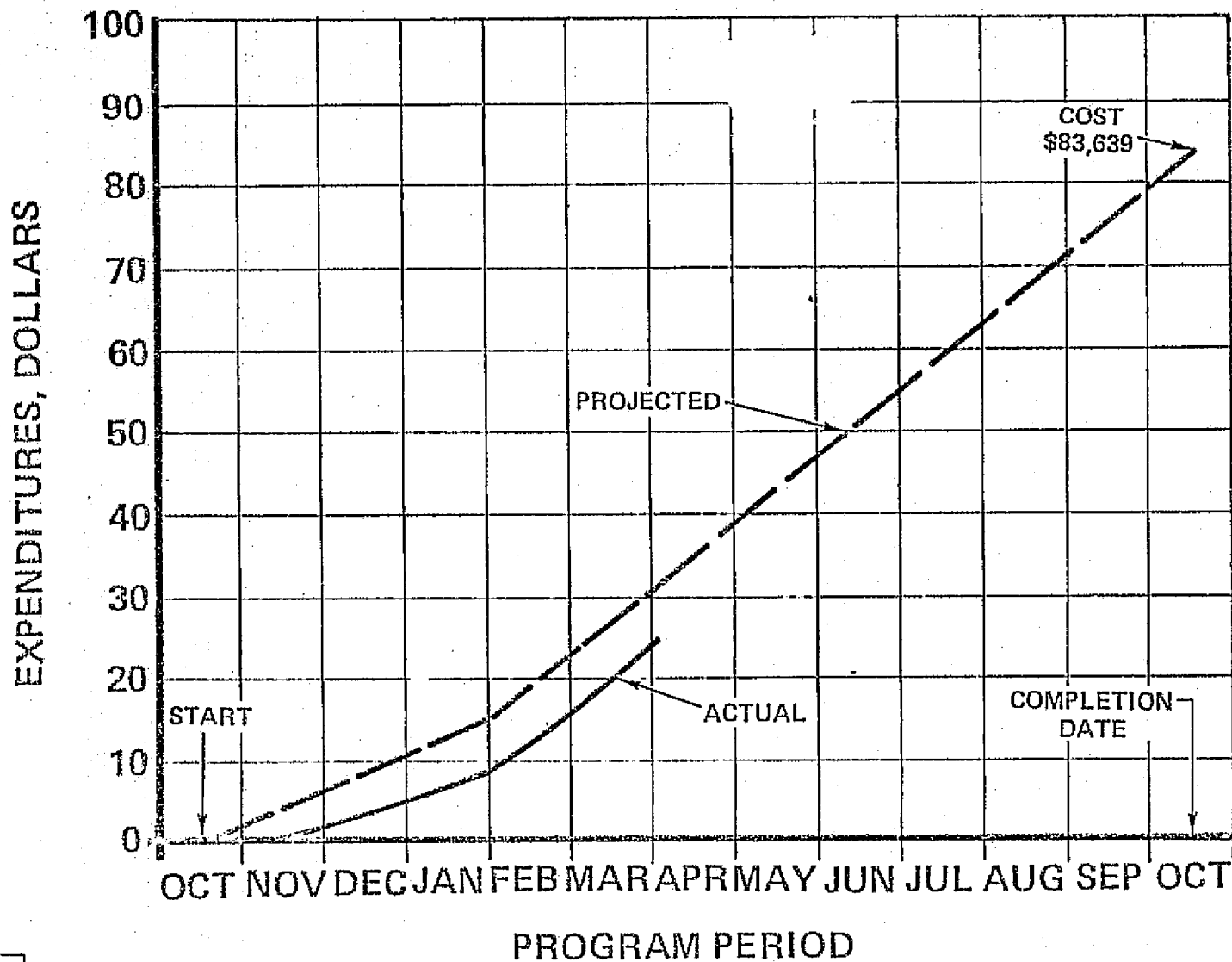
STATE-OF-ART REVIEW

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

(1) The LiBr/H_2O absorption system

(2) The Rankine-powered air conditioner

(3) Air conditioning by the adsorption process
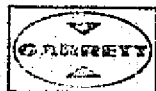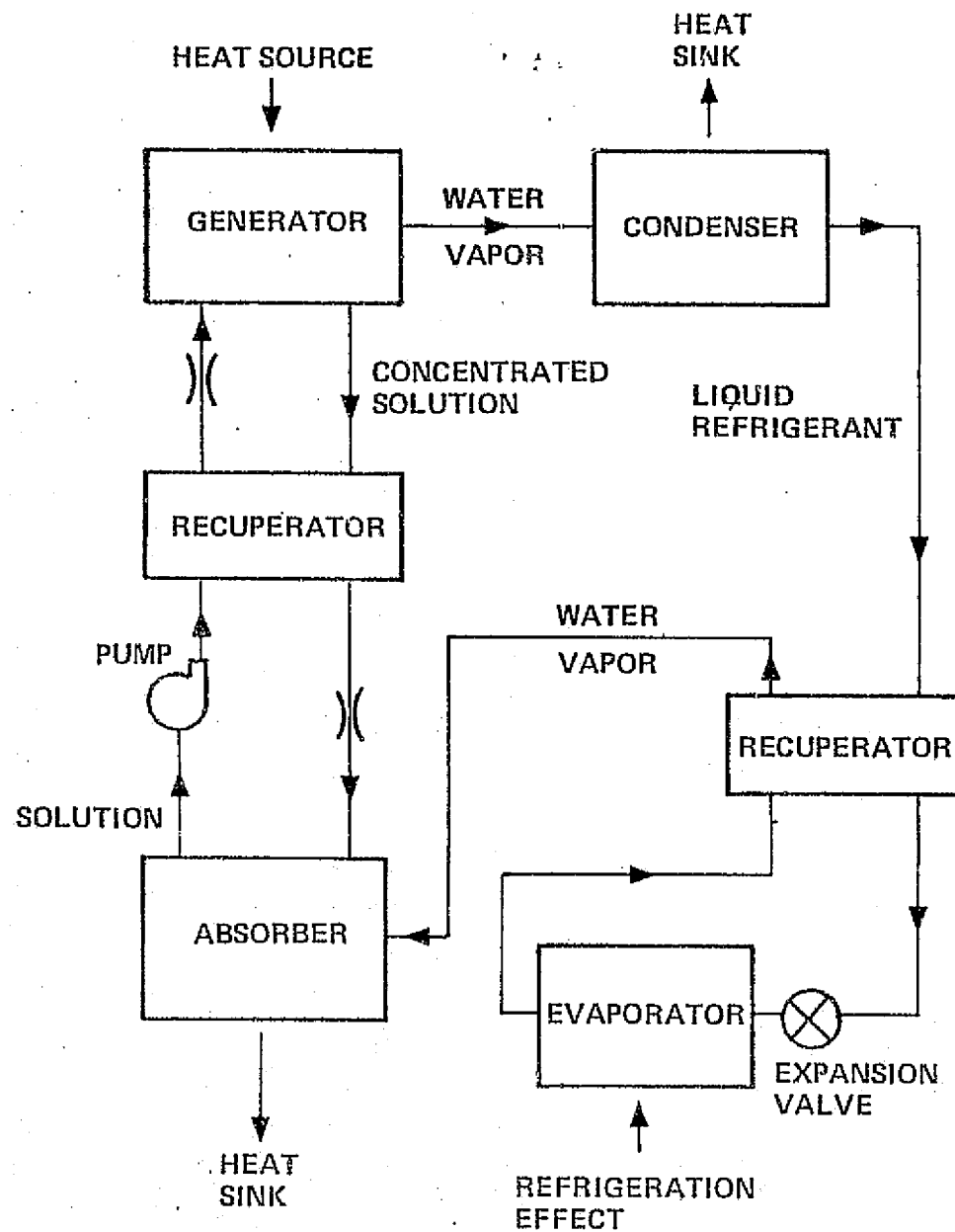
STATE-OF-ART REVIEW (TASK 1)

5-94771

LiBr/H$_2$O ABSORPTION CYCLE

A simple schematic of a LiBr/H$_2$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 is in the NASA solar house.

Currently Arkla is under contract to NSF to further modify the basic LiBr/H$_2$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$_2$O ABSORPTION CYCLE

HEAT SOURCE

HEAT SINK

GENERATOR

WATER VAPOR

CONDENSER

CONCENTRATED SOLUTION

LIQUID REFRIGERANT

RECUPERATOR

WATER VAPOR

PUMP

RECUPERATOR

SOLUTION

ABSORBER

EVAPORATOR

EXPANSION VALVE

HEAT SINK

REFRIGERATION EFFECT

GARRETT

S-94783

BASELINE LiBr/H$_2$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 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. These data are consistent with data obtained on the NASA solar house unit.

The cost of the LiBr/H$_2$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. A list price of $1250 for a 3-ton unit is considerably higher than the cost of a conventional vapor compression air conditioner.

BASELINE LiBr/H$_2$O SYSTEM CHARACTERISTICS

COOLING CAPACITY

10.54 KW (3 TONS)

HOT WATER SOURCE TEMPERATURE

363.7 K IN/358.2 OUT (195°F IN/185°F OUT)

CHILLED WATER TEMPERATURE

285.9 K IN/280.4 OUT (55°F IN/45°F OUT)

WATER CONSUMPTION

25.2 μM^3/SEC (24 GAL/HR)

EVAPORATIVE HEAT REJECTION

298.7 K (78°F WB AIR IN)

COEFFICIENT OF PERFORMANCE

0.65

ELECTRICAL CONSUMPTION

875 WATTS MAXIMUM

SYSTEM COST

$1250 (LIST PRICE)
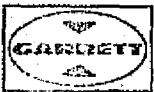
# LiBr/H$_2$O SYSTEM OFF-DESIGN PERFORMANCE

Off-design performance of a typical LiBr/H$_2$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 TRW[*]. 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 180°F, for example, at constant cooling water temperature, system capacity will drop to about 50 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 so as to control the minimum generator water temperature at 185°F; cooling tower water temperature was maintained between 70 and 80°F over the operating range.

[*]"Solar Heating and Cooling of Buildings, Phase 0 Final Report," NSF/RA/N-74-022C (Appendix Volume), TRW Systems Group, May 1974.

LiBr/H₂O SYSTEM OFF DESIGN PERFORMANCE

5-94782
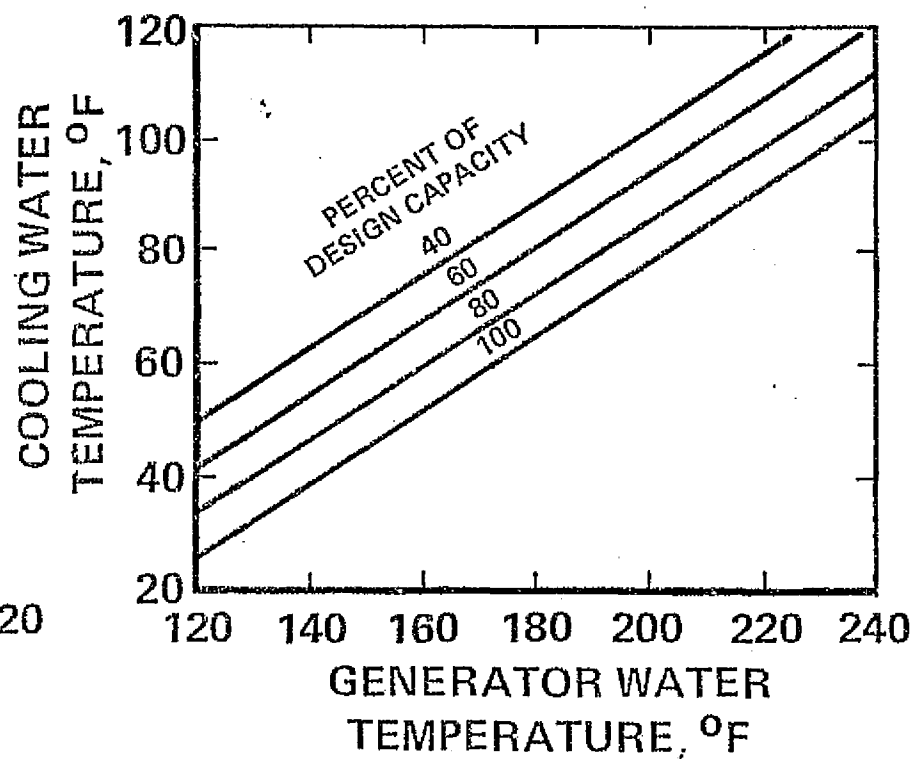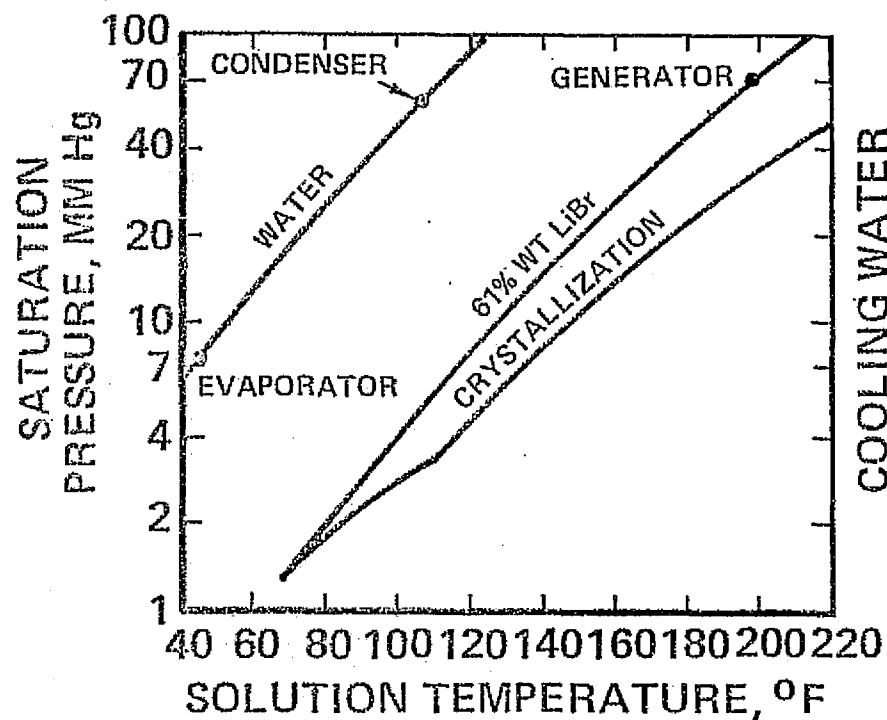
A number of organizations are currently involved in analytical and experimental investigations directly related to the application of the Rankine cycle to solar-powered air conditioners. The most significant work found in the literature is summarized below.

Barber Nichols has designed and developed a 3-ton air conditioner for evaluation in the Honeywell trans-portable solar laboratory. The Rankine power loop 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 Rankine fluid is R-113. 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. System operation with a 170°F water temperature was demonstrated; under this condition, system capacity dropped to 1 ton and COP decreased to 0.25.

General Electric Company has been engaged in the development of a multivane expander to drive a conven-tional refrigerant compressor. Efficiencies as high as 75 percent have been achieved with R-11 as the working fluid. Test units have been subjected to 1000 hours of development testing.

As early as 1970, AiResearch delivered to the U.S. Army two heat-powered 5-ton refrigeration units (water chiller and air conditioner). These systems utilized the thermal energy contained in the exhaust stream of a gas turbine to drive a Rankine turbine. Turbine power is expended in driving the centrifugal compressor of a refrigeration loop. 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. The system working fluid is R-11. Preliminary analysis indicates that this unit could be used to provide 4.5 tons of air conditioning under 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 per-cent have been achieved on test with inlet and outlet pressures of 56 and 185 psia.

United Aircraft Research Laboratories is engaged in a turbocompressor feasibility demonstration program. The test unit is a modified turbocompressor originally designed to produce 8 tons of cooling with R-114 as the working fluid and gas turbine exhaust as the heat source. Peak efficiencies for the compressor and turbine op-erating separately have been shown to be 0.69 and 0.78 respectively. Matching problems in the solar powered application will result in somewhat lower efficiencies (0.62 and 0.75 for the compressor and turbine respectively).

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 claimed as 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 72 PERCENT OBTAINED ON TEST WITH R-11

● AIRESEARCH — 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

ADSORPTION AIR CONDITIONER

A simple schematic of an adsorption 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 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 the Rankine cycles.

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

*Lunde, Peter, "Solar Desiccant System Analysis and Materials," Workshop Proceedings, Solar Cooling for Buildings, Los Angeles, California, February 1974.

ADSORPTION AIR CONDITIONER
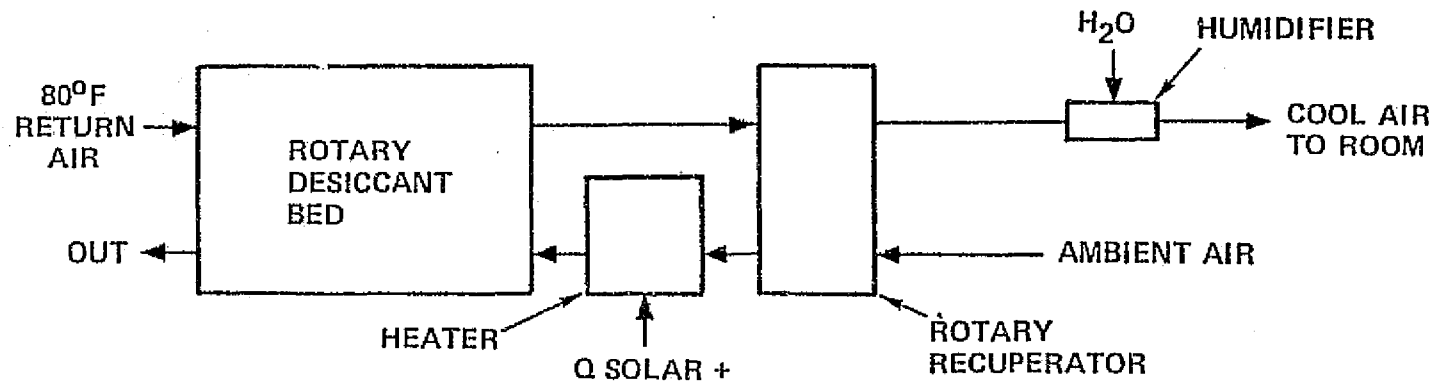
ADVANTAGES

1. AIR IS WORKING FLUID

2. RELATIVELY LOW TEMPERATURE

3. USES HEAT DIRECTLY

4. POTENTIAL FOR AIR COLLECTOR

5. COOLING MODE COP'S OF 0.7 — 0.8 AT MAX. TEMP OF 200°F

S-94020-A

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 will be considered together in the study. The distribution subsystem establishes the air conditioner cooling requirements and does not include any equipment that will affect the design of the air conditioner itself. The distribution fan will be taken as part of the air conditioner package. The solar source subsystem will be considered separately and defined by its performance interfaces and other pertinent characteristics that will permit trade studies and optimization at the overall system level.

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 will be 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 technique used to provide a heat sink:

Air-Cooled Air Conditioners--Where the cycle waste heat is dumped directly to an ambient air stream.
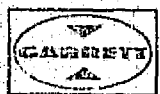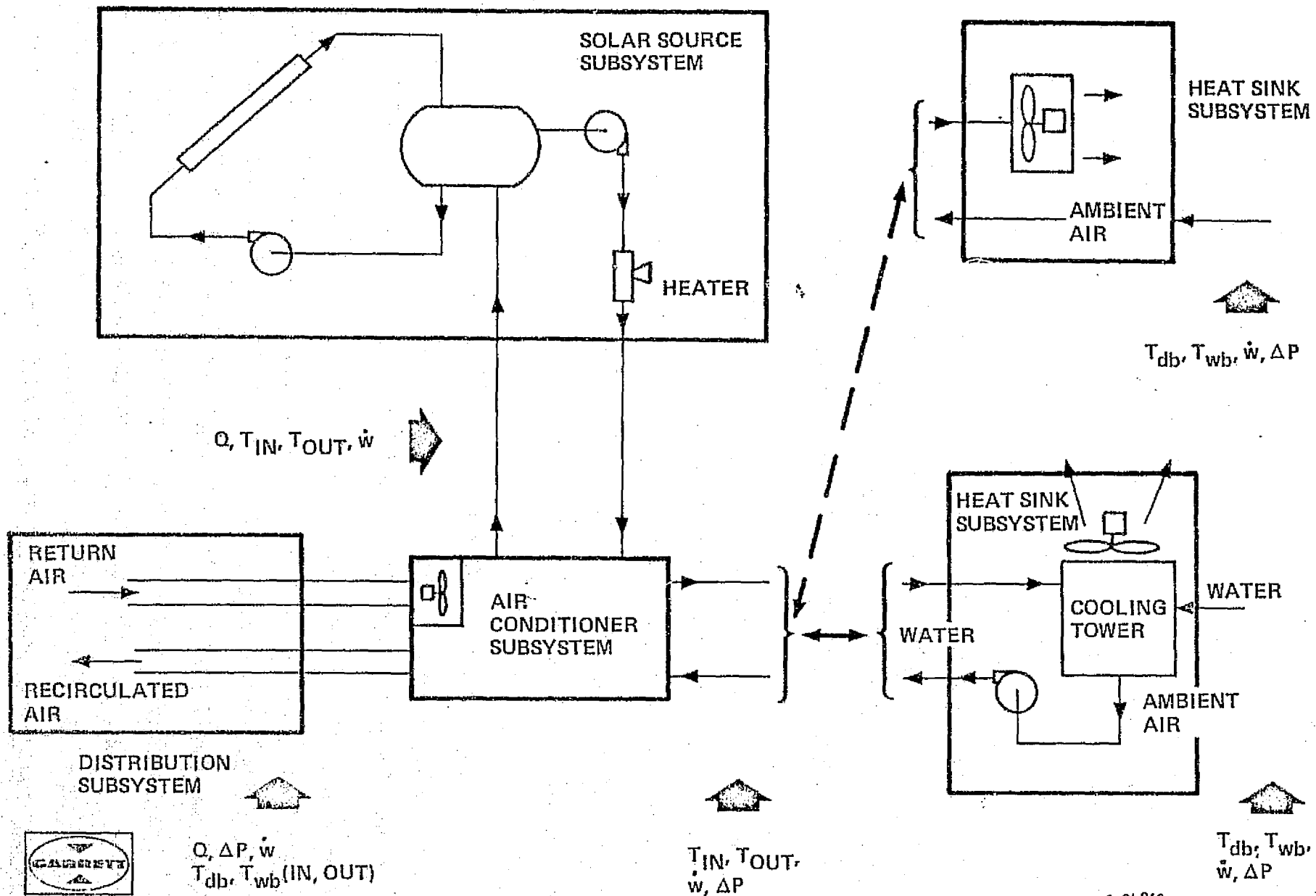
Water-Cooled Air Conditioners--Where the cycle waste heat is dumped into water from a cooling tower.

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. Definition of the interfaces listed on the schematic will provide the data necessary for the design of the air conditioner (including the heat sink).

A number of parameters internal to the air conditioner will be 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, pump, and compressor.

Using a fixed set of interface data (design point conditions), equipment parametric data can be generated in terms of cycle parameters, and air conditioner cost and performance can be determined. By changing the design point over a range of conditions, parametric cost and performance is obtained. This optimization procedure will be used in the study.

SYSTEM INTERFACES

SOLAR SOURCE
SUBSYSTEM

HEATER

HEAT SINK
SUBSYSTEM

AMBIENT
AIR

$T_{db}, T_{wb}, \dot{w}, \Delta P$

$Q, T_{IN}, T_{OUT}, \dot{w}$

RETURN
AIR

AIR
CONDITIONER
SUBSYSTEM

RECIRCULATED
AIR

DISTRIBUTION
SUBSYSTEM

HEAT SINK
SUBSYSTEM

WATER

WATER

COOLING
TOWER

AMBIENT
AIR

$Q, \Delta P, \dot{w}$
T_{db}, T_{wb} (IN, OUT)

$T_{IN}, T_{OUT},$
$\dot{w}, \Delta P$

$T_{db}, T_{wb},$
$\dot{w}, \Delta P$

S-94812

DESIGN AND EVALUATION PARAMETERS

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

Load profile and solar collector/thermal storage unit data will be necessary to determine auxiliary power usage through typical summer conditions.

DESIGN AND EVALUATION P ARAMETERS

COOLING CAPACITY	3 TONS (1½ TO 5 TONS)
AMBIENT AIR	95°F DB, 75°F WB
SOLAR HEAT SOURCE	180°F (170 TO 240°F)
RETURN AIR	400 CFM/TON; 80°F DB, 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	TBD
COLLECTOR-TANK PERFORMANCE	TBD
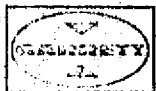
ECONOMIC ANALYSIS

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 systems for residential application.

The rationale used in the development of the cost model 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 will be used for the purpose of system evaluation and tradeoff studies. The final cost of the selected system will be determined more accurately through detailed cost estimates using equipment sizes and problem statements derived from the system and component analyses.

ECONOMIC ANALYSIS

SYSTEM COST MODEL

Air conditioners are arketed through distributors and dealers to the consumer. Manufacturer list prices represent the approximate price that the user will ultimately pay. List prices, however, are used mostly as a base from which the actual prices to the distributor, dealer, and consumer are computed.

Normally, the manufacturer price to the distributor is about 35 percent of the list price. The distributor markup is roughly 100 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 30 percent of the list price. Consequently, the cost to the user will be about the same as 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
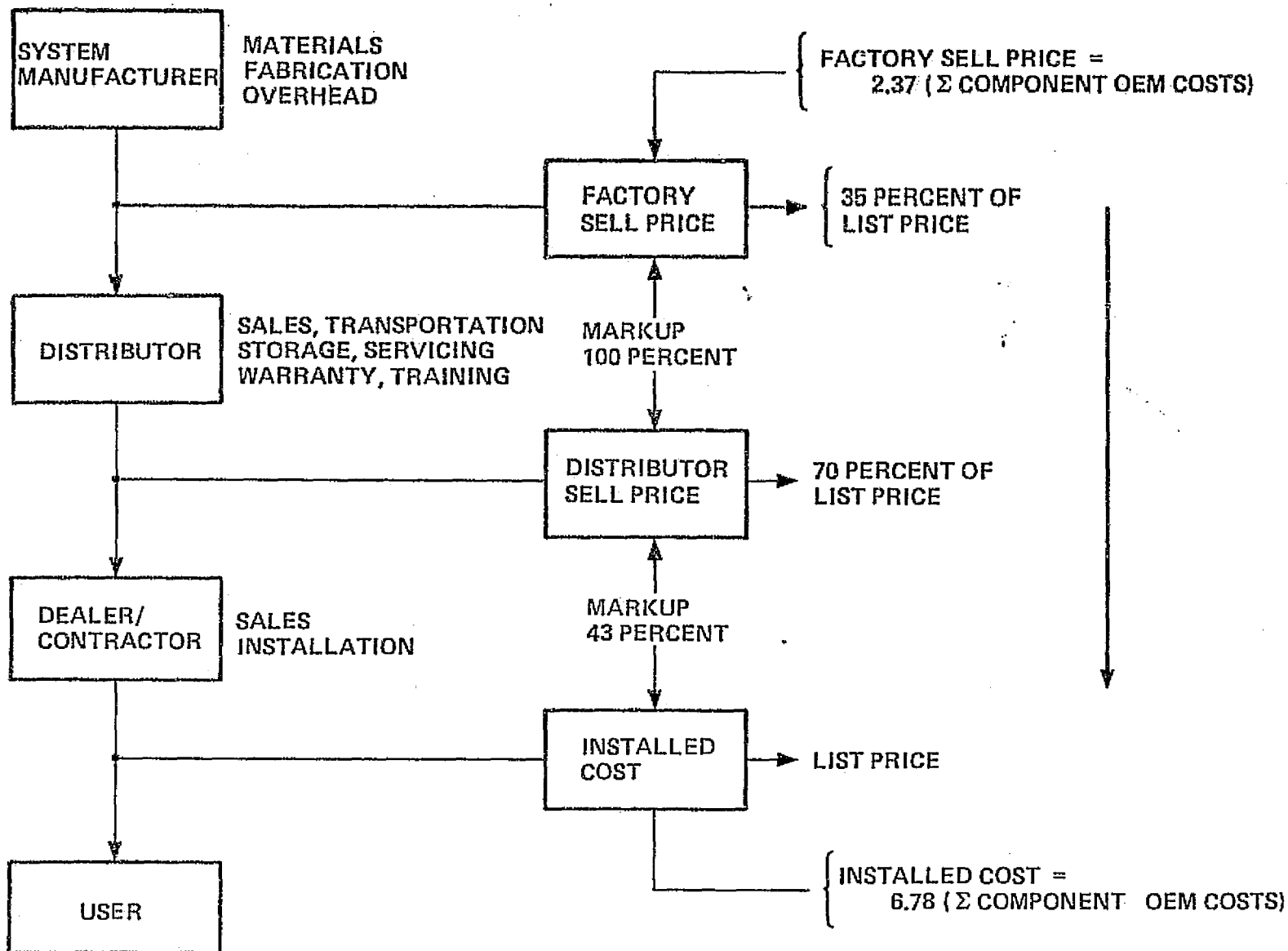
$$\text{System factory direct cost} = \frac{1}{0.7} \left(\Sigma \text{ component OEM costs} \right)$$

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 considered in establishing factory sell price:

(a) Labor constitutes about 11 percent of the total system cost for systems in the 7.0- to 10.5-kw (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

$$\text{Factory sell price} = 2.37 \left(\Sigma \text{ component OEM costs} \right)$$

SYSTEM COST MODEL

SYSTEM MANUFACTURER	MATERIALS FABRICATION OVERHEAD

FACTORY SELL PRICE = 2.37 (Σ COMPONENT OEM COSTS)

FACTORY SELL PRICE → 35 PERCENT OF LIST PRICE

DISTRIBUTOR	SALES, TRANSPORTATION STORAGE, SERVICING WARRANTY, TRAINING

MARKUP 100 PERCENT

DISTRIBUTOR SELL PRICE → 70 PERCENT OF LIST PRICE

DEALER/ CONTRACTOR	SALES INSTALLATION

MARKUP 43 PERCENT

INSTALLED COST → LIST PRICE

INSTALLED COST = 6.78 (Σ COMPONENT OEM COSTS)

USER

S-94781

COMPONENT COST MODELS

System costs can be estimated if the cost of the major system components are 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 the condenser and shell and tube units for 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 $0.89/lb. Shop labor cost, assembly line setup cost, and shop overhead represent approximately 30 percent of the total factory cost. Manufacturer markup, including administrative and operating expenses other than factory, is about 30 percent; using these factors, the manufacturer's sell price can be computed from the heat exchanger core weight. The cost of the shell and tube units that will be used as boilers is estimated in a fashion similar to that of the finned tube units. Factory sell price is about $3.60/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.

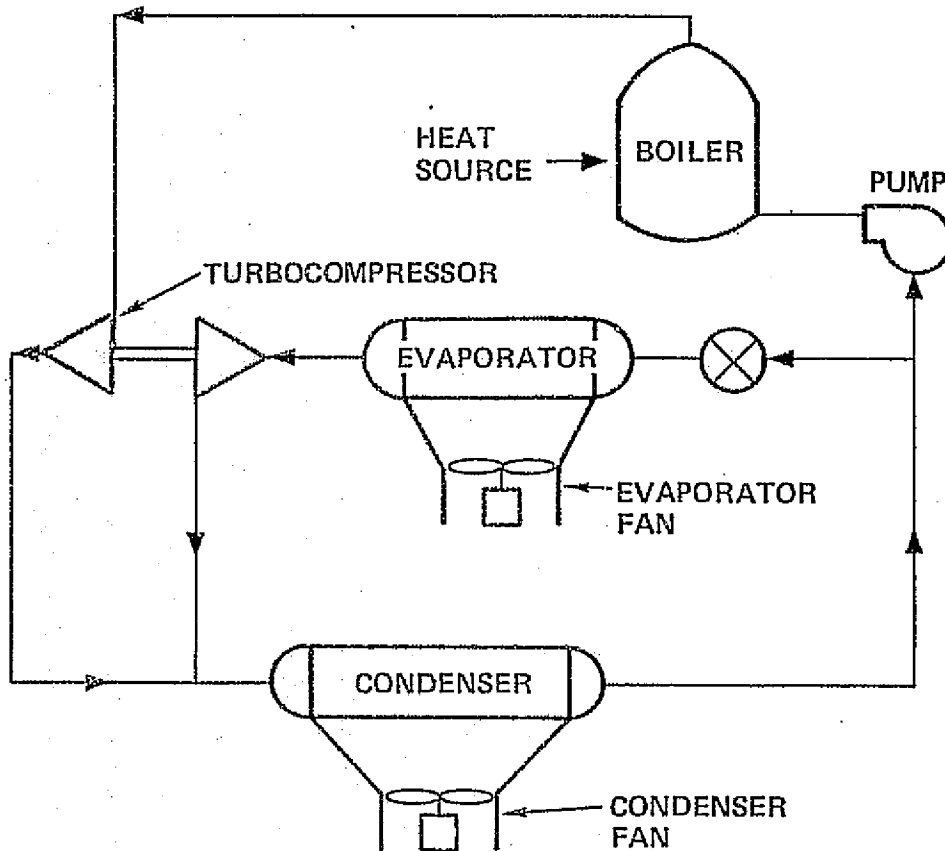Turbocompressors—Cost data on turbocompressors of considerable aerodynamic sophistication were obtained from the AiResearch Industrial Division (AID) of The Garrett Corporation. Here, material cost is a small portion of the total cost of the unit. For the purpose of system comparison, a unit price of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

Fans and Blowers—Manufacturers' catalogues and price lists were used to determine vane axial fan cost. The OEM cost of these fans (steel units) can be approximated at $2.20/lb. This value will be 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
HEAT EXCHANGERS	
● EVAPORATOR	
● CONDENSER	1.8 (HX CORE WEIGHT, LB)
● BOILER	3.6 (HX WEIGHT, LB)
REFRIGERANT PUMP	50
TURBOCOMPRESSOR	100
FANS/BLOWERS	2.20 (FAN WEIGHT, LB)
ELECTRIC MOTORS + DRIVES	25 + 45 (KW)
CONTROLS AND WIRING	10 PERCENT OF TOTAL

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SYSTEM DATA

In evaluating a turbocompressor system, it is necessary to determine the efficiencies of the turbine and compressor at the specified conditions and assure that the speed of these two components will be matched at these conditions. For this purpose, generalized compressor and turbine performance models were used in the development of a cycle analysis computer program. This program also incorporates fluid property data and has been used to mechanize the generation of parametric cycle data. Most of the data presented in the remainder of this presentation were obtained with this program.

Currently, the cycle analysis program is being developed into a system design program where system components are characterized in terms of significant performance parameters. These component data are then used to construct system cost using the system and component cost models presented previously.

The following discussions are concerned with (1) some of the basic component data used in the cycle analysis and system design programs; and (2) cycle analysis results, including fluid selection studies, turbomachinery efficiency, and cycle performance data. Techniques for system augmentation using external energy sources are also discussed.

SYSTEM 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

(b) Specific speed

(c) Tip Mach number

(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

$$\frac{1-\eta}{1-\eta_o} = \left(\frac{10^6}{Re}\right)^{0.1}$$

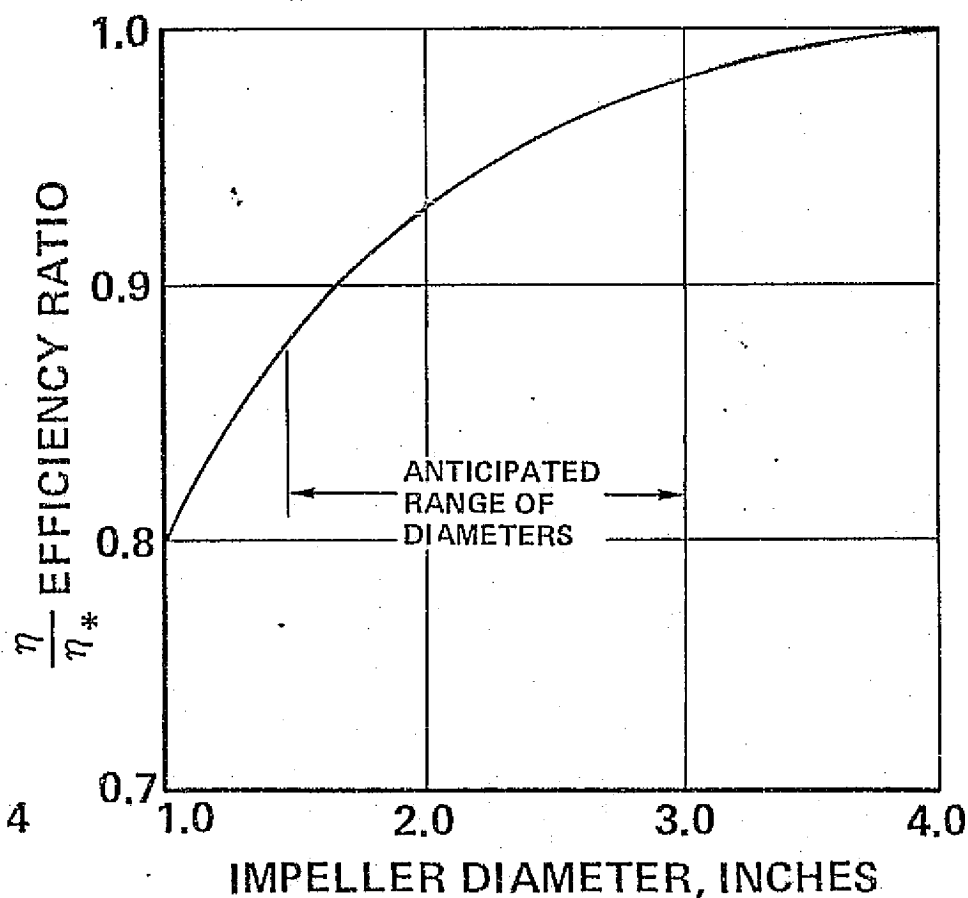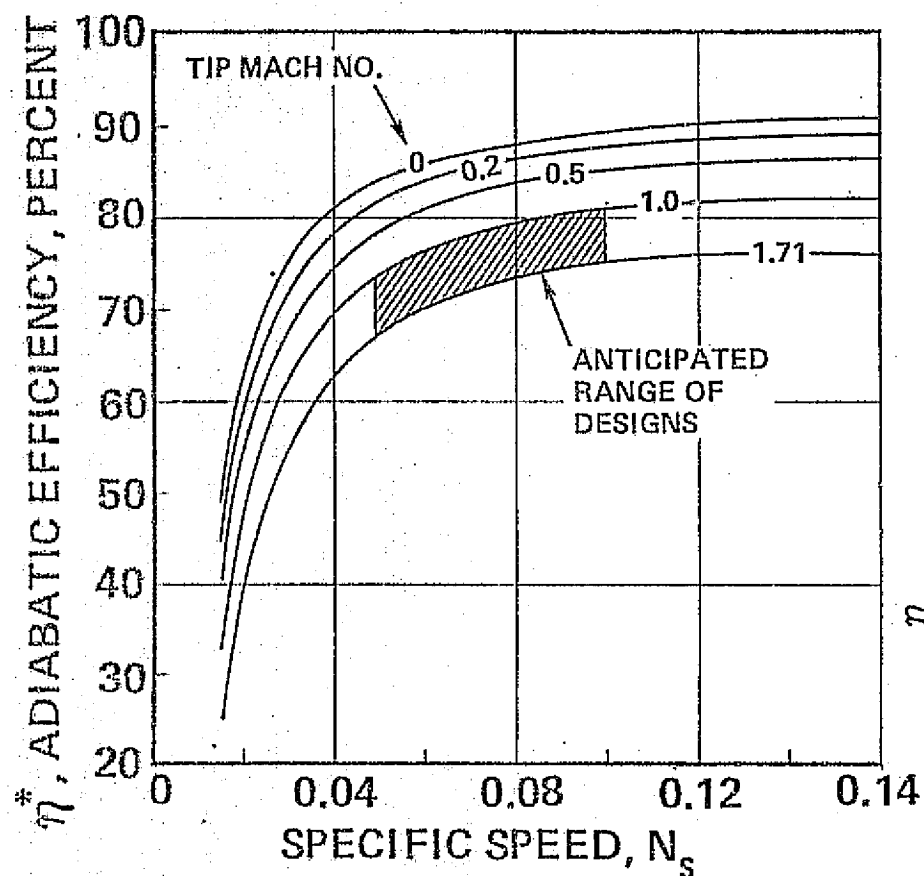Where η is the corrected efficiency and η_o is the efficiency determined for $Re > 10^6$

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.

COMPRESSOR PERFORMANCE

η = CORRECTED EFFICIENCY

η^* = EFFICIENCY FOR IMPELLER
$\geqq$ DIAMETER   4.0 INCHES

Left plot: η, η^*, ADIABATIC EFFICIENCY, PERCENT vs SPECIFIC SPEED, N_s

TIP MACH NO.
0, 0.2, 0.5, 1.0, 1.71

ANTICIPATED RANGE OF DESIGNS

Right plot: $\frac{\eta}{\eta^*}$ EFFICIENCY RATIO vs IMPELLER DIAMETER, INCHES

ANTICIPATED RANGE OF DIAMETERS

TURBINE PERFORMANCE

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 map 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 calculations methods. As shown on the figure, 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 10^6. For lower Reynolds numbers, a correction factor (determined experimentally) must be applied:

$$\frac{1-\eta}{1-\eta_o} = \left(\frac{10^6}{Re}\right)^{0.2}$$

where η is the corrected efficiency and η_o is the efficiency obtained from the efficiency plot.

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.

TURBINE PERFORMANCE

ADIABATIC EFFICIENCY, PERCENT vs SPECIFIC SPEED, N_s

D = 6 IN.
3 IN.
2 IN.
1.5 IN.
1.0 IN.
0.5 IN.

RANGE OF SPECIFIC
SPEEDS AND DIAMETERS

$$\frac{U}{C_O} \approx 0.7$$

$Re > 10^6$

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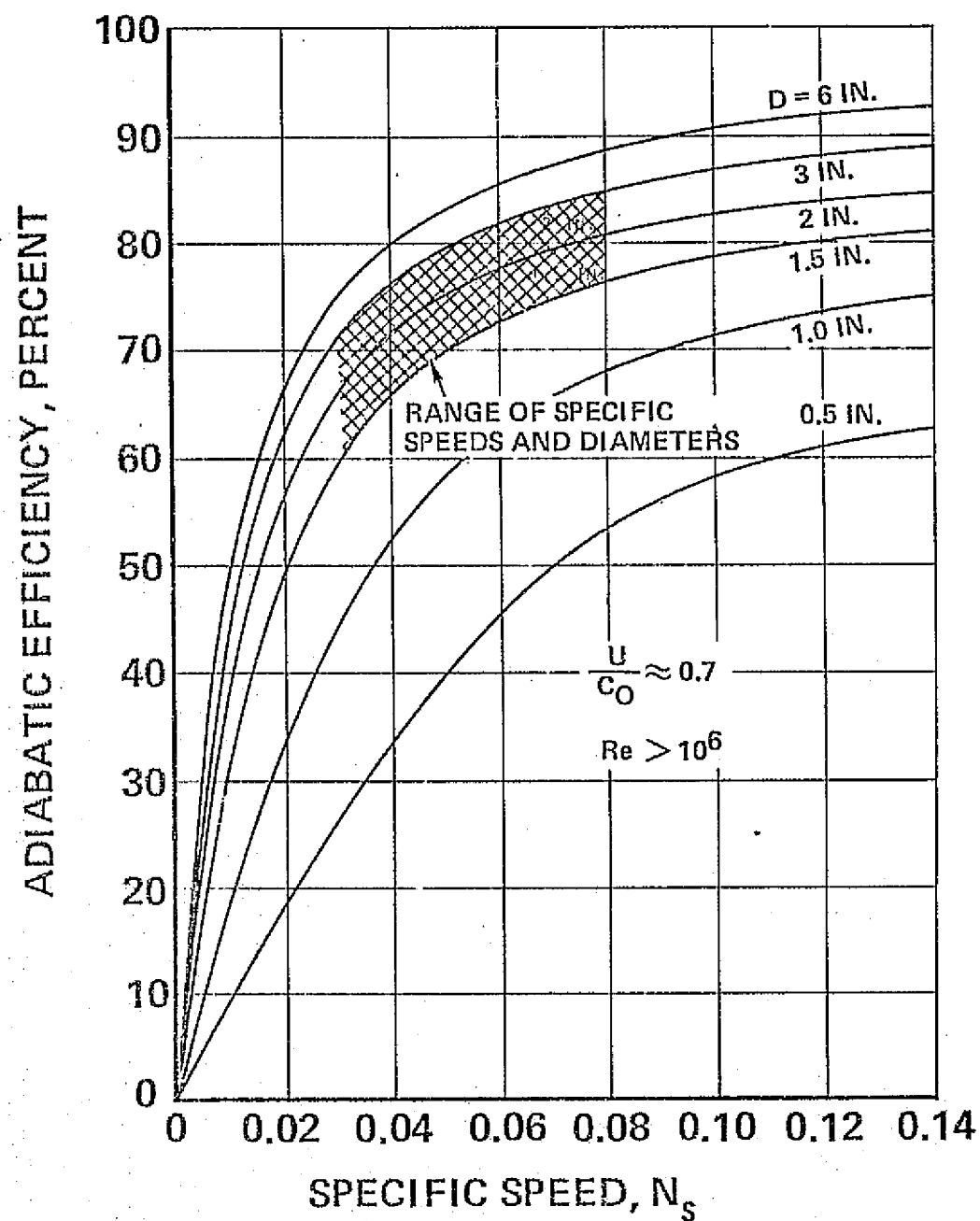
FAN CHARACTERISTICS

Several types of fans or blowers can be used to provide air circulation through air conditioners, 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 units have relatively low efficiency--between 20 and 40 percent. Efficiencies as high as 65 percent are typical of larger capacity squirrel cage blowers (up to 25,000 cfm) 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 done by providing the speed flexibility afforded by a belt drive.

Fan data which will be 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 ΔP'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. Final fan selection and design will be done under Tasks 5 and 6.

FAN CHARACTERISTICS

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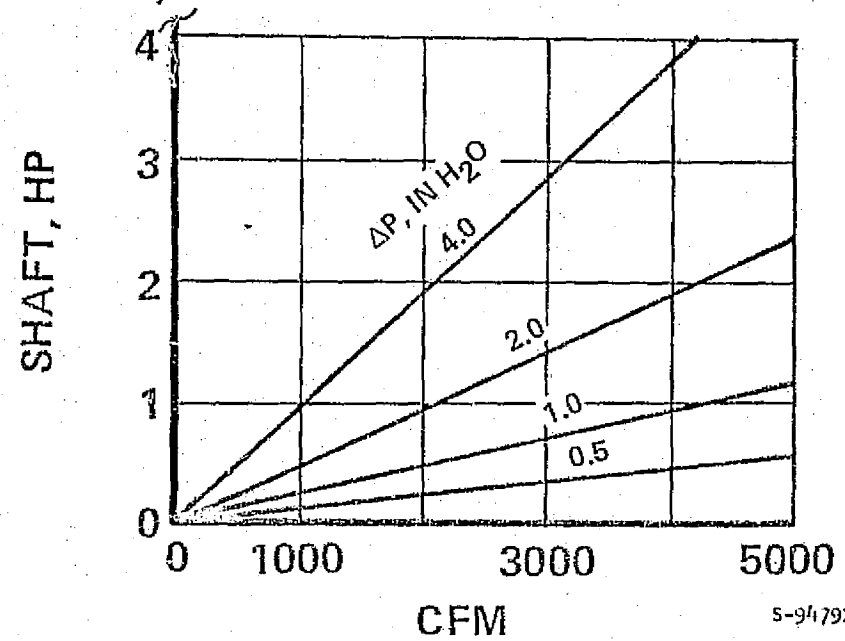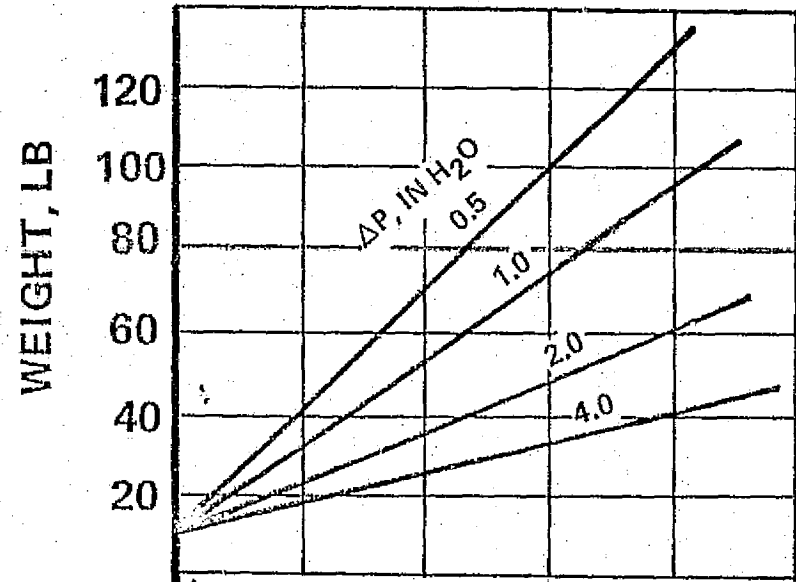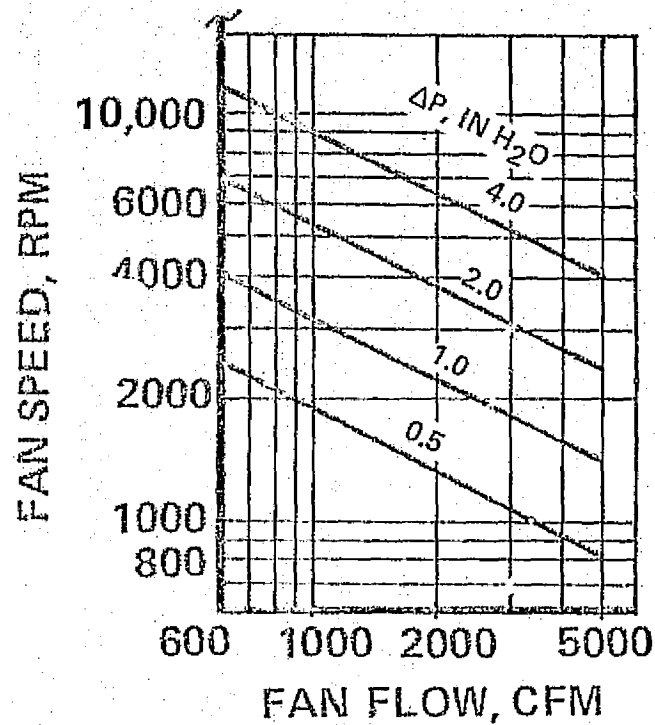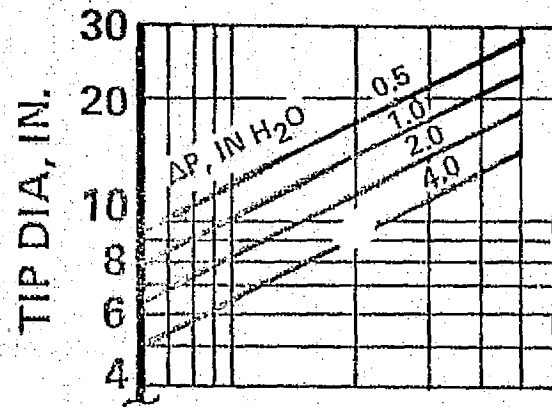
FINNED TUBE HEAT EXCHANGER CHARACTERISTICS

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 condenser. This type of heat exchangers can be fabricated very cheaply, primarily because of the mechanical bonds between the fins and the tubes. As mentioned previously, the factory sell price for these units is about $1.80/lb. By comparison, aluminum plate-fin units in production quantities of over 10,000 per year are sold (EOM) for about $4.50/lb. It is doubtful that the price of these plate-fin units could be reduced to the $1.80/lb level, even with production quantities of 100,000 per year.

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 in the listing below. Commercial units are usually designed with air velocities of about 500 ft/min. At this velocity, the heat transfer conductances (hA, Btu/hr ^oF) on the air and refrigerant sides are about the same (235 Btu/hr ^oF).

The heat transfer and pressure drop correlations listed below correspond to dry operation of the heat exchangers. In the computer program, allowances are made to (1) determine whether the surface is wet or dry, and (2) calculate wet performance, if necessary, from wet bulb temperature and total heat content air data.

FINNED TUBE HEAT EXCHANGER CHARACTERISTICS

GEOMETRY

10 FINS/IN., ALUMINUM, .006 IN. THICK
0.375 IN. TUBES, COPPER, TRIANGULAR PITCH 1 IN. C/C.

SPECIFIC HEAT TRANSFER AREA

OUTSIDE   225 FT^2/FT^3
INSIDE   16 FT^2/FT^3

CORE WEIGHT

23 LB/FT^3

FRAME WEIGHT

10% OF CORE (MAX.)

HEAT TRANSFER COEFFICIENT

OUTSIDE $h = 0.3 (V)^{0.625}$ BTU/HR-FT^2-$^\circ$F (DRY)
WHERE V IS FACE VELOCITY, FT/MIN
INSIDE (PHASE CHANGE) $h = 200$ BTU/HR-FT^2-$^\circ$F

PRESSURE DROP

$2.3 \times 10^{-6} V^{1.69} N^{0.746}$, IN H_2O (DRY)
WHERE N IS NUMBER OF TUBE ROWS.

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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

  Condenser temperature from 85° to $110^{\circ}F$

  Boiler temperature from 170° to $190^{\circ}F$

All computations were done for an evaporator temperature of $45^{\circ}F$.

Plotted below are the data obtained for an air conditioning load of 1-1/2 tons and a boiler inlet temperature of $180^{\circ}F$. These plots are typical of the other capacities and boiler temperatures investigated. The cycle COP is defined as follows:

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

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

As illustrated in these plots, R-12 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 will not be considered further 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

BOILER INLET TEMPERATURE: 355.4K (180°F)
AIR CONDITIONING LOAD: 5.3KW (1-½ TONS)

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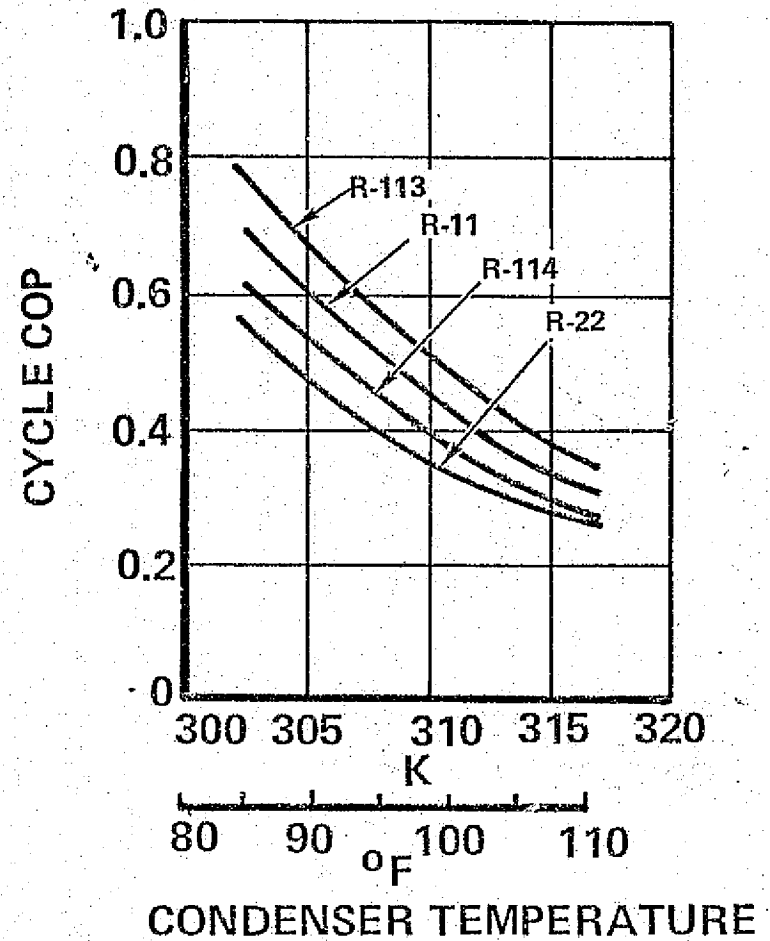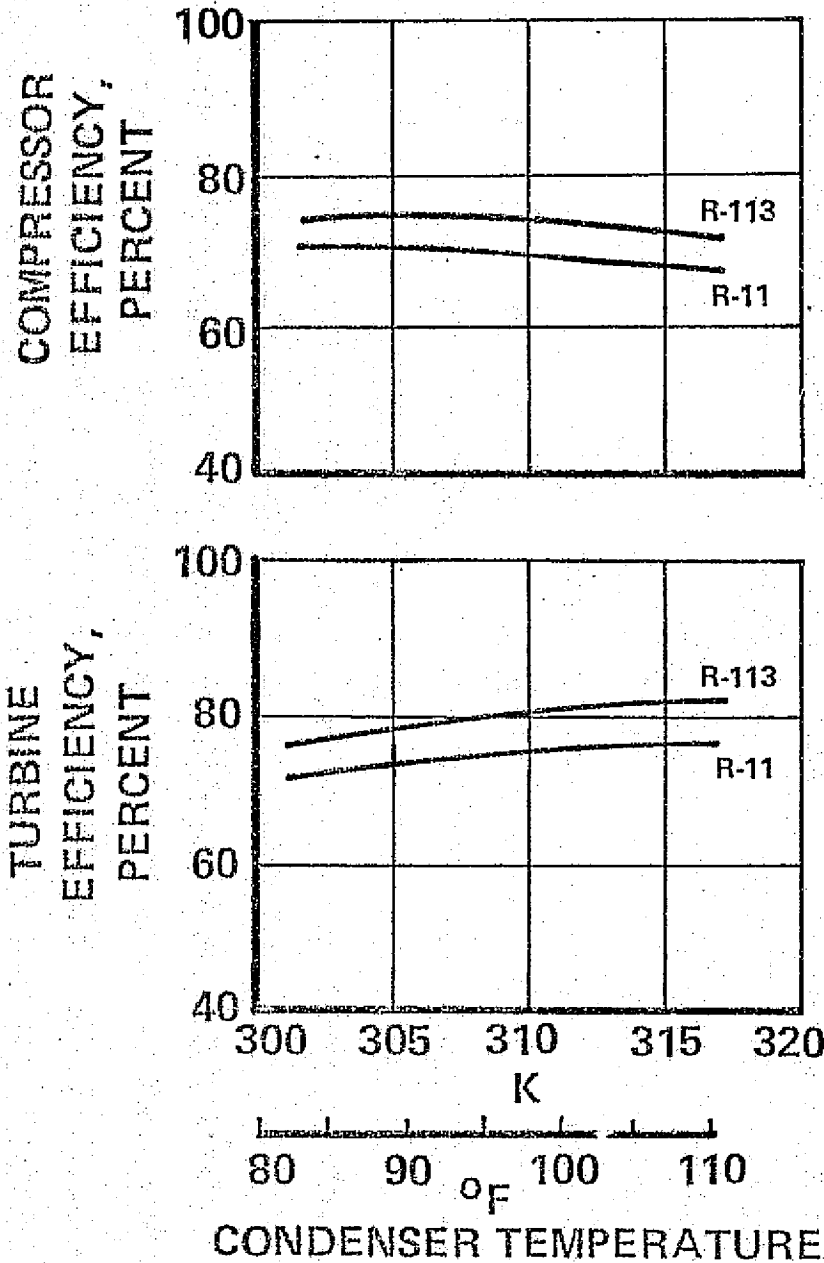
FLUID SELECTION

Refrigerants for conventional residential-commercial air conditioners are selected on the basis of considerations other than 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 very 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 is selected for further system studies.

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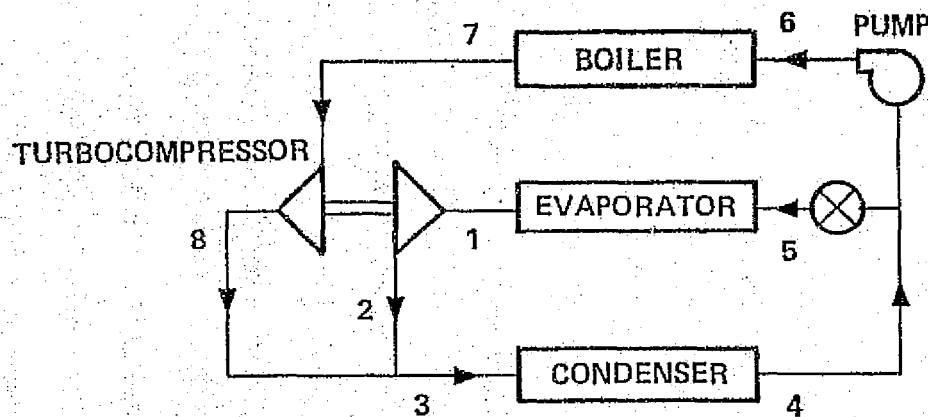
STATION	REFRIGERANT PRESSURE, PSIA	
	R-113	R-11
1*	3.04	8.0
2	13.4	29.8
3	13.4	29.8
4**	12.8	28.4
5	3.19	8.4
6	43.3	84.2
7***	41.2	80.2
8	13.4	29.8

*EVAPORATOR TEMPERATURE: 45°F
**CONDENSER TEMPERATURE: 110°F
***BOILER TEMPERATURE: 180°F

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

GARRETT

s-94813

POWER AND REFRIGERATION LOOP EFFICIENCIES

The cycle analysis computer program was used to generate power and refrigeration loop efficiencies for a range of condenser and boiler temperatures. Such data for a 3-ton system are presented below. The temperatures shown are cycle temperatures, i.e., working fluid temperatures.

The thermodynamic limitations imposed on the power loop by the low boiler temperatures are evident in this plot. With a boiler at 190°F, the power loop efficiency will be lower than 12 percent, even with a condenser effectiveness of 1.0. The effect of boiler and condenser temperatures on the power loop efficiency is about the same; the efficiency of the loop will drop by roughly 1 percent for every degree Fahrenheit of temperature drop in either the boiler or the condenser.

The COP of the refrigeration loop is significantly more sensitive to condensing temperature than the power loop efficiency; the slope of the COP curve (at $T_{cond} = 100°F$) represents a 1.6 percent COP change per degree Fahrenheit of condenser temperature.

This difference in sensitivity to condenser temperature suggests that separate condensers should be used for maximum overall efficiency. With an air condenser, the cooling loop condensing temperature could be decreased by 5 to 6°F in this manner with resultant overall system COP gains as high as 12 percent. This is very significant when it is realized that the size of the solar collector and thermal storage subsystem is a direct function of the overall system COP.

POWER AND REFRIGERATION LOOP EFFICIENCIES

3-TON UNIT
REFRIGERANT R-11
EVAPORATOR TEMPERATURE: 45°F

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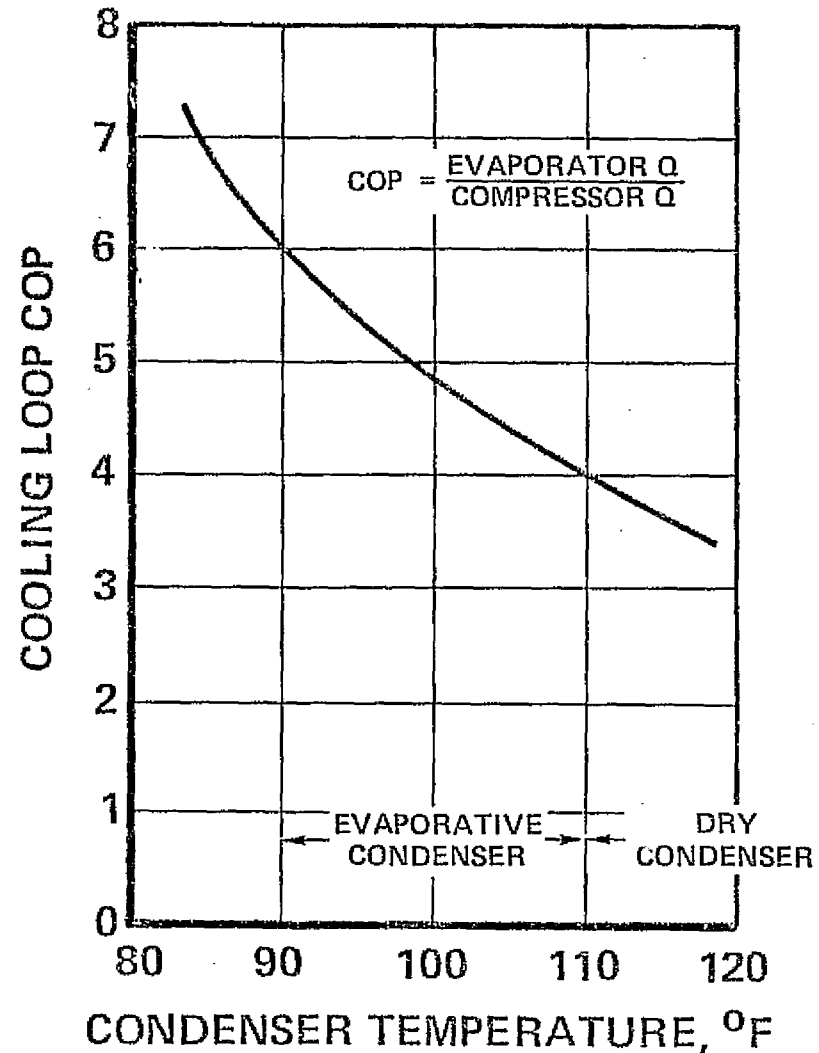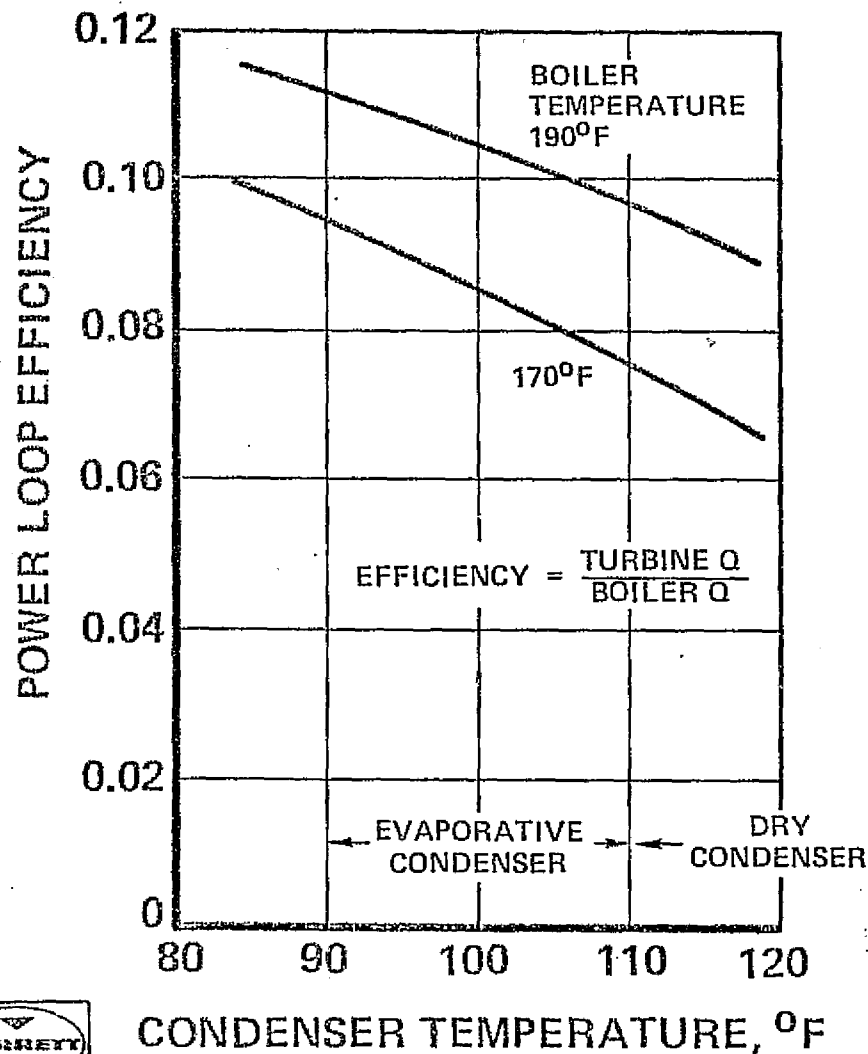
OVERALL SYSTEM PERFORMANCE

The performance of the complete Rankine air conditioner is expressed by the product of the power loop efficiency and refrigeration loop COP. (Mechanical losses, taken as 5 percent, are included in the compressor work.) Since both of these factors are strong functions of condenser temperature, it follows that this parameter is by far the most significant factor in designing the system for high COP. Quantitatively, the plot below shows that a drop in condenser temperature of 1°F has the same effect on system COP as a boiler temperature drop of 3°F.

Four types of condensers may be considered:

(a)   Water-cooled unit--With a cooling tower as the ultimate heat sink. Water temperatures of about 80°F can be obtained with this arrangement. Low approach temperatures can also be achieved because of the very high heat transfer coefficient attainable with water as the coolant. Condenser temperatures of 95°F are reasonable; with a boiler temperature of 190°F, system COP is estimated at 0.6.

(b)   Evaporative condenser--With this type of unit, water evaporation takes place directly on the condenser coil surface so that the effective temperature of the heat sink approaches the wet bulb temperature of the air. Condensing temperature less than 90°F can be achieved corresponding to a system COP of 0.7 at a boiler temperature of 190°F.

(c)   Dry condenser with condenser air humidifier--In this case, humidification of the condenser airstream will result in lowering of the dry bulb temperature to a level approaching the wet bulb temperature. At design point this will amount to about 20°F. Condensing temperatures of about 95°F can be expected with this scheme. Again, with a boiler temperature of 190°F, a COP of 0.6 is estimated.

(d)   Dry condenser--This represents the worst arrangement in terms of efficiency, as illustrated in the plot below. COP's lower than 0.4 will be achieved with this approach.

Although system data are necessary for final evaluation of these four condenser approaches, the results of the cycle analyses performed suggests that some form of evaporative cooling will have to be provided to enhance the heat sink potential of ambient air at design point conditions (95°F db, 75°F wb).

OVERALL SYSTEM PERFORMANCE

S-94019
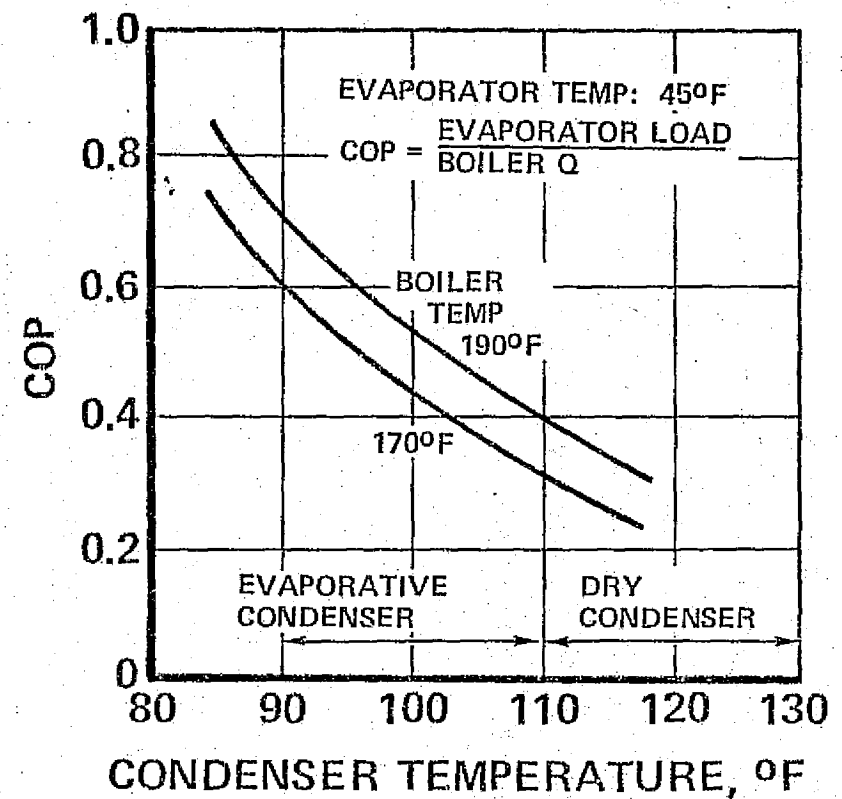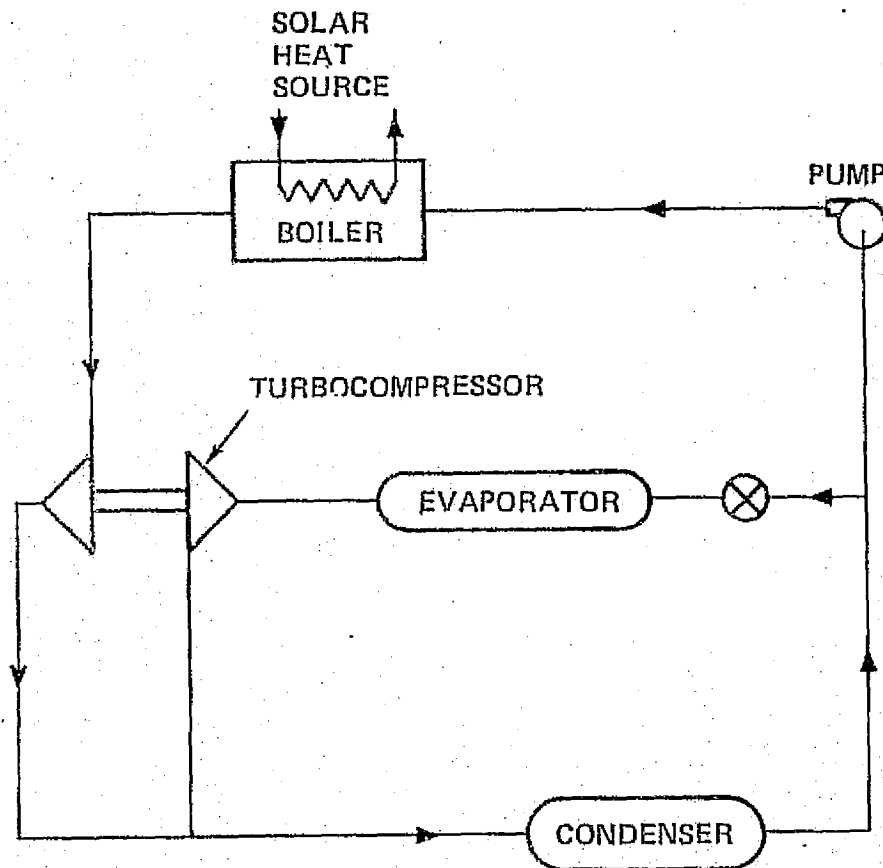
COMPARISON OF RANKINE AND ABSORPTION SYSTEM

A comparison was made, based on cycle data, of the performance of Rankine-powered and LiBr/H_2O absorption air conditioners. The COP's plotted below are defined as

$$COP = \frac{\text{evaporator load}}{\text{thermal energy input}}$$

The absorption cycle data were generated using a computer program developed at AiResearch for the analysis of absorption cycle refrigeration systems. The computation procedure involves iterative material and heat balance calculations which are repeated until the concentrations of the solution in the absorber and the generator satisfy the thermodynamic and mass transfer requirements established by the input parameters.
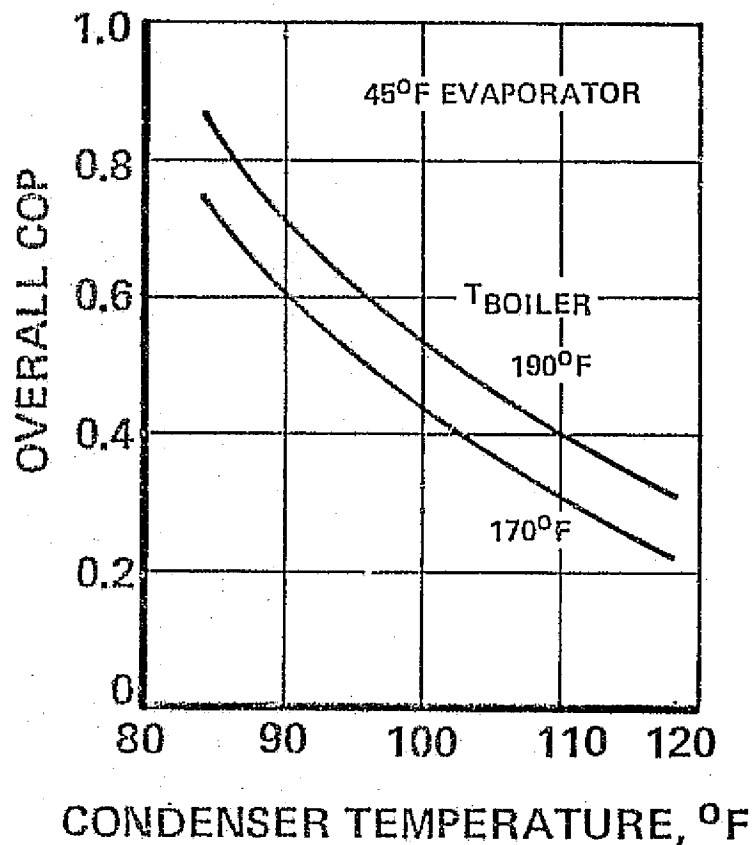
The COP plots represent design data for the parameters shown and should not be interpreted to derive off-design performance information.

Relative to the Rankine-powered system, at a given condenser temperature, say $100^{\circ}F$, cycle COP will drop from 0.53 to 0.44 when the design temperature drops from 190 to $170^{\circ}F$. By comparison, the absorption cycle performance plot reveals that for the same condenser temperature ($100^{\circ}F$), as the design generator temperature drops to $170^{\circ}F$ the system COP drops to 0, indicating that operation under such conditions is impossible.
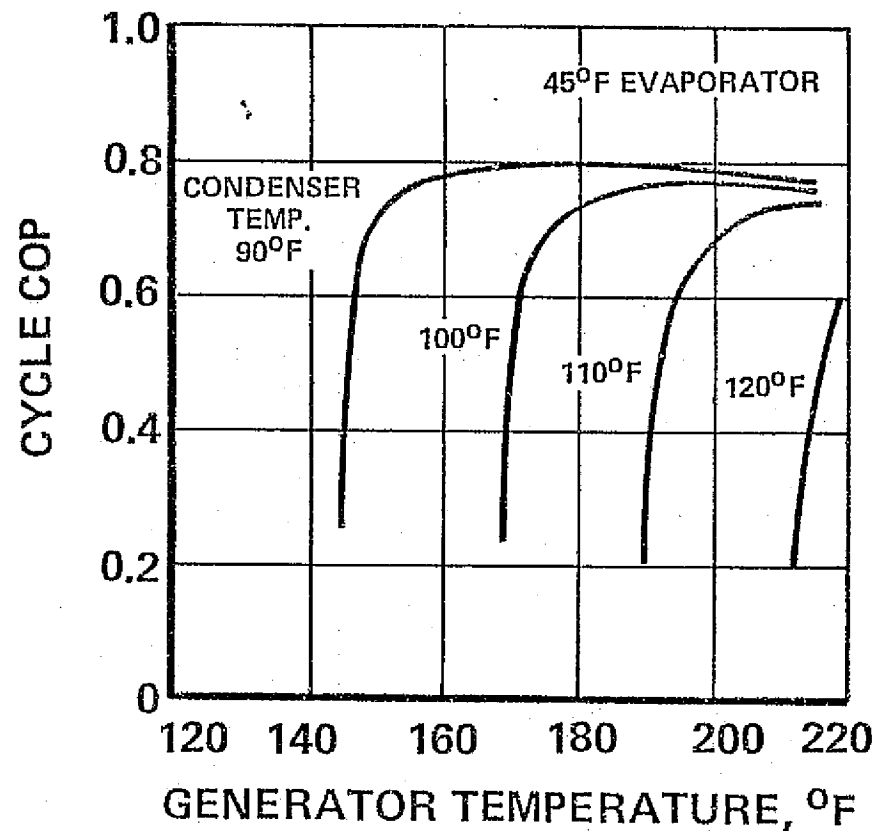
This illustrates the limitations of the LiBr/H_2O cycle in comparison to the Rankine powered cycle.

COMPARISON OF RANKINE
AND ABSORPTION CYCLES

RANKINE CYCLE PERFORMANCE

45°F EVAPORATOR

T_{BOILER}

190°F

170°F

OVERALL COP

CONDENSER TEMPERATURE, °F

Li Br/H_2O ABSORPTION CYCLE PERFORMANCE

45°F EVAPORATOR

CONDENSER TEMP. 90°F

100°F

110°F

120°F

CYCLE COP

GENERATOR TEMPERATURE, °F

5-94816

RANKINE POWERED SYSTEM AUGMENTATION

Any solar powered system will require augmentation to assure operation when thermal energy from the solar source is not available in sufficient quantity to satisfy the energy demand established by the loads. This could occur as a result of (1) extensive periods of overcast, (2) nighttime operation, (3) system overload, or (4) solar collector or thermal storage subsystem malfunction.

Several techniques can be used for system augmentation. Most of these are wasteful of high quality energy; others entail system complexity which appears unacceptable for residential application. System augmentation can be effected by heat addition to the heat transport fluid from the solar energy system to the Rankine boiler. Heat can be added through combustion of natural gas or fuel oil or by dissipation of electrical power. In terms of energy conservation, the use of electricity, rather than gas or oil, is wasteful as a heat source. In this case the energy added to the system is used in the Rankine power loop at an efficiency of about 10 percent, as shown previously. This approach appears extremely wasteful of energy resources.

A similar augmentation technique would be the addition of heat energy directly into the Rankine loop working fluid upstream of the turbine. The refrigerant pressure upstream of the turbine will be determined by the boiler temperature so that heat addition downstream of the boiler will result in working fluid superheat. Under these conditions, the power loop efficiency will be even lower than at design conditions. This approach is not recommended.

In terms of overall energy usage, it appears that the optimum approach to system augmentation is by supplementing the turbocompressor capability with an auxiliary compressor. This auxiliary compressor can be driven electrically or through a high-temperature organic Rankine power system. Both approaches are discussed in more detail later. With an electrical compressor, the auxiliary energy utilization efficiency will be as high as 70 percent (motor efficiency). In the case of the organic Rankine system, the energy utilization will be about 25 percent.

One aspect to system augmentation which warrants serious consideration is the peaking demand imposed on the gas or electricity supplier. Since all solar-powered systems in a given geographic location will usually require augmentation at about the same time, the gas or electrical energy demand on the utility companies will create a severe problem that will undoubtedly result in high installed capacity and therefore high rates. One of several approaches to reduce peak demand is to supplement the system through the use of fuel oil, which is basically stored by the consumer and replenished at regular intervals.

SYSTEM AUGMENTATION

TECHNIQUES

- HEAT ADDITION TO WATER LOOP

- HEAT ADDITION TO POWER LOOP

- AUXILIARY COMPRESSOR

  ELECTRIC DRIVEN

  RANKINE ENGINE DRIVEN

CONSIDERATIONS

- ENERGY UTILIZATION EFFICIENCY

- PEAK DEMAND ON UTILITY COMPANY

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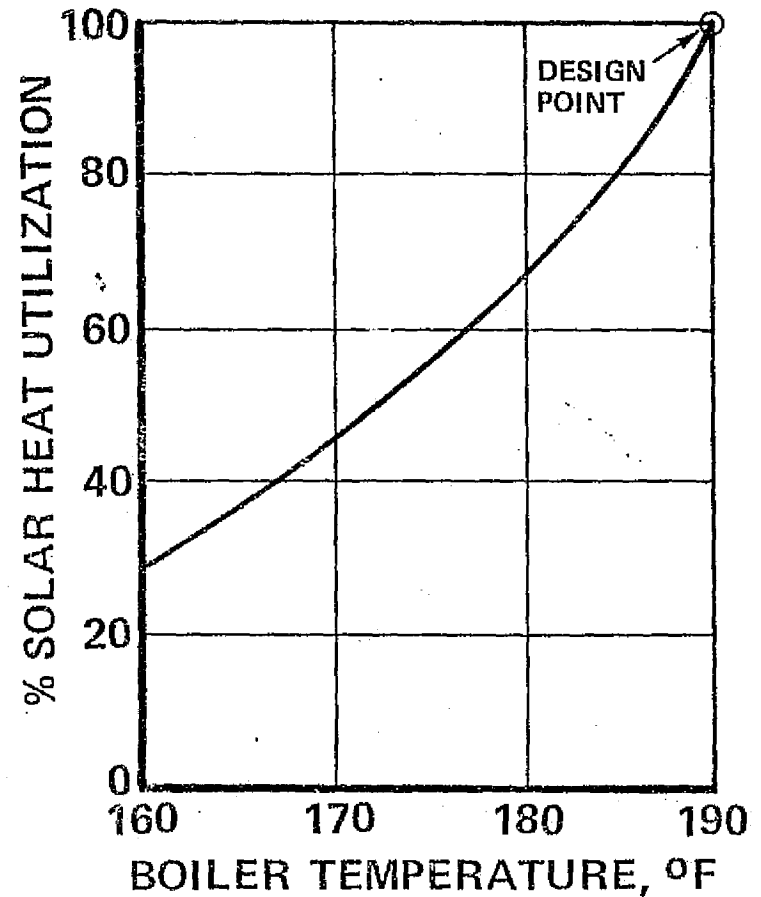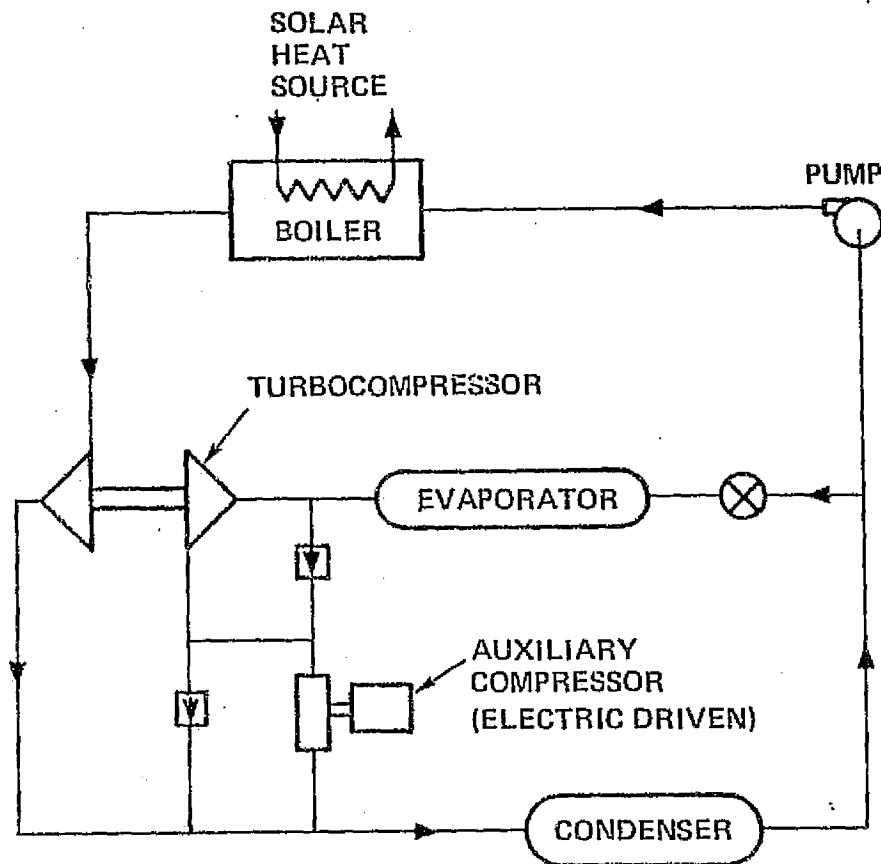
ELECTRICALLY AUGMENTED SYSTEM

An electrically augmented system is depicted below. The auxiliary compressor can be designed to carry the entire load if necessary. Check valves are provided to prevent recirculation within the refrigeration loop. Operation is as follows, assuming constant condenser temperature. As the temperature of the heat source drops, the pressure at turbine inlet will also drop; and the power developed by the turbine will decrease rapidly. This will result in a reduction in the speed of the turbomachine. As a consequence, the evaporator pressure will rise and the working fluid flow through the refrigeration loop will drop. Depending on the demand and the extent of the heat source temperature drop, the system control may command the system to produce more refrigeration than possible with the power available from the turbine. This will be sensed by excessive evaporator pressure. This signal will be used to start the auxiliary compressor, which will act as a boost compressor and restore the desired evaporator pressure.

The plot shown below is an estimate of the fraction of the total power supplied by the Rankine power loop as the boiler temperature drops below design point while all other conditions (condenser temperature and load) remain the same. With a boiler temperature as low as 30°F below design value (190°F), as much as 30 percent of the power necessary to drive the air conditioner at design load is still supplied by the Rankine power loop; the remainder, 70 percent, is auxiliary electrical power.

Under normal cycling through day and night, the temperature of the boiler heat source will drop at night. However, under these cyclic conditions, the air conditioning night demand will also be reduced. More significant, as discussed previously, will be the nocturnal drop in the temperature of the heat sink and the corresponding reduction in the temperature of the condenser. Data from the NASA solar house indicate that in general the water tower temperature level dropped by about 10°F at night (in August). This would be adequate to make up for a significant drop in boiler temperature (on the order of 20 to 30°F).

ELECTRICALLY AUGMENTED SYSTEM

$$\text{ELECTRICAL COP} \left(= \frac{Q_{EVAP}}{\text{MOTOR POWER}}\right)$$

ESTIMATED AT 5.5 WITH BOILER AT 170°F AND CONDENSER AT 105°F

S-94013

HEAT AUGMENTED SYSTEM

The schematic below shows a method for augmenting the baseline Rankine air conditioner by means of a high-speed centrifugal compressor in the refrigeration loop. In this case, the auxiliary compressor is driven through a magnetic coupling by a high-temperature organic Rankine power system. The efficiency of the auxiliary power loop can be as high as 28 percent with turbine inlet temperatures of 800°F. In terms of overall energy utilization, this system is comparable to the electrically augmented approach discussed previously. One potential advantage of this arrangement is that the burner could be fired with fuel-oil, thus eliminating peak demands on the gas or electric utility companies.

The complexity of this approach may be warranted for a commercial or industrial installation of significant tonnage. However, for a residential air conditioner, the system appears too complex and probably too costly.

Possibly the only thermal augmentation approach that may be competitive with the electric-driven auxiliary compressor may be heat addition to the solar collector/storage circuit through combustion of fuel oil. Although the energy utilization efficiency is only about half (10 percent of heat input) of that of the electric driven compressor (70 percent of electric power input or 20 percent of power plant heat input), elimination of the peak energy demands may be sufficient to justify the inefficiency of this approach. This will depend essentially upon the total quantities of auxiliary energy required for operation.

HEAT (GAS OR OIL) AUGMENTED SYSTEM
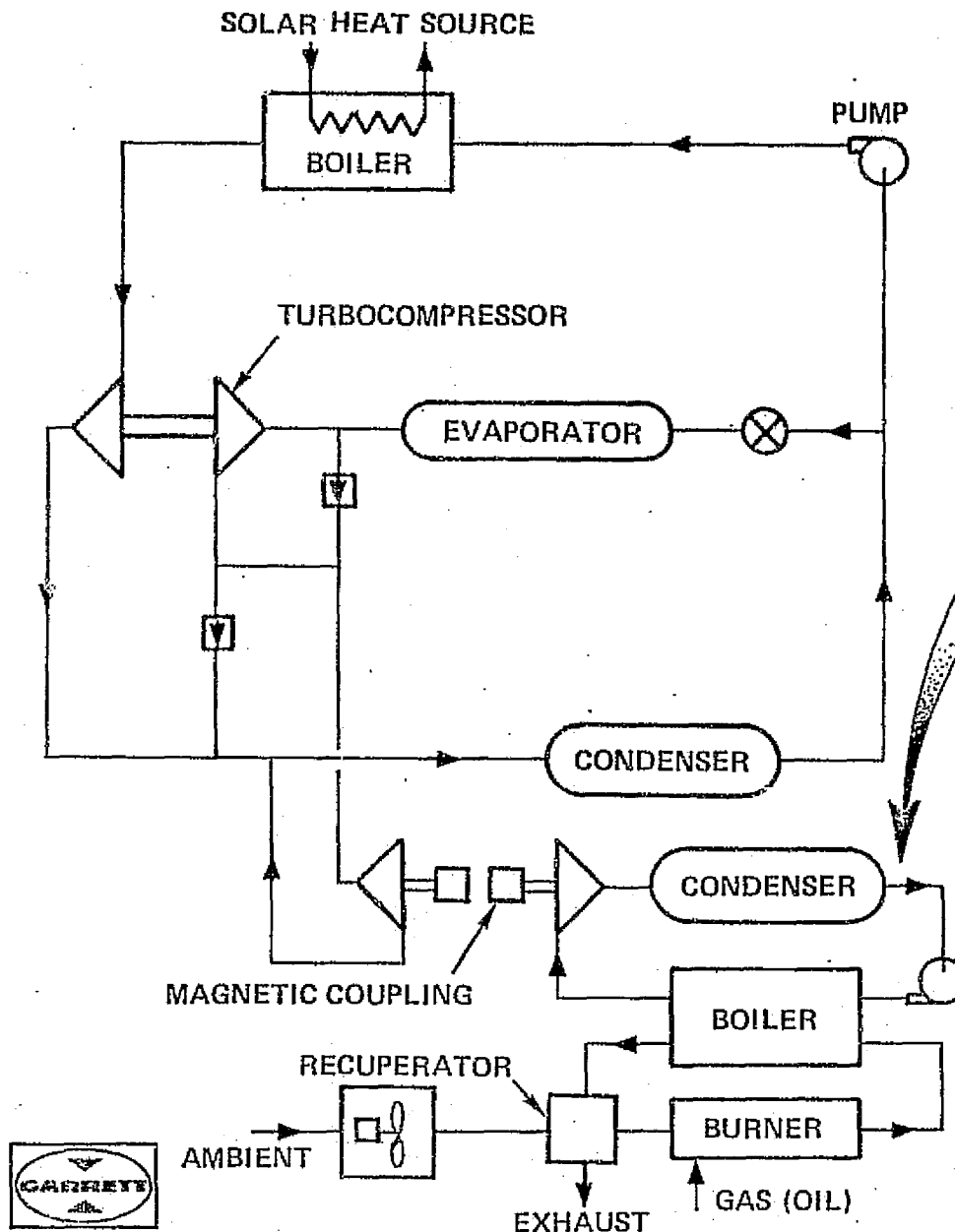
AUXILIARY RANKINE POWER LOOP

- FLUID: CP-25 (TOLUENE)

- BOILER OUTLET TEMP: 800°F

- POWER LOOP EFFICIENCY: 28%

- ENERGY UTILIZATION EFFICIENCY: 25%

- GAS COP $\left(\dfrac{Q_{EVAP}}{Q_{GAS}}\right)$ AS HIGH AS 3.0 (180°F HEAT SOURCE)

S-94030

CONCLUSIONS AND FUTURE WORK

CONCLUSIONS

The listing below summarizes the major study conclusions. The effectiveness of the Rankine-powered air conditioner is comparable to that of the absorption and adsorption cycles. The off-design performance of the Rankine approach makes it more attractive for use over a range of heat source/heat sink temperatures such as those available in a solar powered/ambient cooled system.

Basically, the overall efficiency of the system is limited by the Carnot cycle efficiency expressed in terms of source and sink temperatures. The source temperature level (200 to 170°F) has been specified by NASA as a result of extensive solar collector/thermal storage system tests. Only slight gains could be made here at the cost of solar collector efficiency and/or energy storage subsystem capacity.

The sink temperature level, however, and the quality of the heat sink in terms of capacity, can be enhanced through water evaporation techniques. The beneficial effect of lowering the temperature of ambient air through evaporation has been shown in the data presented. It appears that such a technique will be necessary in order to achieve relatively high overall COP's and thus reduce the size of the solar collector necessary for operation. A major portion of the effort will be devoted to the selection of the optimum "evaporative" condenser approach.

CONCLUSIONS

- THE ADSORPTION PROCESS MAY OFFER SIGNIFICANT ADVANTAGES BY COMPARISON TO BOTH RANKINE AND ABSORPTION AIR CONDITIONERS.

- R-11 IS SELECTED AS THE REFRIGERANT.

- IN THE SIZE OF INTEREST FOR RESIDENTIAL APPLICATION COMPRESSOR AND TURBINE EFFICIENCIES AS HIGH AS 80 AND 83 PERCENT RESPECTIVELY ARE PREDICTED.

- CONDENSER PERFORMANCE ENHANCEMENT BY WATER EVAPORATION APPEARS HIGHLY DESIRABLE.

- THE COP OF RANKINE POWERED AIR CONDITIONERS IS COMPARABLE TO THAT OF THE ABSORPTION CYCLE AT CONDITIONS WHERE THE ABSORPTION CYCLE CAN BE OPERATED.

- THE MAJOR ADVANTAGE OF THE RANKINE POWERED AIR CONDITIONER IS THE CAPABILITY OF THE SYSTEM TO OPERATE AT OFF-DESIGN CONDITIONS.

- FOR MAXIMUM ENERGY UTILIZATION ELECTRICAL AUGMENTATION IS RECOMMENDED.

- FOR MINIMUM PEAK DEMAND ON THE ELECTRIC UTILITY COMPANIES FUEL-OIL AUGMENTATION IS POSSIBLE.

S-94797

FUTURE WORK

The overall program logic has not been changed as a result of the data generated thus far. However, the emphasis on some of the program tasks has been changed. The major changes involve (1) extension of the screening analyses of Task 4, and (2) the inclusion under Task 1 of the adsorption process investigations to demonstrate feasibility. Otherwise the program remains as originally conceived.

Future activities under the program tasks and subtasks are listed below.

FUTURE WORK

SYSTEM CLASSIFICATION (TASK 1)

1. STATE-OF-THE-ART SURVEY
2. ADSORPTION PROCESS FEASIBILITY

REQUIREMENTS ANALYSIS (TASK 2)

UPDATE AND COMPLETE DEFINITION PRIOR TO SYSTEM OPTIMIZATION

SCREENING ANALYSIS (TASK 4)

1. CHECKOUT SYSTEM DESIGN COMPUTER PROGRAM
2. INVESTIGATE CANDIDATE CONDENSER APPROACHES
3. GENERATE PARAMETRIC DESIGN DATA
4. SELECT BASIC SYSTEM

SYSTEM OPTIMIZATION (TASK 5)

1. COMPLETE GENERALIZED COMPRESSOR AND TURBINE MAPS
2. DEVELOP OFF-DESIGN COMPUTER PROGRAM
3. SELECT DESIGN POINT
4. CHARACTERIZE SYSTEM IN TERMS OF COST, ENERGY UTILIZATION, AND INTERFACE REQUIREMENTS

SELECTED SYSTEM PRELIMINARY DESIGN (TASK 6)

1. DEVELOP COMPONENT PROBLEM STATEMENTS
2. PREPARE PACKAGE LAYOUTS
3. DEVELOP PRELIMINARY SPECIFICATION

S-94801