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Screening Analysis

DEVELOPMENT OF A SOLAR-POWERED RESIDENTIAL AIR CONDITIONER

Contract NAS8-30758

74-10996(7)

July 25, 1975

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Prepared for

George C. Marshall Space Flight Center
National Aeronautics and Space Administration
Marshall Space Flight Center
Huntsville, Alabama 35812

GARRETT

AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

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CONTENTS

INTRODUCTION

This report summarizes the results of screening analyses aimed at the definition of an optimum configuration of a Rankine-cycle solar-powered air conditioner designed for residential applications. These investigations are conducted in fulfillment of Task 4 of Contract NAS8-3078. 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, the screening analyses were concerned with the generation of parametric design data for the four different condenser approaches defined in Figure 1 and identified as follows:

(a) Concept A--Ambient air condenser

(b) Concept B--Humidified ambient air condenser

(c) Concept C--Evaporative condenser

(d) Concept D--Water condenser (with a cooling tower)

All systems considered feature a high-performance turbocompressor and a single refrigerant (R-11) for the power and refrigeration loops. The selection of R-11 as the working fluid is supported by the results of fluid evaluation studies reported in Reference 1.* The data (presented in subsequent discussions) 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) Auxiliary electric power for fans and pumps

(c) System installed cost or cost to the user

BASELINE DESIGN CONDITIONS

For the purpose of comparison, the following interface conditions were used to generate parametric system characteristics:

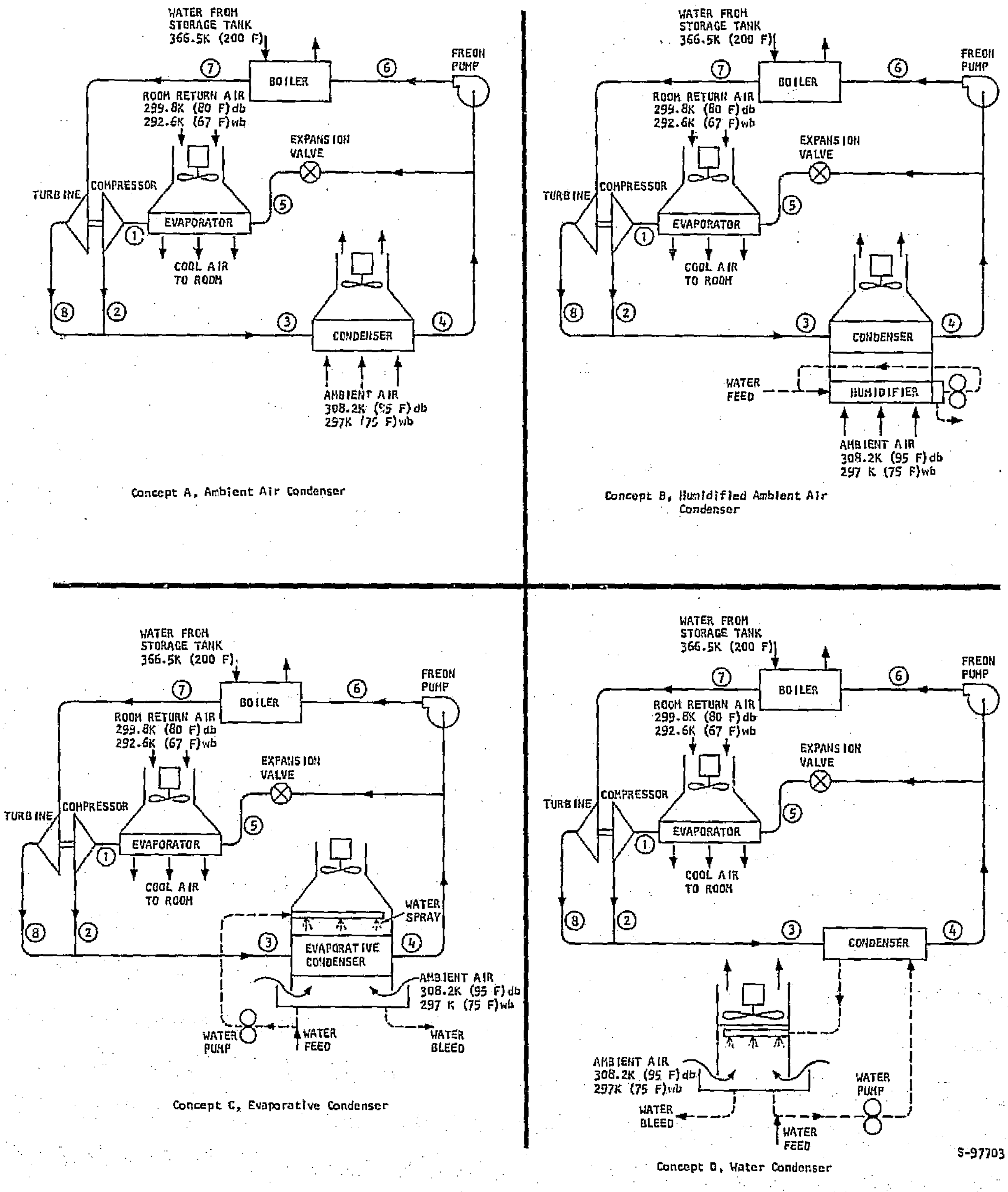
(a) Water temperature at boiler inlet (thermal storage temperature): 366.5 K (200 F)

(b) Room return air temperatures: 299.8 K (80 F) db, and 292.6 K (67 F) wb

(c) Ambient air temperatures: 308.2 K (95 F) db, and 297 K (75 F) wb

The 366.5 K (200 F) water temperature from the thermal storage units is representative of the level attainable from a flat-plate solar collector. The

*References are presented at the end of this document (before appendixes).



Concept A, Ambient Air Condenser

Concept B, Humidified Ambient Air Condenser

Concept C, Evaporative Condenser

Concept D, Water Condenser

Figure 1. Candidate System Configurations

room return air and ambient air temperatures are those specified by the Air Conditioning and Refrigeration Institute (ARI) for the purpose of rating air conditioners. All screening analyses were performed for a 10.5-kw (3-ton) air conditioner capacity.

METHODOLOGY

The logic used by the computer program is illustrated in Figure 2. Basically, the computation of component and system characteristics follows the approach described in Reference 2. Cycle parameters defining the conditions of the refrigerant within the power and refrigeration loops of the 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 also to assure design feasibility for the turbomachinery. For this purpose, generalized compressor and turbine performance models were used in the computer program.

The efficiency of a single-stage centrifugal compressor can be determined by analytical and experimental data correlated in terms of the following parameters:

(a) Adiabatic head

(b) Adiabatic head coefficient

(c) Specific speed

(d) Tip mach number

(e) Reynolds number

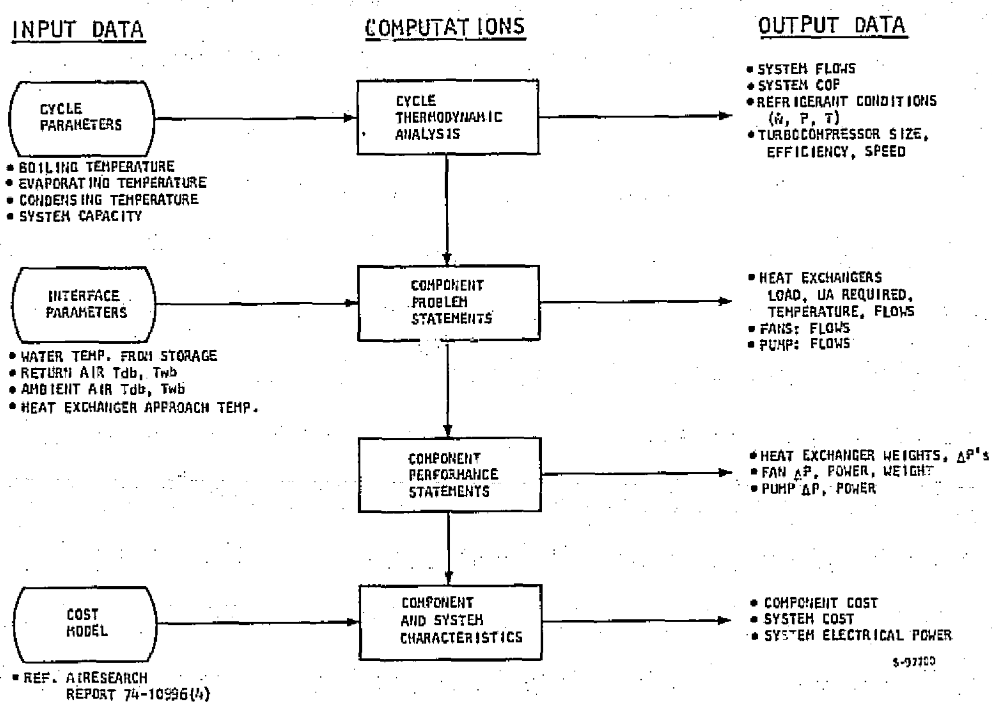


Figure 2. Methodology Used in Screening Analyses

The data of Figure 3 show the achievable efficiency of centrifugal compressors plotted as a function of specific speed and tip Mach number. The plot is based on experimental data extending to specific speeds as low as 0.02. The data are representative of recent machines that feature efficient exit diffusers and are fabricated using modern techniques to minimize friction losses by smooth surface finishes and assure high volumetric efficiency by maintaining close tolerances throughout. These fabrication constraints do not preclude low production cost as evidenced by present reciprocating engine turbocharger technology.

The efficiency plot of Figure 3 corresponds to impeller diameters larger than 10.2 cm (4 in.) and Reynolds numbers higher than 10^6. For smaller compressor sizes and lower Reynolds numbers, the efficiency obtained from Figure 3 must be corrected to account for additional losses. The size correction factor also derived from empirical correlation is shown in Figure 4. The Reynolds number correction factor can be computed by

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

where η is the corrected efficiency and η^* is the efficiency determined for $Re > 10^6$.

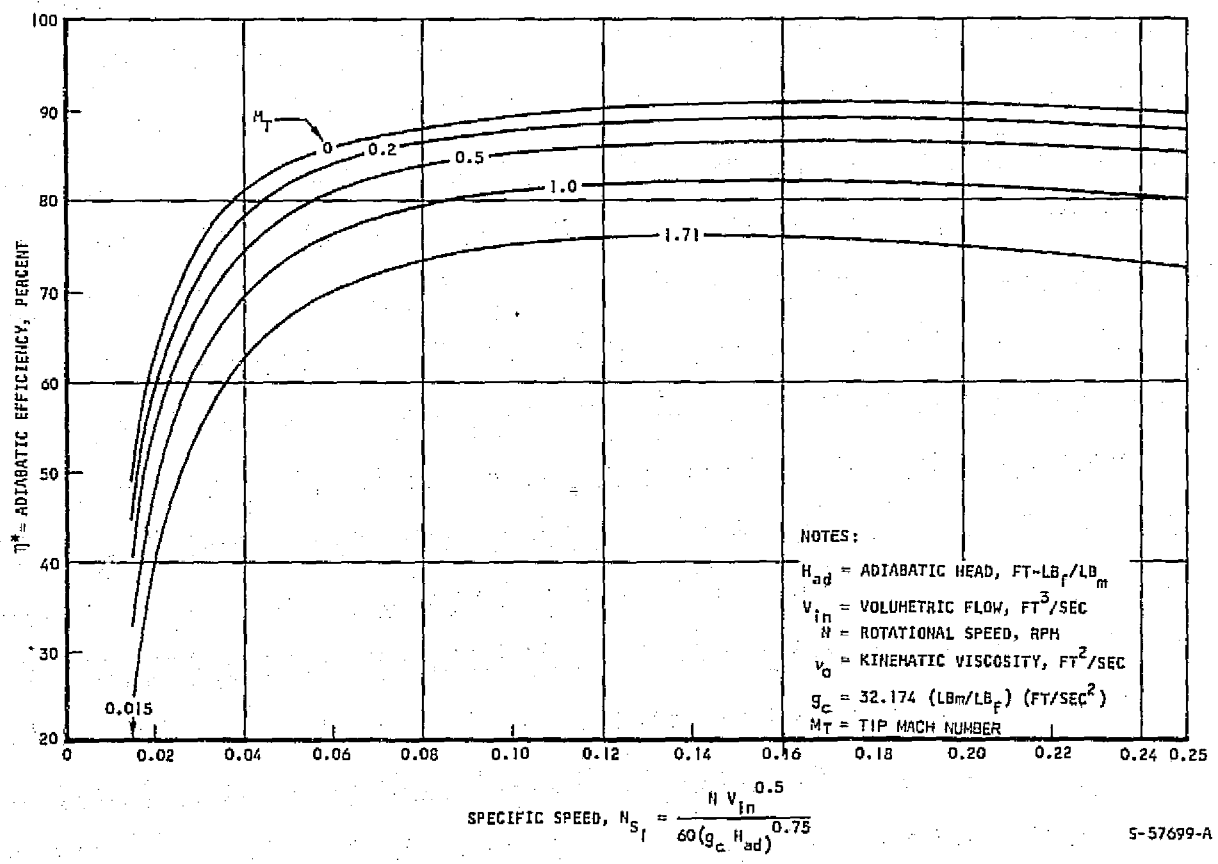


Figure 3. Generalized Centrifugal Compressor Efficiency

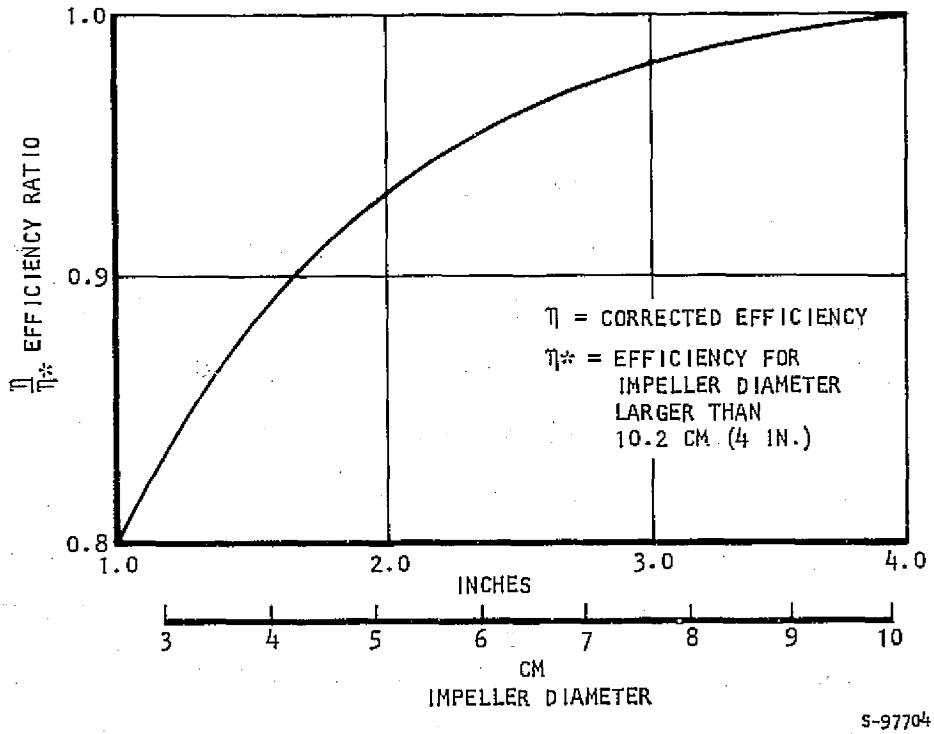


Figure 4. Effect of Impeller Size on Centrifugal Compressor Stage Efficiency

As for the centrifugal compressor, radial reaction turbine data have been collected from published literature. These data were correlated and the generalized plot of Figure 5 was prepared. The data below 50 percent represent extrapolation of the test data. Actually, the range of designs used in the screening analysis is well above 50 percent; the reason for the extrapolation is stability of the iterative computer calculations.

The efficiency data of Figure 5 apply to machines with Reynolds numbers larger than 200,000. For lower Reynolds numbers, a correction factor must be applied as follows:

$$\frac{1-\eta}{1-\eta^*} = 0.4 + 0.6\left(\frac{Re}{200,000}\right)^{-0.2}$$

where η is the corrected turbine efficiency and η^* is the value obtained from Figure 5 for Re > 200,000.

Through an iterative procedure designed to match the compressor and turbine speed and power, the computer program determines the system flows, refrigerant conditions, and the system COP. These refrigerant data, or cycle data, are then used together with specified interface and heat exchanger approach temperatures to generate problem statements for the heat exchanger, fans, and pumps. The

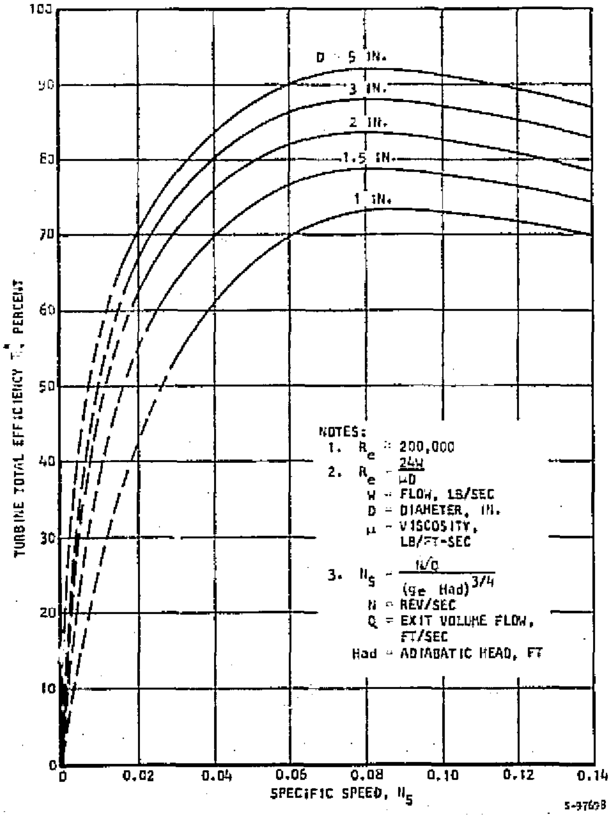


Figure 5. Generalized Efficiency

characteristics of these components are then determined in terms of parameters that can be related to cost. Finally, the models described in Reference 3 are used to determine component and overall system cost.

ASSUMPTIONS

A number of assumptions were made in order to develop relatively simple but sufficiently accurate techniques for the characterization of the components and the entire system. These assumptions are summarized in Table 1. Most of the data concerned with component characterization were derived from commercial equipment catalogs and are representative of typical design conditions for this type of equipment. The equipment cost models are substantiated in Reference 3. The system cost model used is also from the same document; equations used in the computer program are as follows:

System factory cost = 1.65 (Σ major component costs)

User's cost = 6.13 (Σ major component costs)

COMPUTER LISTING

A listing of the computer program is presented in Appendix A, which also includes the nomenclature of the input data. The program was written in Fortran V language for use on the UNIVAC 1108 computer. Examples of the

MAJOR ASSUMPTIONS FOR COI

Cycle analysis	1. Refrigerant properties from publication, by Allied Chemical, "Genetron 11 Thermodynamic Properties", 1957. 2. Saturated vapor and liquid properties used in table form. 3. Vapor and liquid refrigerant assumed ideal fluids with cp vapor = 586.6 J/kg K (0.14 Btu/lb F) and cp liquid = 879.9 J/kg K (0.21 Btu/lb F). 4. R-11 saturated vapor at boiler and evaporator outlet; saturated liquid at condenser outlet. 5. Heat exchanger pressure drop on refrigerant side assumed 5 percent of inlet pressure. 6. Turbocompressor mechanical losses assumed 10 percent of turbine power.
Dry condenser	1. Heat transfer surface: 0.95-cm (3/8-in.) dia copper tubes with wavy aluminum fins. Tube pitch: triangular on 1-in. center. Core density: 560 kg/m^3 (35 lb/ft^3). 2. Overall heat transfer coefficient: 669.6(w/K) tube row/m^2 of face area (118 Btu/hr F) tube row/ft^2 of face area; data from commercial units at face velocity of 2.54 m/sec (500 ft/min). 3. Air-side pressure drop: $\Delta P = 21.9\ (\text{no rows})^{0.746}$ N/m^2 (0.088 (no rows) 0.746 in. H_2O); data from commercial unit at face velocity of 2.54 m/sec (500 ft/min^2). 4. Wrap-up factor: 10 percent of core weight; typical of commercial equipment. 5. Cost: $1.67/kg ($0.76/lb tot.)
Evaporative condenser	1. Heat transfer surface: copper tubes with extended surface inside; tube thickness: 0.41 mm (0.016 in.). 2. Evaporative side heat capacity, q, w = 0.506 ΔhA, where Δh, J/kg, is the log mean difference between the enthalpy of air at the metal temperature and the enthalpy of air at inlet and outlet conditions, and A is the surface area in m^2 (in english units q, Btu/hr = 373Δh(Btu/lb) A(ft^2)). 3. Condensing coefficient is taken as 1702 w/Km^2 (300 Btu/hr ft^2F). 4. Air-side pressure drop taken as 124.4 N/m^2 (0.5 in. H_2O). 5. Cost = $5/kg ($2.3/lb) of core weight. 6. Pump cost: $40 fixed.
Humidifier/ condenser	1. Condenser sizing based on same assumptions as dry condenser above. 2. Humidifier performance: effectiveness of 90 percent assumed; (Tdb in − Tdb out)/ (Tdb in − Twb) = 0.9 3. Humidifier pressure drop taken as 24.9 N/m^2 (0.1 in. H_2O). 4. Humidifier cost: 0.2 (condenser cost).
Liquid condenser	1. Heat transfer surface: copper tubes; tube thickness: 0.41 mm (0.016 in.). 2. Condensing side heat transfer coefficient controlling, h = 1135 w/Km^2 (200 Btu/hr ft^2F). 3. Wrap-up factor: 2.0 4. Cost: $3.3/kg tot ($1.53 lb tot).

FOLDOUT FRAME

LE 1

PONENT AND SYSTEM DESIGN

Cooling tower	1. Water outlet temperature assumed in the system calculations. 2. Fan power: PW in watts = 10.1 Q where Q in kw (PW in watts = 2.96 x 10^{-3} Q where Q in Btu/hr); typical of commercial cooling towers. 3. Water pump power: PW in watts = 6.5 Q where Q in kw (PW in watts = 1.9 x 10^{-3} Q where Q in Btu/hr); typical of commerical equipment. 4. Cooling tower cost: \$(40 + 6.5Q) where Q in kw, \$(40 + 1.905 x 10^{-3}Q where Q in Btu/hr).
Boiler	1. Heat transfer surface: Copper tubes with extended surface inside; tube thickness: 0.41 mm (0.016 in.). 2. Boiling side heat transfer coefficient is controlling; h = 1135 w/K m^2 (200 Btu/hr ft^2). 4. Cost: \$3.3/kg tot. (\$1.53/lb tot.).
Freon pump	1. Wet motor design. 2. Pump efficiency: 50 percent. 3. Pump cost: \$40 (fixed value).
Water pumps	1. Same efficiency and cost as above.
Fans	1. Axial flow blowers 2. ΔP_{TOT}, N/m^2 = 4/3 ΔP_{STAT} + 37 (4/3 ΔP + 0.15 in. H_2O). 3. Fan efficiency: 70 percent; motor efficiency: 70 percent. 4. Fan weight: See Figure 5 of Reference 3. 5. Fan cost: \$1.9/kg (\$0.88/lb)
Motor	1. Motor efficiency: 70 percent 2. Motor cost = \$(10 +32.5 kw) where kw is output power.
Evaporator	1. Heat transfer surface: 0.95-cm (3/8-in.) dia tubes with wavy aluminum fins; tube pitch: traingular on 1-in. center; core density: 560 kg/m^3 (35 lb/ft^3). 2. Overall heat transfer coefficient: (1) dry portion, 669.6 (w/k)/tube row/ m^2 of face area (118 Btu/hr F/tube row/ft^2 of face area; (2) wet or condensing portion, U = 1870 (w/k)/tube row/m^2 of face area (330 But/hr F/tube row/ft^2 of face area). Data from commercial units at face velocity of 2.54 m/s (500 ft/min). 3. Air-side pressure drop, $\Delta P = 21.9 \text{ (no rows)}^{0.746}$, N/m^2, $(0.088 \text{ (no rows)})^{0.746}$ in. H_2O) for dry portion; for wet portion ΔP increases by factors of 1.36. Data from commercial units at face velocity of 2.54 m/sec (500 ft/min). 4. Wrap-up factor: 10 percent of core weight. 5. Cost: 1.67/kg tot (\$0.76/lb tot).

FOLDOUT FRAME 2

computer input and output are presented in Appendix B for the four system concepts defined in Figure 1. The output data include:

(a) Refrigerant temperature, pressure, ethalpy, flow rate, and density at the system stations defined in Figure 1

(b) Heat exchanger flows, temperatures, heat loads, and UA requirement

(c) Heat exchanger weight and cost

(d) Fan characteristics including flow, pressure rise, and power

(e) Wetbulb temperature of the air at inlet and outlet of the evaporator and condenser where applicable

(f) Cycle characteristics: power loop efficiency, refrigeration loop COP, and overall system COP. COP is defined as follows:

$$\text{Refrigeration loop COP} = \frac{\text{refrigeration load}}{\text{compressor power input}}$$

$$\text{Overall system COP} = \frac{\text{refrigeration load}}{\text{boiler heat input}}$$

(g) Turbine and compressor characteristics: efficiency, impeller diameter, and speed

(h) Electric power requirements for the fans and pumps

(g) System cost data

The program was written using the english system of units as defined in the nomenclature and the output data printouts.

SCREENING ANALYSIS RESULTS

As mentioned previously, four condenser arrangements defined as concepts A through D were investigated. Design point data were generated for a range of cycle conditions (refrigerant temperatures) and heat exchanger approach temperatures. Temperature profiles through the system heat exchangers illustrating approach temperature in terms of fluid inlet and outlet temperatures are shown in Figure 6. The schematics and plots shown depict counterflow heat exchanger configurations. This arrangement was used for illustration purposes only. In practice, the heat exchangers will generally be of a cross-counterflow design. However, the approach temperature remains a major design factor in determining the size of heat exchangers of any flow configuration.

The system characteristics obtained by computer analysis were plotted in terms of the significant design parameters and are discussed below. The conditions listed in Table 2 were used for purposes of comparison. Data presented later show the sensitivity of the system to water temperature at boiler inlet as high as 300 F.

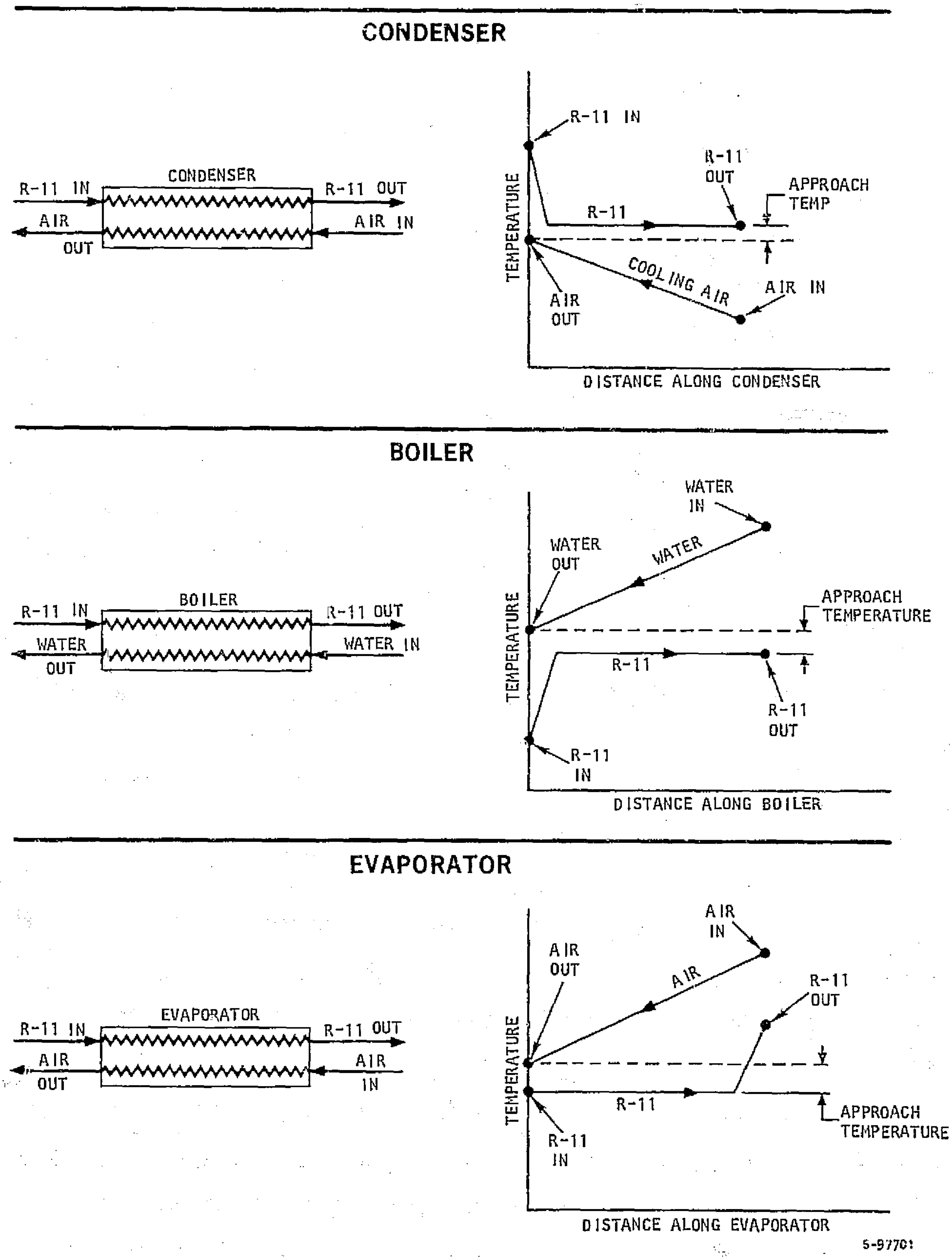


Figure 6. Typical Temperature Profiles

TABLE 2

BASELINE DESIGN CONDITIONS

Parameter	Design Condition
Water inlet temperature to the boiler	366.5 K (200 F)
Ambient air drybulb temperature	308.2 K (95 F)
Ambient air wetbulb temperature	297 K (75 F)
Room return air drybulb temperature	299.8 K (80 F)
Evaporating temperature	280.4 K (45 F)
Room return air wetbulb temperature	292.6 K (67 F)
System capacity	10.5 kw (3 tons)

Concept A, Ambient Air Condenser

The schematic of Concept A is presented in Figure 7 below. Parametric performance and cost data are shown in Figure 8.

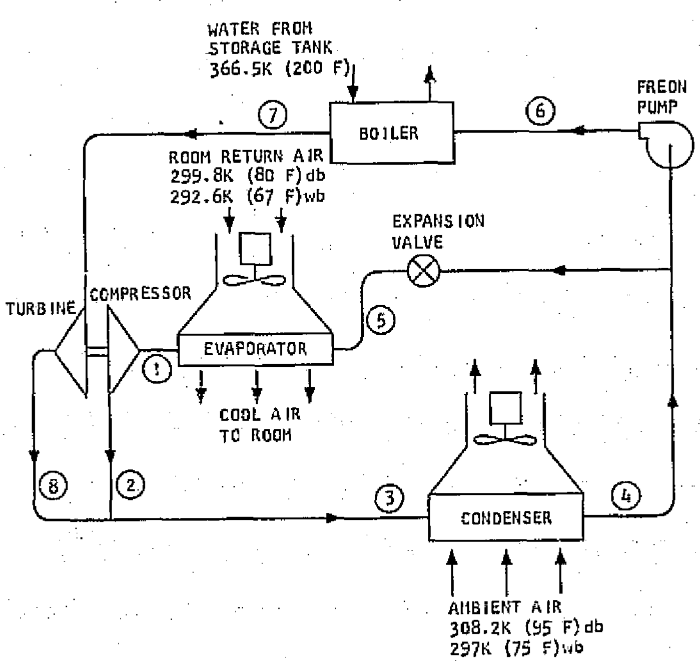


Figure 7. Concept A, Ambient Air Condenser

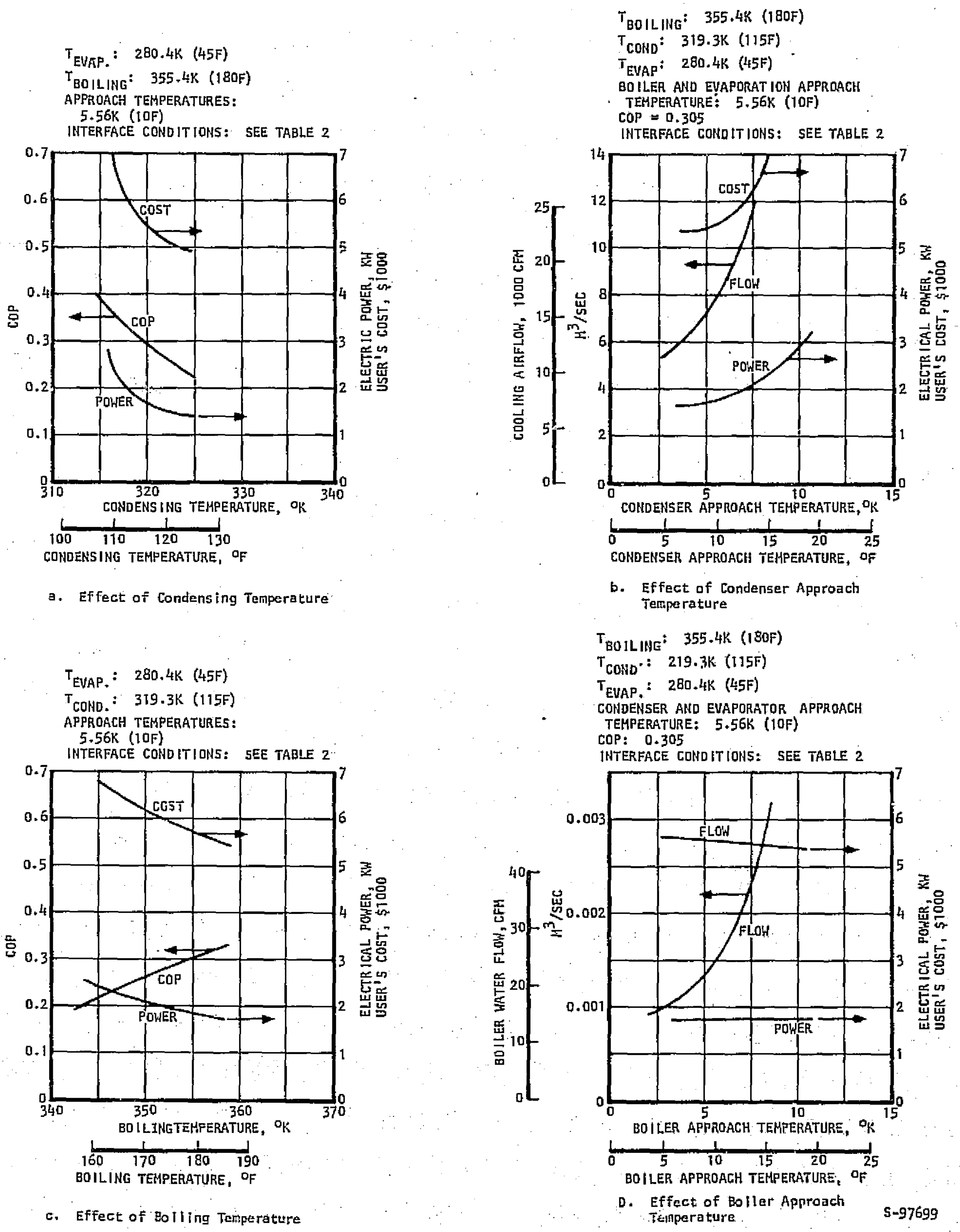


Figure 8. Parametric Data for Concept A

1. Effect of Condensing Temperature (Figure 8a)

The condenser and its fan constitute the most sensitive equipment in terms of system cost, size, and electrical power. While the system COP increases at low condensing temperature, resulting in lower boiler and condenser heat loads, the lower ΔT (condensing temperature - air temperature) potential for heat transfer with a fixed ambient air heat sink overshadows this effect. As a result, both system power (with condenser fan as the main contributor) and system cost increase rapidly as the design condensing temperature drops below about 320 K (115 F). At that condensing temperature and for the conditions listed on the figure, the overall system COP will be about 0.3.

The sensitivities of these parameters at a condensing temperature of 319 K (115 F) calculated in terms of condensing temperature are as follows:

COP sensitivity:	-0.017/K (-0.0092/F)
Power sensitivity:	-0.12 kw/K (-0.065 kw/F)
Cost sensitivity:	-\$190/K (-\$104/F)

An overall system-level trade can be performed involving system COP and cost because both COP and cost increase as condensing temperature drops. However, at 319 K (115 F), a 0.1 improvement in system COP will cost \$1120; this represents a prohibitive cost increase for higher performance. A condensing temperature of 319 K (115 F) appears about minimum for this type of system designed for the conditions listed in Figure 8a.

2. Effect of Condenser Approach Temperature (Figure 8b)

The condenser cooling airflow $\dot{w}$ can be approximated by

$$\dot{w} = \frac{Q}{Cp\ \Delta T_{air}} = \frac{Q}{cp} \times \frac{1}{(T_{cond} - T_{air\ in}) - \Delta T_{approach}}$$

where Q is the condenser heat load, cp is the specific heat of the cooling air, and T_{cond} is the condensing temperature. Reference is made to Figure 6 for definition of the approach temperature, ΔT approach, in terms of the fluid temperatures at inlet and outlet of the condenser. As shown by this correlation and plotted in Figure 8b, the cooling airflow through the condenser increases rapidly with increasing air temperature for fixed values of the condensing and ambient air temperatures. This effect overshadows the low condenser UA requirements and smaller condenser size because the thermal design requirements are relaxed at higher approach temperature. For example, the condenser required at 4.2 K (7.5 F) and 8.3 K (15 F) approach temperatures are estimated as 50 kJ/sec m^2K (8810 Btu/hr ft^2 F) and 68.2 kJ/sec m^2 K (12,000 Btu/hr ft^2 F), respectively. As a result, system cost and electrical power requirements will increase with the condenser design approach temperature. In terms of system cost and electrical power requirements (and also detail design of the condensing heat exchanger), an approach temperature of 5.6 K (10 F) appears to be a reasonable compromise for the operating conditions noted in Figure 8b.

3. Effect of Boiling Temperature (Figure 8c)

The cost and power dependancy on boiling temperatures are relatively mild by comparison to condensing temperature. In this case, the major effect is the most favorable system operating conditions (higher COP) obtained at higher boiler temperature. As the boiling temperature increases from 344 K (160 F) to 355 K (180 F), the quantity of heat processed at the condenser decreases from 60.4 kw (206,300 Btu/hr) to 44.9 kw (153,100 Btu/hr) for a 10.5-kw (3-ton) capacity air conditioner. This effect is reflected in the plot of Figure 8c.

The sensitivity of the system characteristics to boiling temperature (around 355 K (180 F)) are:

COP sensitivity:	+0.008/K (0.0044/F)
Power sensitivity:	-0.035 kw/K (-0.019 kw/F)
Cost sensitivity:	-$60.4/K (-$33.6/F)

These data show that the COP sensitivity due to boiling temperature is about one-half of that due to condensing temperature, while power and cost are only one-third as sensitive.

The plot of Figure 8c shows that high boiling temperature is highly desirable. Boiler design considerations, however, limit the boiling temperature to about 258.2 K (185 F) with a water inlet temperature of 366.5 K (200 F).

4. Effect of Boiler Approach Temperature (Figure 8d)

The boiler anproach temperature has only a negligible effect on system cost and electrical power requirements. The only significant effects are system level considerations such as water flow rate and thermal energy storage tank thermal management. For a fixed water temperature at boiler inlet of 366.5K (200 F), a lower approach temperature will reduce the hot water flow considerably and enhance thermal energy utilization if adequate stratification is provided in the water storage tank design. This presents significant system operational advantages because water at a high temperature level will be available for a longer period.

With a fully mixed tank, the only advantage of a lower approach temperature is the lower pump flow required. However, a higher boiler effectiveness will be required. Detail design studies of the boiler will be necessary in final selection of the boiler approach temperature. For the conditions noted near the selected design point, an approach of 4.2 K (7.5 F) appears acceptable in view of the high heat transfer coefficients achievable on both sides (water and R-11 side) of this unit.

5. Effect of Evaporator Approach Temperature (Figure 8e)

The evaporating temperature was fixed at 280.4 K (45 F) for all system concepts considered. This represents a maximum value to provide for latent heat removal and control of humidity within the air conditioned space. Data similar to that presented previously are shown in Figure 8e. As shown, the evaporator approach temperature has only a relatively small effect on overall system cost and power requirement. The only limiting factor is the evaporator airflow rate, which exceeds the ARI specification of 0.054 (m^3/sec)/kw (400 cfm/ton) at approach temperatures higher than 9 K (16 F). An evaporator approach temperature of 5.6 K (10 F) appears reasonable in view of the thermal design of the evaporator and the ARI flow limitations.

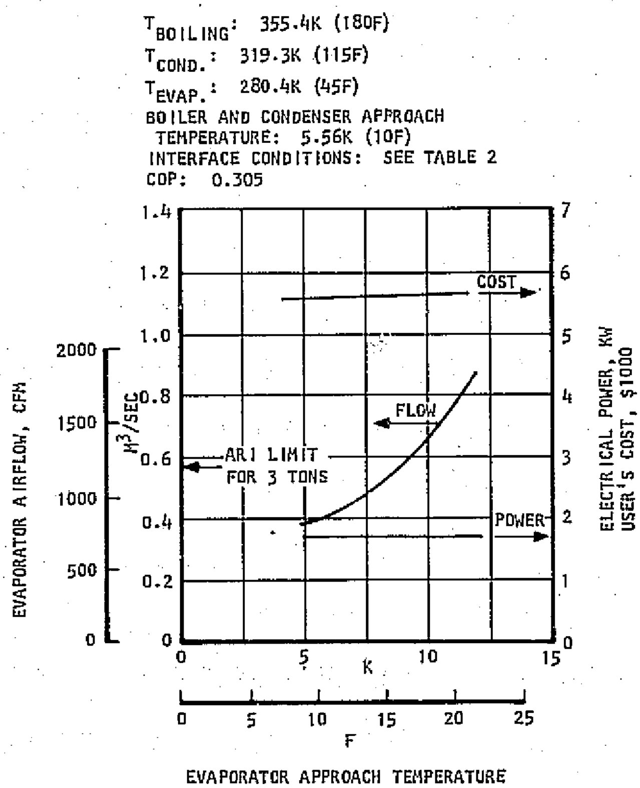


Figure 8e. Effect of Evaporator Approach Temperature

6. System and Component Characteristics

Previous discussions have been concerned with the characteristics of the system as a function of cycle parameters and heat exchanger thermal design. As a result of these discussions, a design point was selected for Concept A equipment corresponding to the interface data of Table 2. The characteristics of this system and its components are listed in Table 3.

Concept B, Humidified Ambient Air Condenser (Figure 9)

Examination of the computer data for Concept B shows a dependence of system characteristics on boiling temperature similar to that for Concept A. Furthermore, the boiler and evaporator approach temperatures have only a mild effect on overall air conditioner cost and power requirements at conditions near the selected design point for this concept. The major parameter affecting cost, power, and COP is again the condensing temperature and the condenser approach temperature. The relationships between these factors are shown in Figure 10.

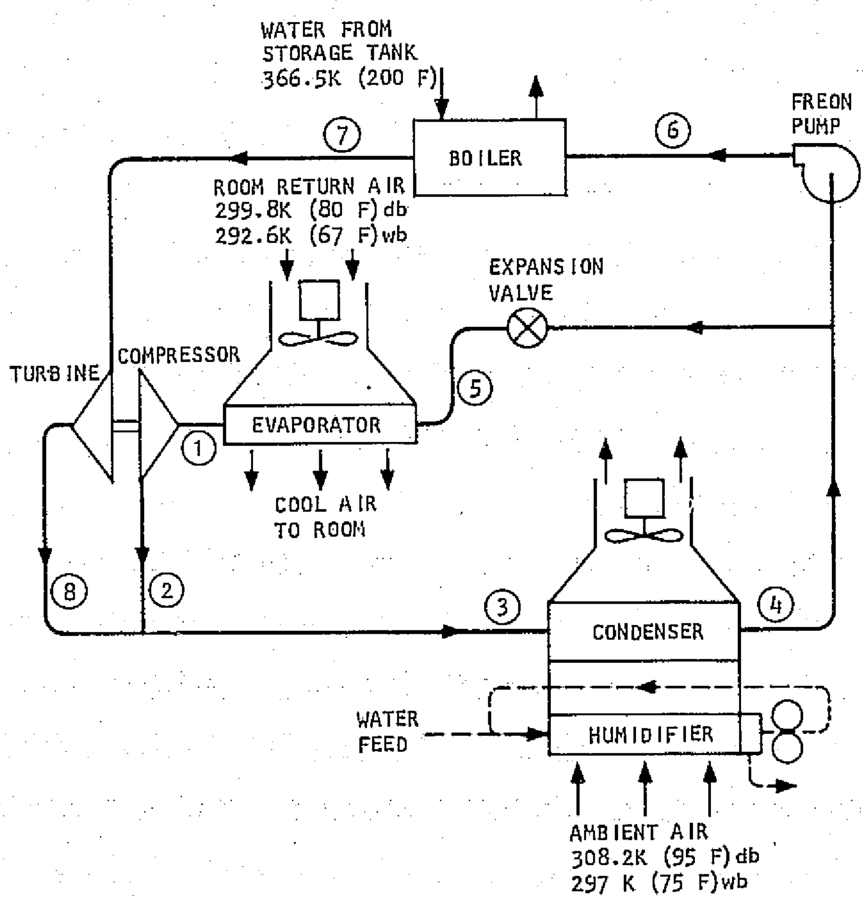


Figure 9. Concept B, Humidified Ambient Air Condenser

TABLE 3

SYSTEM AND COMPONENT CHARACTERISTICS FOR CONCEPT A

Design Conditions

Capacity: 10.55 kw (3 tons)

Hot water supply temperature: 366.5 K (200 F)

Ambient temperatures: 308.2 K (95 F) db, 297 K (75 F) wb

Conditioned air return temperatures: 299.8K (80 F) db, 292.6 K (67 F) wb

Overall System Parameters

COP: 0.519

Electrical power requirements: 1.46 kw

User's cost: $3970

Cycle Data

Boiling temperature: 358.2 K (185 F)

Condensing temperature: 310.9 K (100 F)

Evaporating temperature: 280.4 K (45 F)

Power loop efficiency: 10 percent

Refrigeration loop COP: 5.75

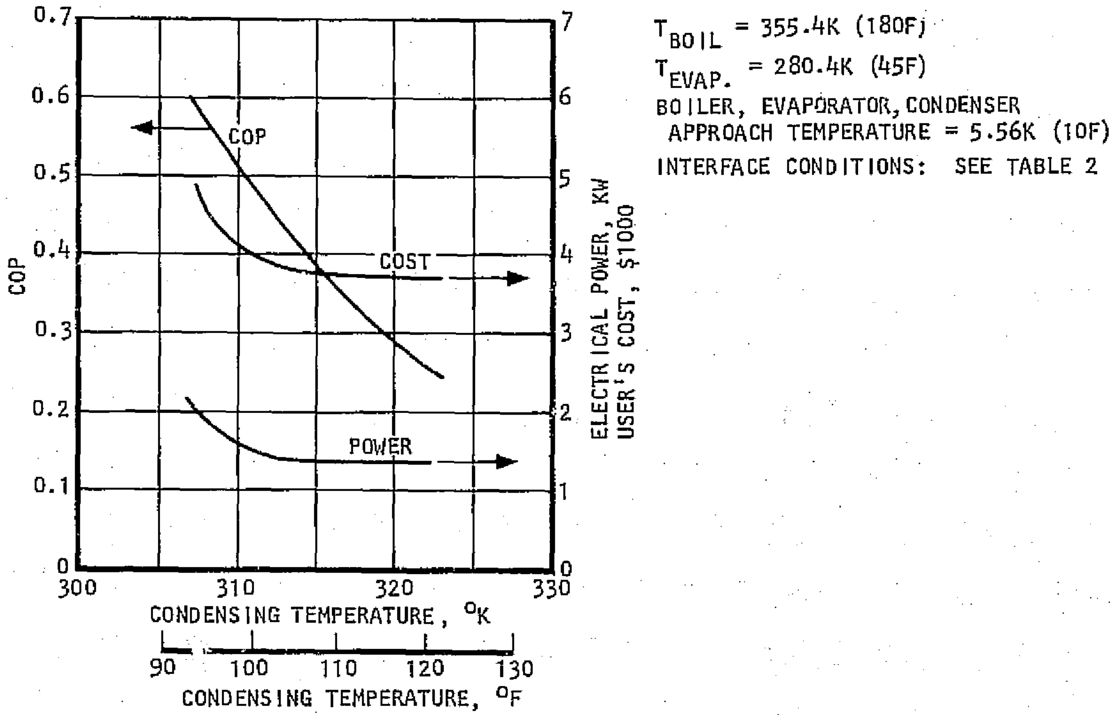
Overall COP: 0.519

Equipment Data

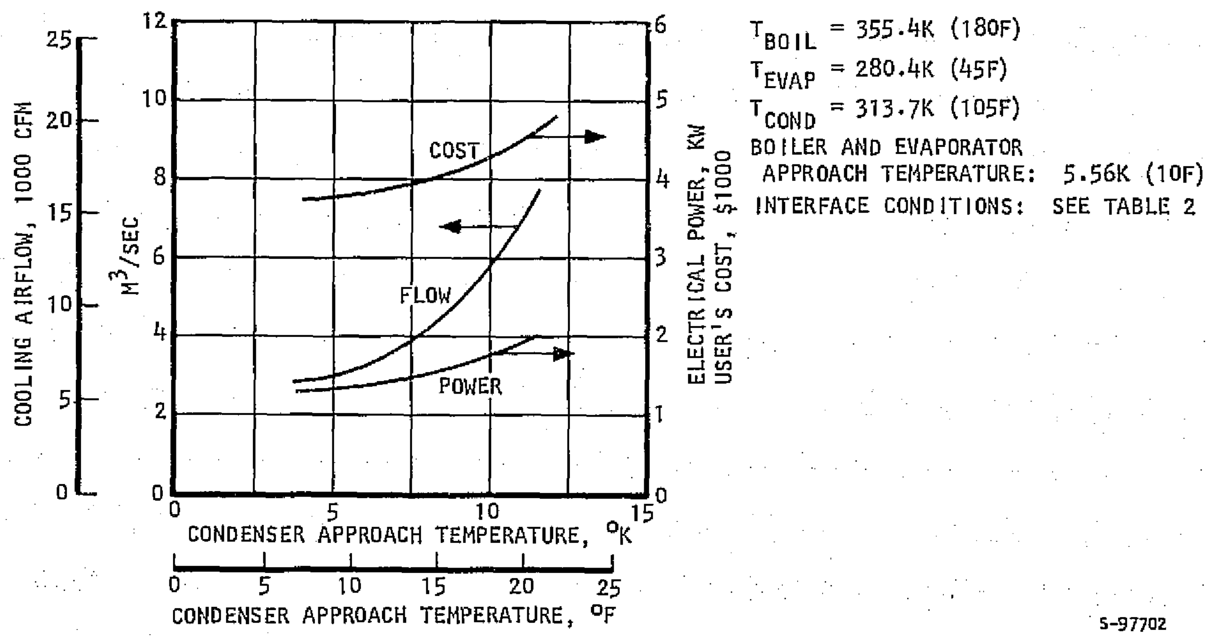
1. Heat Exchangers	Boiler	Condenser	Evaporator	Humidifier
Heat load, kw(Btu/hr)	20.3 (69,130)	31.0 (104,600)	10.55 (63,000)	38.2 (130,500)
UA, kw/m^2K (Btu/hr ft^2 F)	36.2 (6390)	38.1 (6705)	-	-
Cold fluid	R-11	Humidified air	R-11	Water
inlet temperature, K(F)	311.7 (101.3)	298.2 (77)	280.4 (45)	294.3 (70)
Outlet temperature, K(F)	358.2 (185)	305.4 (90)	280.4 (45)	-
Flow rate, kg/sec(lb/hr)	0.102 (808)	-	0.066 (520)	.015 (122)
m^3/sec (cfm)	-	3.67 (7780)	-	-
m^3/sec (gpm)	-	-	-	-
Hot fluid	Water	R-11	Return air	Ambient air
Inlet temperature, K(F)	366.5 (200)	332.1 (120.1)	299.8 (80) db 292.6 K (67) wb	308.2 (95) db, 297 (75) wb
Outlet temperature, K(F)	362.3 (192.5)	310.9 (100)	285.9 (55) db, 285 (53.4) wb	298.2 (77) db, 297 (75) wb
Flow rate, kg/sec (lb/hr)	-	0.168 (1328)	-	-
m^3/sec (cfm)	-	-	0.40 (850)	3.67 (7780)
m^3/sec (gpm)	0.0012 (18.4)	-	-	-

2. Turbomachines	Turbine	Compressor	
Flow, kg/sec (lb/hr)	0.102 (808)	0.066 (520)	
Inlet pressure, k N/m^2 (psia)	592.9 (86)	55.16 (8)	
Pressure ratio	3.47	3.1	
Diameter, cm (in)	4.93 (1.94)	5.69 (2.24)	
Speed, rpm	61,716	61,716	
Efficiency, %	78.9	72.5	

3. Blowers and Pumps	Condenser Blower	Evaporator Blower	Freon Pump	Water Pump
Flow, kg/sec (lb/hr)	-	-	0.102 (808)	0.152 (1200)
m^3/sec (cfm)	3.67 (7780)	0.40 (850)	-	-
Inlet pressur, k N/m^2 (psia)	101.3 (14.7	101.3 (14.7	162.6 (23.6)	101.3 (14.7)
Pressure rise, N/m^2 (in. H_2O)	147 (0.59)	214 (0.86)	-	-
Pressure ratio	-	-	3.83	2
Electrical power, kw	1.11	0.180	0.064	0.105



a. Effect of Condensing Temperature



b. Effect of Condenser Approach Temperature

Figure 10. Parametric Data for Concept B

1. Effect of Condensing Temperature

In this case, the ambient air drybulb temperature is reduced by about 10 K (18 F) from 308.2 K (95 F) by adiabatic humidification upstream of the condenser. As a result, significantly lower condensing temperatures can be achieved without significant cost and power penalties. As shown in Figure 10a, condensing temperatures of 310.9 K (100 F) can be used for design, corresponding to a COP of 0.48. This represents a 50 percent increase over Concept A. Concept B also results in significant cost savings (about $2000 for a 10.5-kw (3-ton) unit) and a 20 percent savings in fan/pump power requirements.

As the condensing temperature drops below 310.9 K (100 F), both system cost-to-the-user and system power increase rapidly, although the performance of the system improves. Again, in this case this effect is due to the much higher condenser airflow rates necessary at lower condensing temperatures. The cost increase associated with the higher COP at a condensing temperature of 310.9 K (100 F) is estimated at $780 for 0.1 increase in COP. Tentatively, a condensing temperature of 310.9 K (100 F) is selected for this approach.

2. Effect of Condenser Approach Temperature

This effect is depicted in the plot of Figure 10b for conditions representative of system design point. Again, the condenser airflow required increases rapidly with approach temperature above a ΔT approach of 5.6 K (10 F). The higher heat exchanger effectiveness and weight required at low approach temperature is more than offset by the much lower cooling airflows necessary. There are no apparent problems in designing the condenser for a 5.6 K (10 F) approach.

3. System and Component Characteristics

Table 4 summarizes the characteristics of the system (and its components) identified as Concept B and shown schematically in Figure 9. Design point conditions are listed in the table. Boiling temperature is taken as 358.2 K (185 F) for a boiler approach of 4.2 K (7.5 F); evaporating temperature is 280.4 K (45 F) for an evaporator approach of 5.6 K (10 F). The condensing temperature is 310.9K (100 F) for an approach of 5.6 K (10 F). Under these conditions, system COP is 0.480.

Concept C, Evaporative Condenser (Figure 11)

A schematic of this arrangement is shown in Figure 11. Water is sprayed on the tubes of the condenser where it evaporates. The vapor formed is entrained by the airstream through the unit. The air is only used as a means for evaporating water. At inlet, near-adiabatic saturation will reduce the drybulb temperature of the air to about 298.2 K (77 F). As the air drybulb temperature increases through the unit, its capacity for water vapor increases rapidly. At outlet, the drybulb temperature of the air exiting the condenser is lower than at inlet.

TABLE 4

SYSTEM AND COMPONENT CHARACTERISTICS FOR CONCEPT B

Design Conditions

Capacity: 10.55 kw (3 tons)

Hot water supply temperature: 366.5 K (200 F)

Ambient temperatures: 308.2 K (95 F) db, 297 K (75 F) wb

Conditioned air return temperatures: 299.8 K (80 F) db, 292.6 K (67 F) wb

Overall System Parameters

COP: 0.326

Electrical power requirements: 1.72 kw

User's cost: $5630

Cycle Data

Boiler temperature: 358.2 K (185 F)

Condenser temperature: 319.3 K (115 F)

Evaporator temperature: 280.4 K (45 F)

Power loop efficiency: 8.5%

Refrigeration loop COP: 4.27

Overall COP: 0.326

Equipment Data

1. Heat exchangers	Boiler	Condenser	Evaporator
Heat load, kw (Btu/hr)	32.3 (110,100)	42.6 (145,500)	10.55 (36,000)
UA, kw/m^2K (Btu/hr ft^2 F)	57.9 (10,200)	57.3 (10,100)	-
Cold fluid	R-11	Ambient air	R-11
Inlet temperature, K(F)	319.9 (116.2)	308.2 (95)	280.4 (45)
Outlet temperature, K(F)	358.2 (185)	313.7 (105)	280.4 (45)
Flow rate, kg/sec (lb/hr)	0.169 (1337)	-	0.069 (545)
m^3/sec (cfm)	-	6.63 (14,050)	-
m^3/sec (gpm)	-	-	-
Hot fluid	Water	R-11	Return air
Inlet temperature, K(F)	366.5 (200)	330.2 (134.6)	299.8(80) db, 292.6 (67) wb
Outlet temperature, K(F)	362.3 (192.5)	319.3 (115)	285.9(55) db, 285 (53.4) wb
Flow rate, kg/sec (lb/hr)	-	0.238 (1882)	-
m^3/sec (cfm)	-	-	0.4 (850)
m^3/sec (gpm)	0.0019 (29.3)	-	-

2. Turbomachines	Turbine	Compressor
Flow, kg/sec (lb/hr)	0.169 (1337)	0.069 (545)
Inlet pressure, kN/m^2 (psia)	592.9 (86.0)	55.16 (8.0)
Pressure ratio	2.66	4.04
Diameter, cm (in.)	4.63 (1.82)	6.70 (2.64)
Speed, rpm	58,720	58,720
Efficiency, percent	0.81	0.71

3. Blowers and Pumps	Evaporator Blower	Condenser Blower	Freon Pump
Flow, kg/sec (lb/hr)	-	-	0.169 (1337)
m^3/sec (cfm)	0.4 (848)	6.63 (14,050)	-
Inlet pressure, kN/m^2 (psia)	101.3 (14.7)	101.3 (14.7)	212.4 (30.8)
Pressure rise, N/m^2 (in.H_2O)	214 (0.86)	102 (0.41)	-
Pressure ratio	-	-	2.93
Electrical power, kw	0.18	1.45	0.1

In Concept B, the water-carrying capacity of the condenser airstream is limited by the wetbulb temperature of the air at inlet 297 K (75 F), and condenser cooling is affected essentially by sensible heat transfer. In Concept C, the water capacity of the air increases through the heat exchanger as the drybulb temperature of the air increases. Also, much higher heat transfer coefficients can be achieved by water evaporation on the surfaces of the condenser tubes.

Parametric data generated by computer indicate that the boiler and evaporator approach temperatures have only minor effects on the overall system cost and electrical power requirements. As before, these parameters were selected as 4.2 K (7.5 F) and 5.6 K (10 F), respectively. For maximum COP, a boiling temperature of 358.2 K (185 F) was selected.

1. Effect of Condensing Temperature

Figure 12a shows system cost and power requirements as a function of condensing temperature for two values of the approach temperature, 5.6 K (10 F) and 8.3 K (15 F). At condensing temperatures above about 314 K (105 F), cost and power are about the same for the two values of the approach temperature. At lower condensing temperatures, the lower approach yields preferable characteristics. Further, in order to achieve a condensing temperature as low as 305.4 K (90 F) with attendant high COP (0.66), the lower approach is necessary.

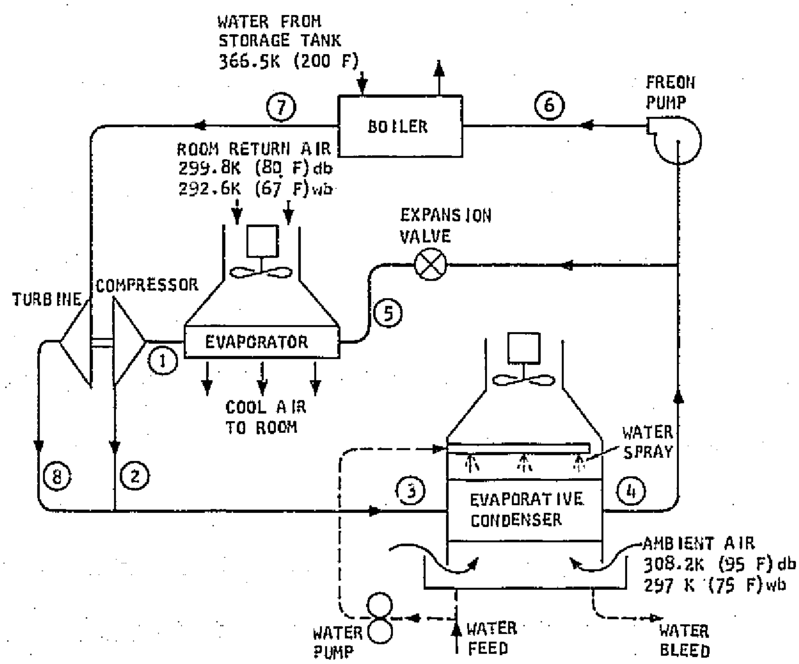
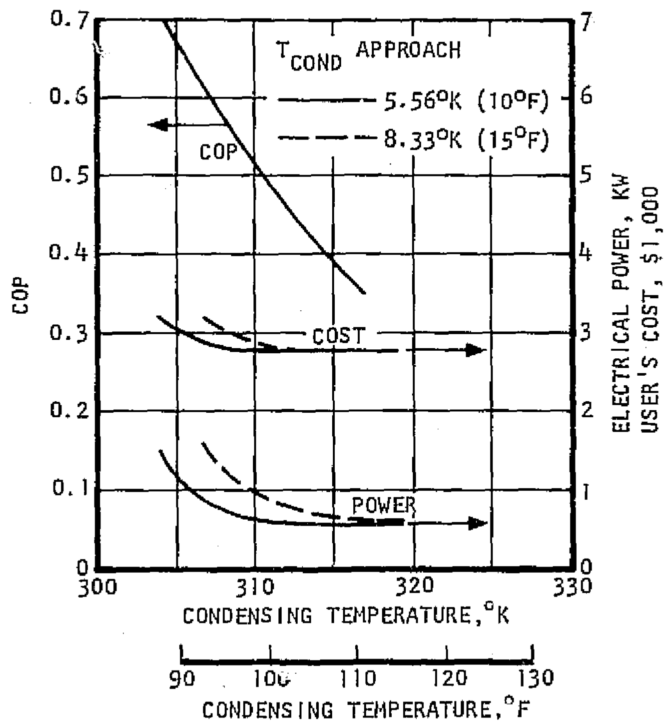
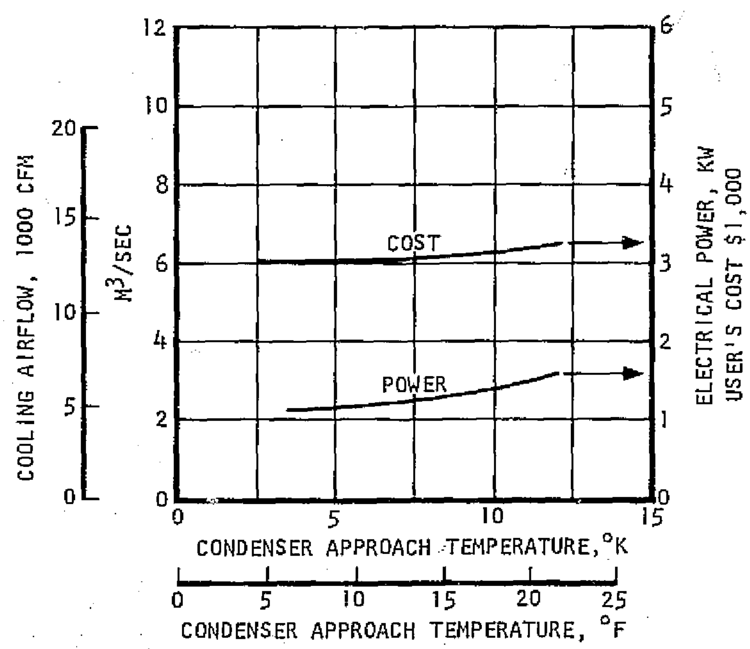


Figure 11. Concept C, Evaporative Condenser



$T_{BOILING}$ = 355.4K (180F)
T_{EVAP} = 280.4K (45F)
BOILER AND EVAPORATOR APPROACH TEMPERATURE: 5.56K (10F)
CONDENSER APPROACH TEMPERATURE: (SEE NOTE ON FIGURE)
INTERFACE CONDITIONS: SEE TABLE 2

a. Effect of Condensing Temperature



$T_{BOILING}$ = 355.4K (180F)
T_{EVAP} = 280.4K (45F)
T_{COND} = 305.4K (90F)
BOILER AND EVAPORATOR APPROACH TEMPERATURE = 5.56K (10F)
INTERFACE CONDITIONS: SEE TABLE 2

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b. Effect of Condenser Approach Temperature

Figure 12. Parametric Data for Concept C

2. Effect of Condenser Approach Temperature

In this case, the approach temperature is defined as the difference between the condensing temperature and the wetbulb temperature of the air at condenser outlet. As the approach temperature increases, both cost and power requirement increase since more air is required to carry the water load. The optimum occurs at an approach temperature of about 5.6 K (10 F) with a condensing temperature of 305.4 K (90 F). Thus, with a condenser temperature of 305.4 K (90 F), the temperature of the air at condenser outlet will be 299.8 K (80 F) -- wetbulb and drybulb. The system cost and power are considerably lower than either Concepts A or B over the range investigated.

3. System and Component Characteristics

Table 5 lists the significant system characteristics and component performance for the design of the system corresponding to the schematic of Figure 11. The data are presented for a condensing temperature of 305.4 K (90 F), a boiling temperature of 358.2 K (185 F), and an evaporating temperature of 280.4 K (45 F). System overall COP is 0.691 -- a considerable improvement over the two concepts (A and B) discussed previously.

Concept D, Water Condenser/Cooling Tower (Figure 13)

This arrangement also has the potential for low condensing temperature through the use of an efficient cooling tower. The cool water from the cooling tower is used as the condenser heat sink so that condensing temperatures on

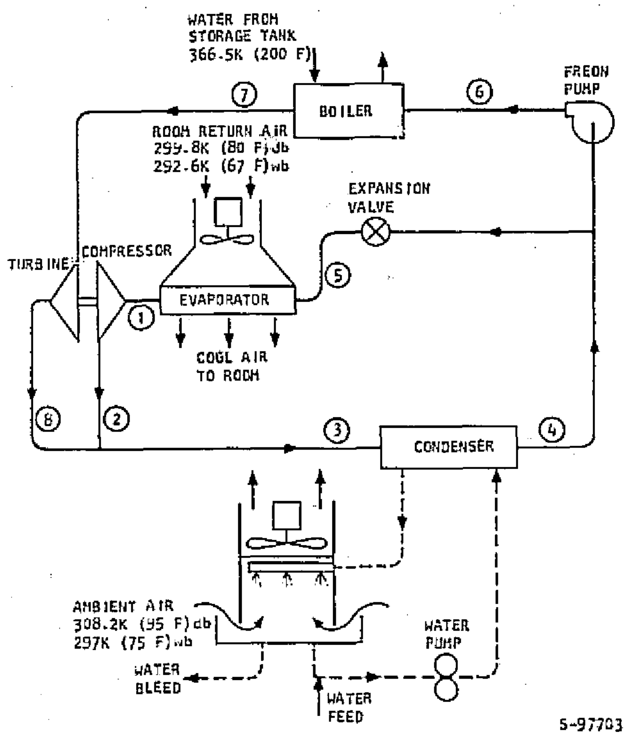


Figure 13. Concept D, Water Condenser/Cooling Tower

TABLE 5

SYSTEM AND COMPONENT SUMMARY FOR CONCEPT C

Design Conditions

Capacity: 10.55 kw (3 tons)

Hot water supply temperature: 366.5 K (200 F)

Ambient temperatures: 308.2 K (95 F) db, 297 K (75 F) wb

Conditioned air return temperatures: 299.8 K (80 F) db, 292.6 K (67 F) wb

Overall System Parameters

COP: 0.691

Electrical power requirements: 1.15 kw

User's cost: $3055

Cycle Data

Boiling temperature: 358.2 K (185 F)

Condensing temperature: 305.4 K (90 F)

Evaporating temperature: 280.4K (45 F)

Power loop efficiency: 0.108

Refrigeration loop COP: 7.12

Overall COP: 0.691

Equipment Data

1. Heat exchangers	Boiler	Condenser	Evaporator
Heat load, kw (Btu/hr)	14.4 (51,950)	24.3 (87,560)	5.56 (36,000)
UA, kw/m^2 K (Btu/hr ft^{2}F)	27.2 (4800	-	-
Cold fluid	R-11	Air and evaporated water	R-11
Inlet temperature, K(F)	3062 (91.4)	308.2 (95) db, 297 (75) wb	280.4 (45)
Outlet temperature, K(F)	358.2 (185)	299.8 (80) db, 299.8 (80) wb	280.4 (45)
Flow rate, kg/sec (lb/hr)	0.075 (592)	Water evap.: 0.018 (144.6)	0.064 (504)
m^3/sec (cfm)	-	air: 1.91 (4050)	-
m^3/sec (gpm)	-	-	-
Hot fluid	Water	R-11	Return air
Inlet temperature, K(F)	366.5 (200)	316.7 (110.9)	299.8 (80) db, 292.6 (67) wb
Outlet temperature, K(F)	362.3 (192.5)	305.4 (90)	285.9 (55) db, 285 (53.4) wb
Flow rate, kg/sec (lb/hr)	-	0.139 (1097)	-
m^3/sec (cfm)	-	-	0.4 (850)
m^3/sec (gpm)	0.0009 (13.8)	-	-

2. Turbomachines	Turbine	Compressor
Flow, kg/sec (lb/hr)	0.075 (592)	0.064 (504)
Inlet pressure, kN/m^2 (psia)	592.9 (86)	55.2 (8)
Pressure ratio	4.1	2.62
Diameter, cm (in.)	5.54 (2.18)	5.54 (2.18)
Speed, rpm	58,470	58,440
Efficiency, percent	77.1	73.6

3. Blowers and pumps	Condenser Blower	Evaporator Blower	Freon Pump	Water Pump
Flow, kg/sec (lb/hr)	-	-	0.075(592)	0.127 (1000)
m^3/sec (cfm)	1.91 (4050	0.4 (850)	-	-
Inlet pressure, kN/m^2 (psia)	101.3 (14.7)	101.3 (14.7)	137.8 (20)	(101.3) 14.7
Pressure rise, N/m^2 (in. H_2O)	204 (0.82)	214 (0.86)	-	-
Pressure ratio	-	-	4.52	2
Electrical power, kw	0.830	0.18	0.050	0.088

the order of 305.4 K (90 F) can be achieved. For water-to-air heat pumps, ARI standard 240 (Reference 4) specifies a water temperature at condenser inlet and outlet of 297 K (75 F) and 308.2 K (95 F), respectively. The 297 K (75 F) inlet temperature appears very optimistic in view of the 297 K (75 F) wetbulb temperature of ambient air for air-to-air heat pumps. Further, there appears to be no firm basis for the 308.2 K (95 F) condenser outlet temperature other than limiting the water flow rate and the ΔT through the cooling tower. In this study, the cooling water temperature from the cooling tower was taken as 299.8 K (80 F) because this temperature level is more consistant with an ambient air wetbulb temperature of 297 K (75 F).

As for the three previous concepts, the boiling temperature selected in final evaluation was 358.2 K (185 F) to maximize COP. The approach temperatures at the boiler and evaporator were taken as 4.2 K (7.5 F) and 5.6 K (10 F), respectively. Figure 14 shows parametric data related to the operation and design of the condenser.

1. Effect of Condensing Temperature (Figure 14a)

Both power consumption and system cost increase with condensing temperature primarily because of the lower system effectiveness. In this case, however, the condensing heat exchanger/cooling tower constitute a very efficient heat rejection system; thus, the impact of these components on total system cost and power is not as pronounced as for other competing approaches. The optimum design in terms of condensing temperature occurs at about 305.4 K (90 F). With a water temperature of 299 K (80 F) at condenser inlet, this imposes severe limitations on the approach temperature (as shown in Figure 14b).

2. Effect of Condenser Approach Temperature

The high heat transfer coefficient afforded by the water coolant loop allows the design of a very efficient condenser with a low approach temperature. The 2.8 K (5 F) approach selected may have to be increased as a result of condenser detail design investigations. However, this does not represent an unrealistic design point for this unit.

3. System and Component Characteristics

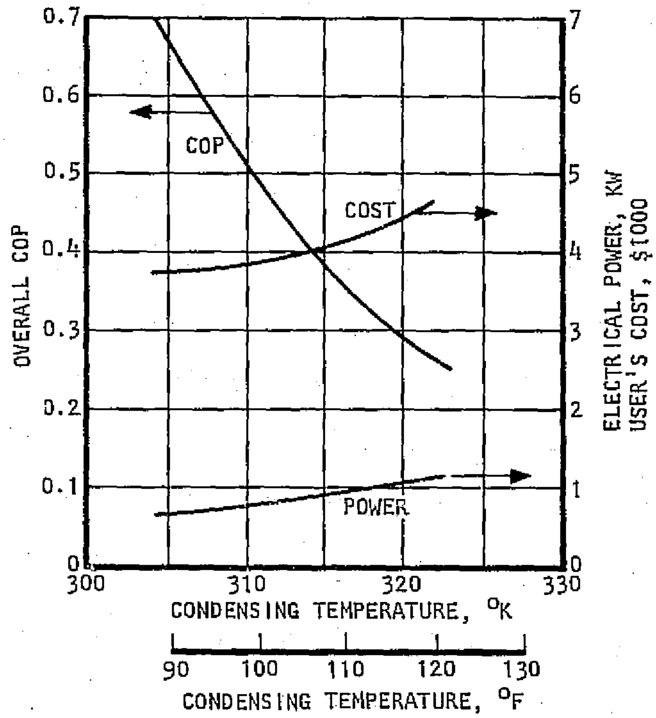
Table 6 summarizes the characteristics of the system and its components at design point. Cycle operating parameters were selected as follows:

Boiling temperature: 358.2 K (185 F)

Condensing temperature: 305.4 K (90 F)

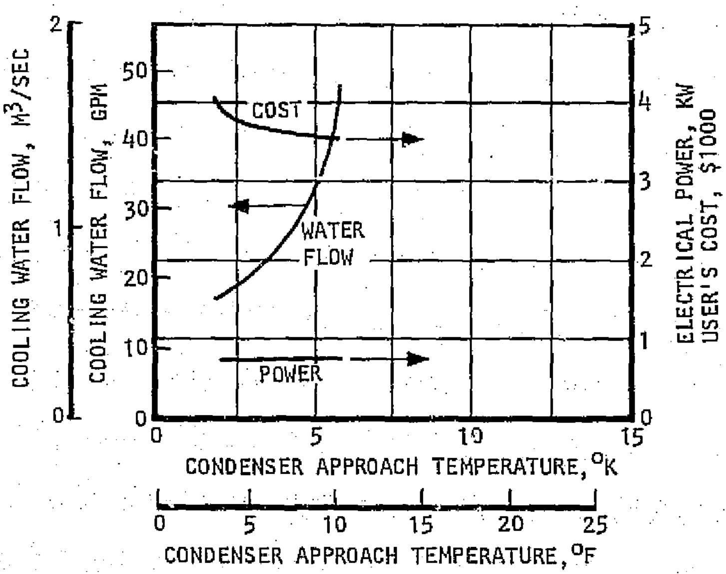
Evaporating temperature: 280.4 K (45 F)

In this case, the very low system power reflects the low pressure drop of the ambient air through the cooling tower. The value used in the computation of cooling tower power was taken as representative of existing cooling tower equipment.



$T_{BOILING}$ = 355.4K (180F)
$T_{EVAP.}$ = 280.4K (45F)
BOILER EVAPORATOR APPROACH
TEMPERATURE = 5.6K (10F)
CONDENSER APPROACH
TEMPERATURE: 2.8K (5F)
INTERFACE CONDITIONS: SEE TABLE 2

a. Effect of Condensing Temperature



$T_{BOILING}$ = 355.4K (180F)
$T_{EVAP.}$ = 280.4K (45F)
$T_{COND.}$ = 305.4K (90F)
BOILER AND EVAPORATOR
APPROACH TEMPERATURE: 5.6K (10F)
INTERFACE CONDITIONS: SEE TABLE 2

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b. Effect of Condenser Approach Temperature

Figure 14. Parametric Data for Concept D

TABLE 6

CONCEPT D SYSTEM AND COMPONENT SUMMARY

Design Conditions

Capacity: 10.55 kw (3 tons)

Hot ater supply temperature: 366.5 K (200 F)

Ambient temperatures: 308.2 K (95 F) db, 297 K (75 F) wb

Conditioned air return temperatures: 299.5 K (50 F) db, 292.6 K (67 F) wb

Water temperature from cooling tower: 299.8 K (80 F)

Overall System Parameters

COP: 0.691

Electrical power requirements: 0.65 kw

User's cost: $3730

Cycle Data

Boiling temperature: 358.2 K (185 F)

Condensing temperature: 305.4 K (90 F)

Evaporating temperature: 280.4 K (45 F)

Power loop efficiency: 10.8 percent

Refrigeration loop COP: 7.12

Overall COP: 0.691

Equipment Data

1. Heat exchangers	Boiler	Condenser	Evaporator
Heat load, kw (Btu/hr)	15.2 (51,950)	25.65 (87,560)	10.55 (36,000)
UA, kw/m^2K (Btu/hr ft^2 F)	27.2 (4800)	68.8 (12,140)	-
Cold fluid	R-11	Cooling tower water	R-11
Inlet temperature, K (F)	306.2 (9.14)	299.8 (80)	280.4 (45)
Outlet temperature, K (F)	358.2 (185)	302.6 (85)	280.4 (45
Flow rate, kg/sec (lb/hr)	.075 (592)	-	0.064 (504)
m^3/sec (cfm)	-	-	-
m^3/sec (gpm)	-	0.0022 (35)	-
Hot fluid	Water	R-11	Return air
Inlet temperature, K (F)	366.5 (200)	316.7 (110.4)	299.8 (80) db, 292.6 (67) wb
Outlet temperature, K (F)	362.3 (192.5)	305.4 (90)	255.9 (55) db, 285 (53.4) wb
Flow rate, kg/sec (lb/hr)	-	0.139 (109.7)	-
m^3/sec (cfm)	-	-	0.4 (850)
m^3/sec (gpm)	.0009 (13.8)	-	-

2. Turbomachines	Turbine	Compressor
Flow, kg/sec (lb/hr)	0.075 (592)	0.064 (504)
Inlet pressure, kw/m^2 (psia)	592.9 (86)	55.8 (8.0)
Pressure ratio	4.1	2.62
Diameter, cm (in.)	5.54 (2.18)	5.54 (2.18)
Speed, rpm	58,440	58,440
Efficiency, percent	77.1	73.6

3. Blowers and Pumps	Cooling Tower Blower	Evaporator Blower	Freon Pump	Water Pump
Flow, kg/sec (lb/hr)	-	-	0.075 (592)	0.127 (1000)
m^3/sec (cfm)	2.61 (5000)			
Inlet pressure, kN/m^2 (psia)	101.3 (14.7)	101.3 (14.7)	137.8 (20)	101.3 (14.7)
Pressure rise, N/m^2 (in.H_2O)	54.7 (.22)	214 (.86)	-	-
Pressure ratio	-	-	4.62	2
Electrical power, kw	0.26	0.18	0.05	0.17

COMPARISON OF APPROACHES

Baseline $LiBr/H_2O$ Absorption System

Data published by Arkla (Reference 5) on the anticipated performance of a $LiBr/H_2O$ absorption air conditioner designed for solar application and featuring an evaporative condenser are summarized in Table 7. The installed price of such a system (including fans) is estimated at about $2500 to $3000 (private communication from Arkla Industries distributor).

The $LiBr/H_2O$ air conditioner has been widely used in conjunction with solar systems and is generally acceptable as the baseline air conditioner. The four Rankine-cycle air conditioner concepts discussed in this report were compared to the $LiBr/H_2O$ system to determine advantages in terms of overall system parameters. The characteristics of the Rankine systems investigated are summarized in Table 8.

Concept Evaluation

Concept A, featuring an ambient air dry condenser, yields excessive system costs and COP's. The high condensing temperatures (319.3 K (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 high-condenser heat loads, high condenser effectiveness, and very high airflows, the cost of the system is prohibitive.

In Concept B, a humidifier upstream of the condenser reduces the drybulb temperature of the ambient air and thus provides an effectively lower temperature heat sink. With this approach, condensing temperatures of 310.9 K (100 F) can be achieved without excessively penalyzing the system. This reduction in condensing temperature improves cycle COP significantly by comparison to Concept A (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. By comparison to the Arkla $LiBr/H_2O$ system, Concept B is not competitive; COP is considerably lower (0.52 vs 0.65); installed cost is higher ($4000 vs $2700); and auxiliary electrical power (for fans and pumps) also is higher (1.5 kw vs 0.88 kw). Therefore, Concept B is rejected on the basis of all three evaluation criteria.

Concept C features an evaporative condenser where water is evaporated to the ambient airstream from the outer surface of the condenser tanks. As shown in Table 8, the installed cost of this system is comparable to that of the Arkla $LiBr/H_2O$ used as a baseline. The condensing temperature can be reduced to 305.4 K (90 F) with an ambient wetbulb temperature corresponding to ARI conditions (297 K (75 F)). The COP of Concept C is 0.69, which is slightly higher than that of the Arkla unit (0.65). The power requirement is estimated at 1.15 kw, which is slightly higher than that of the Arkla unit. This concept represents a significant improvement over Concepts A and B and is considered competitive with the Arkla system. Off-design performance of Concept B must be determined to fully evaluate the advantages of this approach by comparison to the Arkla unit.

Concept D incorporates a cooling tower that provides the cold water used as the air conditioner heat sink. The performance of this concept as expressed by COP is the same as for Concept C, 0.69. The installed cost is higher than Concept C primarily because of the added use of the cooling tower; power requirement, however, is substantially lower.

TABLE 7

ESTIMATED PERFORMANCE OF WATER-FIRED ABSORPTION AIR CONDITIONER

Cooling capacity	10.54 kw (3 tons)
Hot water source temperature	363.7 K in/358.2 K out (195 F in/185 F out)
Chilled water temperature	285.9 K in/280.4 K out (55 F in/45 F out)
Evaporative heat rejection	298.7 K (78 F wb air in)
Water consumption	25.2 μm^3/sec (24 gal/hr)
Coefficient of performance	0.65
Electrical consumption	875 watts maximum

TABLE 8

COMPARISON OF APPROACHES

Capacity: 10.55 kw (3 tons)
Hot water supply temperatures: 366.5 K (200 F)
Ambient air temperatures: 308.2 K (95 F)db, 297 K (75 F)wb
Conditioned air temperatures: 299.8 K (80 F)db, 292.6 K (67 F)wb

Parameter	Dry Condenser (Concept A)	Humidifier (Concept B)	Evaporative Condenser (Concept C)	Cooling Tower (Concept D)
Condensing temperature, K(F)	319.3 (115)	310.9 (100)	305.4 (90)	305.4 (90)
Condenser (cooling tower) airflow, m^3/sec, (cfm)	6.63 (14,050)	3.67 (7780)	1.91 (4050)	2.61 (5000)
Boiler water flow, m^3/sec (gpm)	0.0019 (29.3)	0.0012 (18.4)	0.0009 (13.8)	0.0009 (13.8)
Evaporator airflow, m^3/sec (cfm)	0.4 (850)	0.4 (850)	0.4 (850)	0.4 (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

Detailed investigations of Concept D are warranted to verify by detailed analysis the system cost obtained using the cost model developed earlier in this program.

Operation at Higher Boiler Temperature

While the COP of the absorption system remains about the same over a wide range of heat source temperatures, the Rankine-powered air conditioner performance increases significantly at higher boiler temperature. The data of Figure 15 show this effect. COP's as high as 1.0 can be obtained at a water temperature at boiler inlet of 422 K (300 F). This temperature level could be obtained with a low-performance concentrating collector without sun tracking features. The high COP, relatively low cost, and reasonable power requirements seem to warrant further investigations at temperature levels higher than considered under the present contract.

SYSTEM ARRANGEMENT: CONCEPT C (SEE FIGURE 11)
SYSTEM CAPACITY: 10.5 kw (3 TONS)
CONDENSING TEMPERATURE: 305.4K (90F)
EVAPORATING TEMPERATURE: 280.4K (45F)
BOILING TEMPERATURE: T_{WATER} IN - 8.3K (15F)
EVAPORATOR AND CONDENSER APPROACH TEMPERATURE: 5.6K(10F)

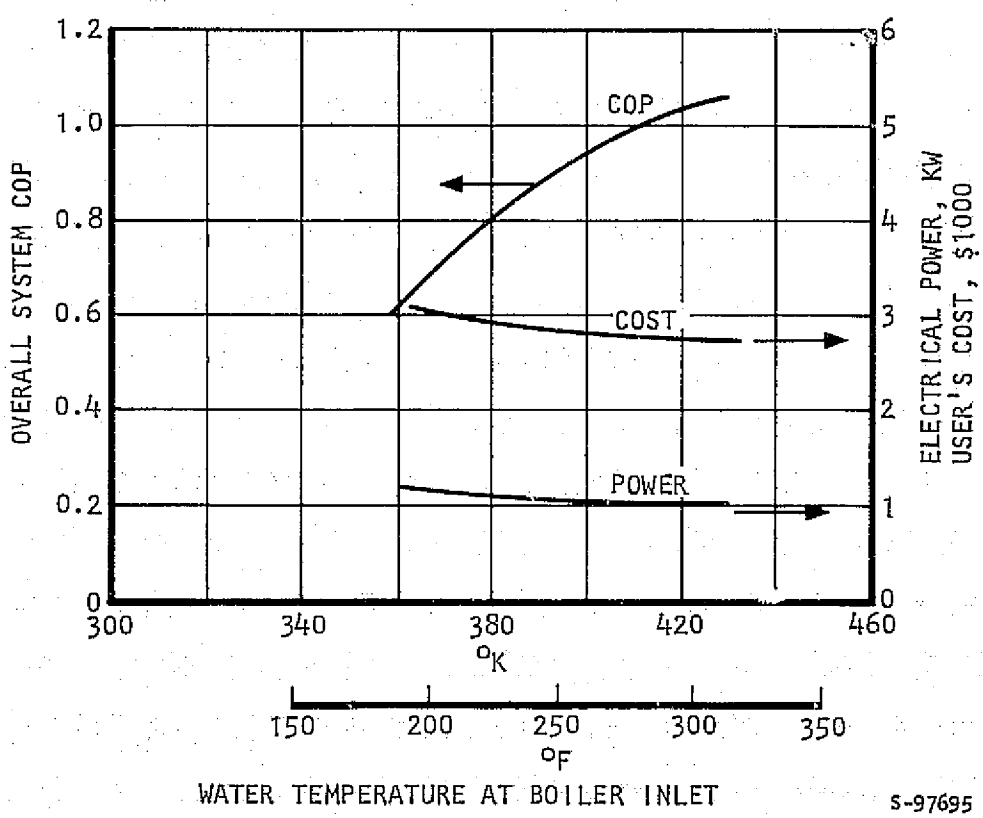

Figure 15. Concept C Characteristics at Higher Boiler Temperature

CONCLUSIONS

The investigations conducted have shown the following:

(a) COP's as high as 0.69 can be obtained with a Rankine-powered system using water evaporation to enhance condenser performance.

(b) Detailed studies are required to determine the desirability of an evaporative-type condenser by comparison to the use of a cooling tower.

(c) On the basis of COP, cost, and electric power usage, the Rankine-powered system is comparable to state-of-art $LiBr/H_2O$ absorption systems.

(d) Off-design performance analyses are necessary to fully assess the relative merits of the Rankine-powered and absorption systems.

(e) The present investigations of Rankine-powered systems should be extended to include higher thermal source temperatures as attainable from semi-concentrator type solar collectors. At source temperatures of 422 K (300 F), COP's higher than 1.0 can be achieved; this compares very favorably to the $LiBr/H_2O$ absorption system.

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3. Economic Analysis, Development of a Solar-Powered Air Conditioner, Contract NAS8-30758, AiResearch Report 74-10996(4), March 1975.

4. ARI Standard 240, Standard for Unitary Heat Pump Equipment, Air Conditioning and Refrigeration Institute publication, 1967.

5. Solar Optimized Absorption Cooling Unit, Arkla Proposal to NSF, November 1973.

APPENDIX A

COMPUTER PROGRAM NOMENCLATURE AND LISTING

This appendix contains a definition of the computer program input nomenclature and the listing of the program. The program was written in Fortran language for use with the Univac 1108 computer. The nomenclature is given in Table A-1, which defines all input data required for execution of 'RANKIN' as contained in the NAMELIST 'INPUT'. The data are presented in the same order as they appear in the computer program input data list. The computer program listing is presented in Figure A-1.

TABLE A-1

INPUT DATA NOMENCLATURE FOR 'RANKIN'

VIST	Viscosity of refrigerant at 15 tabulated temperatures TT, centipoise
TT	15 temperatures at which viscosity VIST is given, °F
TTH	17 temperatures at which following saturated liquid and vapor properties are given, °F
HVT	Enthalpy of saturated vapor at temperatures TTH, Btu/lb
HLT	Enthalpy of saturated liquid at temperatures TTH, Btu/lb
PT	Saturation pressure at temperatures TTH, psia
RHOVT	Density of saturated vapor at temperatures TTH, lb/(cu ft)
CP	Specific heat of vapor at constant pressure, Btu/(°F)(lb)
GAMMA	Specific heat ratio of vapor
AK	Ratio of sonic velocity to square root of absolute temperature, ft/(sec) ($\sqrt{°R}$)
MW	Molecular weight of refrigerant
DPP	HX pressure drop expressed as a fraction of inlet pressure
EFM	Mechanical efficiency of turbocompressor shaft, in fraction
QR	Refrigeration load, Btu/hr
RHOL	Liquid density, lb/(cu ft)
EFPUMP	Efficiency of liquid pump, in fraction

TABLE A-1 (Continued)

TITLE	Name of refrigerant
NTB	Number of boiler temperatures to be used (maximum of 8 allowed)
TBT	Boiler temperatures to be used, °F
NTC	Number of condenser temperatures to be used (maximum of 8 allowed)
TCT	Condenser temperatures to be used, °F
NTE	Number of evaporator temperatures to be used (maximum of 8 allowed)
TET	Evaporator temperatures to be used, °F
KCR	Control index for the type of condenser employed; 1 for dry condenser, 2 for wet condenser, 3 for condenser using a prehumidifier, 4 for water-cooled condenser in conjunction with a cooling tower
UAER	UA per sq ft front area for a dry condenser, Btu/(hr)(°F)(sq ft)
EFFAN	Fan efficiency (combined aerodynamic and electrical)
CPL	Specific heat of liquid refrigerant, Btu/(lb)(°F)
TG	Air temperatures at evaporator inlet, outlet, condenser inlet and outlet respectively, °F
TW	Wet bulb temperatures of air at evaporator inlet, outlet, condenser inlet and outlet respectively, °F
NDTE	Number of evaporator approach temperatures to be used (maximum of 5 allowed)
DTET	Evaporator approach temperatures to be used, °F
NDTB	Number of boiler temperatures to be used (maximum of 5 allowed)
DTBT	Boiler temperatures to be used, °F
NDTC	Number of condenser temperatures to be used (maximum of 5 allowed)
DTCT	Condenser temperatures to be used, °F
NTBIN	Number of boiler inlet hot water temperatures to be used (maximum of 5 allowed)
TBINT	Boiler inlet hot water temperatures to be used, °F
NTCIN	Number of condenser inlet cooling water temperatures to be used for the case KCR = 4 (maximum of 5 allowed)
TCINT	Condenser inlet cooling water temperatures to be used, °F

```
* ELT LAGIN2,1,750721, 65739    , 1

      SUBROUTINE  LAGIN2(ID,X,NP,ND,XO,YO,Y)
C     REVISED FOR FORTRAN IV 8-8-65  S. WONG
      DIMENSION  X(2),  Y(2)
C
      ILO=1
      IF(XO-X(1))10,16,4
    4 IF(XO-X(NP))19,13,7
    7 ILO=NP-1
   10 IHI=ILO+1
      GO TO 46
   13 ILO=NP
   16 YO=Y(ILO)
      RETURN
   19 DO 22 ILO=2,NP
      IF(XO-X(ILO))25,16,22
   22 CONTINUE
   25 IHI=ILO
      ILO=IHI-1
      IF(ND-2)46,46,28
   28 DO 43 I=3,ND
      IF(ILO-1)40,40,31
   31 IF(IHI-NP)34,37,37
   34 IF (2.*XO-X(ILO-1)-X(IHI+1)) 37,37,40
   37 ILO=ILO-1
      GO TO 43
   40 IHI=IHI+1
   43 CONTINUE
   46 YO=0.0
      PN=1.0
      DO 49 I=ILO,IHI
   49 PN=PN*(XO-X(I))
      DO 58 I=ILO,IHI
      P=PN/(XO-X(I))
      DO 55 J=ILO,IHI
      IF(J-I)52,55,52
   52 P=P/(X(I)-X(J))
   55 CONTINUE
      YO=YO+P*Y(I)
   58 CONTINUE
      RETURN
      END
```

Figure A-1. Computer Program Listing

```
# ELT NEWTON,1,750721, 65740    , 1

      SUBROUTINE NEWTON(N1,NGO,X,Y,X0,Y0,XMIN,XMAX,ER)
C     SET N1=1 IN MAIN PROGRAM BEFORE CALL NEWTON.  THE ROUTINE
C     FINDS X FOR Y=0  GIVES NGO=1 IF RECALCULATIONS REQUIRED.
C     GIVES NGO=2 IF CONVERGENCE REACHED.
C
    1 FORMAT (1H0,'EXCEEDED 20 ITERATIONS IN NEWTON')
    2 FORMAT  (10X,'X           X0          XS          Y           Y0          SLOPE
     1        ER'/1X,7G10.4)
C
      N1=N1-1
      XS=X
      IF (ABS(Y)-ER) 8,8,5
    5 IF(N1)7,6,7
    6 Y0=Y
      X0=X
      X=X0+(XMAX-XMIN)*0.01
      IF(X-XMAX)21,21,14
   14 X=X0-(XMAX-XMIN)*0.01
      GO TO 21
    7 SLOPE=(Y-Y0)/(X-X0)
      IF (SLOPE) 40,6,40
   40 Y0=Y
      X0=X
      X=X-Y/SLOPE
      IF(N1+8)20,20,21
   20 X=0.5*(X+X0)
      IF(N1+20)22,22,21
   22 WRITE (6,1)
      WRITE (6,2) X,X0,XS,Y,Y0,SLOPE,ER
      N1=0
      ER=5.*ER
   21 CONTINUE
      IF(X-XMIN)11,11,12
   11 X=XMIN
      IF(X-X0)9,8,9
   12 IF(X-XMAX)9,13,13
   13 X=XMAX
      IF(X-X0)9,8,9
    9 NGO=1
      RETURN
    8 NGO=2
      N1=1
      RETURN
      END
```

Figure A-1. (Continued)

```
# FLT RANKIN,1,750730, 45327   , 1

C      MAIN PROGRAM FOR RANKINE REFRIGERATION CYCLE  K.C. HWANG  MAY 1975
C
       DIMENSION W(15),T(15),G(15),P(15),H(15),DPP(15),RHO(15),VIST(15)
       DIMENSION TT(8),COND(5,4)
       DIMENSION TG(4),HG(4),DTHX(4),TW(4)
       REAL MTC,MW,NSNC,NBC,NC,NST
       DIMENSION TTH(17),HVT(17),PT(17),RHOVT(17),HLT(17)
       DIMENSION TBT(8),TCT(8),TET(8)
       DIMENSION DTET(5),DTBT(5),DTCT(5),TBINT(5),TCINT(5)
       DIMENSION RDATE(2),RTIME(2),DATIME(5)
C
       DATA RTIME,RDATE /6H      ,2H  ,6H      ,3H   /
       DATA DATIME /6HRUN ON,6H      ,6H     A,6HT      ,6H      /
       DATA (COND(I,1),I=1,5)/'DRY CONDENSER EMPLOYED        '/
       DATA (COND(I,2),I=1,5)/
      1                 'WET CONDENSER EMPLOYED        '/
       DATA (COND(I,3),I=1,5)/'PRECOOLER/HUMIDIFIER EMPLOYED '/
       DATA (COND(I,4),I=1,5)/
      1                 'COOLING TOWER EMPLOYED        '/
       NAMELIST /INPUT/ VIST,TT,TTH,HVT,HLT,PT,RHOVT,CP,GAMMA,AK
      1,MW,DPP,EFM,UR,RHOL,EFPUMP,TITLE,NTB,TBT,NTC,TCT,NTE,TET,
      2                    KCR,              UAER,EFFAN,CPL
      3,TG,TW,NDTE,DTET,NDTB,DTBT,NDTC,DTCT,NTBIN,TBINT,NTCIN,TCINT
C
C
     1 CONTINUE
       UAE=0.
       ZR=0.
       READ (5,INPUT,END=150)
       NPAGE=0
       N=0
       WRITE (3,2) TITLE,(COND(I,KCR), I=1,5)
     2 FORMAT (1H1,24X,'SOLAR POWERED AIR CONDITIONING SYSTEM USING',
      1 4X,A6/25X,5A6//)
       WRITE(6,INPUT)
       DO 95 NTBIN1=1,NTBIN
       TBIN=TBINT(NTBIN1)
C      EVAPORATING TEMP VARIATION
       DO 95 NE=1,NTE
       T(1)=TET(NE)
       T(5)=T(1)
C      CONDENSING TEMP VARIATION
       DO 95 NA=1,NTC
       T(4)=TCT(NA)
       DO 95 NB=1,NTB
       POWRCT=0.
       POWRWP=0.
       POWRFC=0.0
       DPCT=0.0
       FANCC=0.
       WTFC=0.0
       T(7)=TBT(NB)
C      EVAPORATOR APPROACH TEMP
       DO 95 NDTE1=1,NDTE
       DTHX(1)=DTET(NDTE1)
       DO 95 NDTB1=1,NDTB
       DTHX(2)=DTBT(NDTB1)
C      CONDENSER APPROACH TEMP
```

Figure A-1. (Continued)

```
      DO 95 NDTC1=1,NDTC
      DTHX(3)=DTCT(NDTC1)
      IF(KCR .NE. 4) GO TO 3
      DO 95 NTCIN1=1,NTCIN
      TCIN=TCINT(NTCIN1)
    3 CONTINUE
      KONT=0
      DO 701 K=1,3
      CALL LAGIN2(1,TTH,17,2,T(7),P(7),PT)
      CALL LAGIN2(2,TTH,17,2,T(1),P(1),PT)
      CALL LAGIN2(3,TTH,17,2,T(4),P(4),PT)
      P(6)=P(7)*(1.+DPP(2))
      P(5)=P(1)*(1.+DPP(1))
      P(3)=P(4)*(1.+DPP(3))
      P(2)=P(3)
      P(8)=P(3)
      CALL VAPOR(P(1),T(1),H(1),RHO(1))
      CALL VAPOR(P(7),T(7),H(7),RHO(7))
      CALL LAGIN2(4,TTH,17,2,T(4),H(4),HLT)
      H(5)=H(4)
      H(6)=H(4)
      Q(1)=QR
      W(1)=Q(1)/(H(1)-H(4))
      W(5)=W(1)
      W(2)=W(1)
      W1=W(1)/60.
      CALL TURCPR(P(1),T(1),P(2),W1   ,P(7),T(7),P(8),W7)
      W(7)=W7 *60.
      CALL PHTR(P(2),H(2),T(2),RHO(2))
      H(2)=H(1)+DHC/EFCF/778.3
      W(8)=W(7)
      W(6)=W(7)
      W(3)=W(2)+W(8)
      W(4)=W(3)
      H(3)=(W(2)*H(2)+W(8)*H(8))/(W(2)+W(8))
      CALL PHTR(P(3),H(3),T(3),RHO(3))
      Q(3)=W(3)*(H(3)-H(4))
      PUMP=P(6)-P(4)
      VL=W(6)/RHOL   /3600.
      PWRLP=VL*PUMP*144./.738/EFPUMP
      DQPWR=PWRLP*3.413
      H(6)=H(4)+DQPWR/W(6)
      T(6)=T(4)+DQPWR/(W(6)*CPL)
      Q(2)=W(6)*(H(7)-H(6))
      COP=Q(1)/(Q(2)+DQPWR)
      NAMELIST /CHECK/T,P,H,Q,RHO,EFC,EFT,W,EFCF,PWRLP,COP
  701 CONTINUE
      CALL BOILER
      CALL EVAP
      CALL CONDSR(KCR)
      IF(KONT .NE. 0) GO TO 95
      TW3=TW(3)
      TW4=TW(4)
      IF(KCR.EQ.4)TW3=1.E20
      IF(KCR.EQ.4)TW4=1.E20
      TGC=TG(3)
      TG4=TG(4)
      IF(KCR.EQ.3) TGC=TG3
      IF(KCR.EQ.4)TGC=TCIN
      IF(KCR.EQ.4)TG4=TCOUT
```

Figure A-1. (Continued)

```
      IF(KCR.EQ.4) GC=WCOPL
      UA3=UAC
      IF(KCR.EQ.2) UA3=0.0
      PCOP=W(7)*(H(7)-H(8))/Q(2)
      RCOP=Q(1)/(W(1)*(H(2)-H(1)))
      COST1=COSTH+COSTE+COSTC+100.+40.+FANCE+FANCC
      COSTF=1.65*COST1
      COSTU=6.13*COST1
      PTOT=POWRFE+POWRFC+POWRCT+POWRWP+PWRLP
   80 NPAGE=NPAGE+1
      CAL  DATE (16,DATIME)
      C     TOD (28,DATIME)
      WRITE (6,90) TITLE,(COND(I,KCR), I=1,5),DATIME,NPAGE
   90 FORMAT(1H1////// 24X,'SOLAR POWERED AIR CONDITIONING SYSTEM USING
     1',4X,A6/24X,5A6,15X,5A6,20X,'PAGE ',I3//
     2                T3,'STATION/ID'T15,'TEMPERATURE'T31,'PRESSURE'T46,'E
     1NTHALPY'T61,'FLOW RATE'T78,'DENSITY'            /T18,'DEG F'T33,'PSIA'
     2T47,'BTU/LB'T63,'LB/HR'T78,'LB/CU FT'            /)
      DO 91 N=1,8
      WRITE(6,92)N,T(N),P(N),H(N),W(N),RHO(N)
   91 CONTINUE
   92 FORMAT(I8,6F15.4)
      WRITE(6,97)
   97 FORMAT(//'     HEAT            HOT FLUID                COLD FLUID
     1     UA        WEIGHT          COST       FAN DP  FAN POWER       Q       WET
     1BULB(F)'/
     1                                                     ' EXCHANGER    F
     2LO       TEMP(F)        FLO      TEMP(F)          (BTU/HR/     (LB)        (
     3US S)     (IN-H2O)   (WATT)     (BTU/HR)      IN        OUT'/
     4                                                      12X'(LB/HR)   IN      OU
     5T    (LB/HR)   IN       OUT        (DEG F)     HX      FAN      HX     FAN'/)
      WRITE(6,98)G,TG(1),TG(2),W(5),T(5),T(1),UAE,WTE,WTFE,COSTE,FANCE,
     1                                                              DPET,POWRFE
     1,Q(1),TW(1),TW(2)
      WRITE(6,99)    WSAB,TBIN,TBOUT,W(6),T(6),T(7),UAB,WTB,ZR,COSTB,ZR,
     1ZR,ZR,Q(2)
      WRITE(6,102)W(3),T(3),T(4),GC,TGC,TGH,UA3,WTC,WTFC,COSTC,
     1FANCC,DPCT,POWRFC,Q(3),TW(3),TW(4)
   98 FORMAT( '  EVAP'
     1T10,F8.0,F8.1,F7.1,F8.0,F7.1,F6.1,F9.2,4F7.1,F4.2,F10.1,F11.0,
     2F7.1,F8.1)
   99 FORMAT( ' BOILER'
     1T10,F8.0,F8.1,F7.1,F8.0,F7.1,F6.1,F9.2,4F7.1,F4.2,F10.1,F11.0,
     2F7.1,F8.1)
  102  FORMAT( ' CONDNSR'
     1T10,F8.0,F8.1,F7.1,F8.0,F7.1,F6.1,F9.2,4F7.1,F4.2,F10.1,F11.0,
     2F7.1,F8.1)
      WRITE(6,104)
  104 FORMAT(///' COEF OF PERFORMANCE',T30,'TURBO-COMPRESSOR',T60,'ELECT
     1RIC POWER REQD(WATT)',  T90,'SYSTEM COST($)'/ )
      WRITE(6,105)PCOP,DC,POWRFE,COSTF
  105 FORMAT(' POWER COP',T15,F8.3,T30,'COMPR DIA(IN)'      ,T45,F8.3,
     2T60,'EVAP FAN',T75,F8.3,T90,'FACTORY COST   ',T105,F8.0)
      WRITE(6,106)RCOP,EFCF,POWRFC
  106 FORMAT(' REFRIG COP',T15,F8.3,T30,'COMPR EFF',T45,F8.3, T60,'CONDS
     1R FAN',T75,F8.3)
      WRITE(6,107)COP,NC,POWRCT,COSTU
  107 FORMAT(' SYSTEM COP',T15,F8.3,T30,'RPM',T45,F8.0,T60,'CL TOWER FAN
     1',T75,F8.3,T90,'USER COST',T105, F8.0)
      WRITE(6,108)DT,POWRWP
```

Figure A-1. (Continued)

```
  108 FORMAT(T30,'TURBN DIA(IN)',T45,F8.3,T60,'WATER PUMP',T75,F8.3)
      WRITE(6,109)EFT,PWRLP
  109 FORMAT(T30,'TURBN EFF',T45,F8.3,T60,'FREON PUMP',T75,F8.3)
      WRITE(6,110)PTOT
  110 FORMAT(T60,'TOTAL',T75,F8.3)
   95 CONTINUE
      GO TO 1
  150 STOP
C
C
C
      FUNCTION VISCF(T)
      CALL LAGIN2(8,TT,8,2,T,VISCF,VIST)
      VISCF=VISCF*6.7197E-4
      RETURN
C
C
      SUBROUTINE VAPOR(P,T,HV,RHOV)
      CALL LAGIN2(17,PT,17,2,P,HSV,HVT)

      CALL LAGIN2(18,PT,17,2,P,RHOVS,RHOVT)
      CALL LAGIN2(19,PT,17,2,P,TS,TTH)
      HV=HSV+CP*(T-TS)
      RHOV=RHOVS*(TS+460.)/(T+460.)
      RETURN
C
C
      SUBROUTINE PHTR(P,H,T,RH)
      N1=1
  601 CONTINUE
      CALL VAPOR(P,T,H1,RH)
      DH1=H-H1
      ER=.005*H
      CALL NEWTON(N1,NGO,T,DH1,T0,DH10,-40.,280.,ER)
      GO TO(601,602),NGO
  602 CONTINUE
      RETURN
C
C
      SUBROUTINE TURCPR(PCI,TCI,PCO,WC,PTI,TTI,PTO,WT)
C
C     THIS IS TURBOCOMPRESSOR DESIGN ROUTINE
C    PCI   CCOMPRESSOR INLET PRESSURE
C    TCI   COMPRESSOR INLET TEMPERATURE
C    PCO   COMPRESSOR OUTLET PRESSURE
C    WC   COMPRESSOR MASS FLOW, LB/MIN
C    PTI   TURBINE INLFT PRESSURE
C    TTI   TURBINE INLET TFMPERATURE
C    PTO   TURBINE OUTLET PRESSURE
C
      DATA GG/32.174/
      R=1.987/MW
      EK=GAMMA/(GAMMA-1.)
      EK1=1./EK
      DHC=EK*R*(T(1)+460.)*((P(2)/P(1))**EK1-1.)*778.3
      DHT=EK*R*(T(7)+460.)*(1.-(P(8)/P(7))**EK1)*778.3
      RHOC=RHO(1)
      VISC=VISCF(T(1))
      UTC=SQRT(GG*DHC/0.60)
      SV=AK*SQRT(TCI+460.)
```

Figure A-1. (Continued)

```
      MTC=UTC/SV
      VCI=WC/(60.*RHOC)
      NSNC=SQRT(VCI)/60/(GG*DHC)**0.75
      NSC=0.02
      WTO=1.E+20
   10 CONTINUE
      NC=NSC/NSNC
      DC=UTC*720./(3.1416*NC)
      DIMENSION NSCT(50),ECT(50),MTCT(5)
      DATA NSCT/
     $.02, .04, .06, .08, .10,.12,.14,.16,.20,.25,
     $.02, .04, .06, .08, .10,.12,.14,.16,.20,.25,
     $.02, .04, .06, .08, .10,.12,.14,.16,.20,.25,
     $.02, .04, .06, .08, .10,.12,.14,.16,.20,.25,
     $.02, .04, .06, .08, .10,.12,.14,.16,.20,.25/
      DATA ECT/.65,.82,.86,.885,.90,.91,.92,.92,.92,.91,
     1.617, .78, .845, .87, .88, .89, .90, .9, .89, .88,
     2.57, .745, .81, .84, .85, .86, .865, .87, .87, .85,
     3.50, .70, .755, .79, .815, .82, .825, .825, .82, .80,
     4.42, .626, .70, .73, .75, .755, .76, .755, .74, .72/
      DATA MTCT/0., 0.2, .5, 1., 1.71/
      EFS=XYZMAP(1,NSCT,ECT,10,MTCT,5,2,2,NSC,MTC,ANS)
      EFEFS=1.0
C
C   EFFECT OF IMPELLER SIZE
C
      DIMENSION DCT(7),EFEFST(7)
      DATA DCT/1., 1.5, 2., 2.5, 3., 3.5, 4./
      DATA EFEFST/.8, .88, .93, .965, .98, .99, 1./
      IF(DC.LT.4.)CALL LAGIN2(100,DCT,7,2,DC,FFEFS,EFEFST)
      EFC=EFS*EFEFS
C
C     CORRECT FOR REYNOLDS NUMBER
C
      EFCF=EFC
      RE=UTC*RHOC*DC/(12.*VISC)
      IF(RE .LT. 1.E6)
     1EFCF=1.-(1.-EFC)*(1.E+6/RE)**0.1
      HPC=WC*DHC/(EFCF*33000.)
      EFWT=WC*DHC/(EFCF*EFM*DHT)
      DATA EFT/.80/
      NI=1
  100 CONTINUE
      WT=EFWT/EFT
      DT=SQRT(GG*DHT)*720./(3.1416*NC)
      HTI=H(7)
      HTO=HTI-DHT*EFT/778.3
      H(8)=HTO
      CALL PHTR(P(8),H(8),T(8),RHO(8))
      RHOT=RHO(8)
      VISCT=VISCF(T(8))
      VTO=WT/(60.*RHOT)
C   VOSCT=
      NST=NC*SQRT(VTO)/60./(GG*DHT)**.75
      DIMENSION NSTT(60),EFTT(60),DTT(6)
      DATA DTT/1.0, 1.5, 2.0, 3., 5., 6./
      DATA NSTT/
     $.0, .01, .02, .03, .04, .06, .08, .1, .12, .14,
     $.0, .01, .02, .03, .04, .06, .08, .1, .12, .14,
     $.0, .01, .02, .03, .04, .06, .08, .1, .12, .14,
```

Figure A-1. (Continued)

```
                 5.0, .01, .02, .03, .04, .06, .08, .1, .12, .14,
                 5.0, .01, .02, .03, .04, .06, .08, .1, .12, .14,
                 5.0, .01, .02, .03, .04, .06, .08, .1, .12, .14/
                  DATA EFTT/
                 10.,0.25,.429,.53,.697,.694,.731,.729,.715,.70,
                 20.,.375,.544,.636,.696,.763,.786,.779,.763,.745,
                 30.,.465,.615,.704,.757,.815,.835,.825,.806,.783,
                 40.,.504,.655,.743,.796,.857,.880,.870,.852,.830,
                 5 0.,.556,.70,.776,.830,.896,.918,.911,.894,.871,
                 6 0.,.556,.70,.776,.830,.896,.918,.911,.894,.871/
                  EFT1=XYZMAP(1,NSTT,EFTT,10,DTT,6,2,2,NST,DT,ANS2)
                  RET=24.*WT/(60.*VISC1*DT)
                  IF(RET.LT. 2.E5) EFT1=1.-(1.-EFT1)*(.4+.6*(RET/2.E5)**-.2)
                  DEFT=EFT-EFT1
                  CALL NEWTON(N1,NGO,EFT,DEFT,EFTO,DEFO,.3511,.9255,.00519)
                  GO TO (100,200),NGO
      C
        200 CONTINUE
            IF(WT.GE.WTO) GO TO 201
            WTO=WT
            NSC=NSC+.005
            GO TO 10
      C
        201 CONTINUE
            RETURN
      C
      C
            FUNCTION HTYF(T)
            VP=VPP (T)
            HTYF=VP*18.0/((14.7-VP)*29.)
            RETURN
      C
            SUBROUTINE BOILER
      C
            TBOUT=T(7)+DTHX(2)
            WSAB=Q(2)/(TBIN-TBOUT)
            DT1=TBIN-T(7)
            DT2=TBOUT-T(7)
            IF((TBIN.LE.TBOUT).OR.(DT1.LE.0.).OR.(DT2.LE.0.))GO TO 202
            DTB=TLAVG(DT1,DT2)
            HB=200.
            UAB=Q(2)/DTB
            AB=UAB/HB
            WTB1=AB*144.*.016*.321
            WTB=WTB1*2.00
            VOLB1=AB/150.
            VOLB=1.1*VOLB1
            COSTB=1.53*WTB
            RETURN
        202 CONTINUE
            KONT=1
            RETURN
      C
            SUBROUTINE SURFT(TA,HTYA,TS,HTY)
      C
            TS=T(1)+0.6*(TA-T(1))
            N2=1
        777 CONTINUE
            HTY=HTYF(TS)
            UAEL=UAER*2./0.27
```

Figure A-1. (Continued)

AIRESEARCH MANUFACTURING COMPANY
OF CALIFORNIA

```
      QS=UAER*(TA-TS)*2.0
      QL=UAEL* (HTYA-HTY)*1060.
      DTI=(QL+QS)/(UAER*2)
      TS1=T(1)+DTI
      DTS=TS-TS1
      ER1=0.01*(TA-T(1))
      CALL NEWTON(N2,NGO2,TS,DTS,TS0,DTS0,T(1),TA,ER1)
      GO TO (777,778), NGO2
  778 CONTINUE
      RETURN
C
      SUBROUTINE EVAP
C
      TG(2)=T(1)+DTHX(1)
      HWE=HWF(T(1))
      HG(1)=HWF(TW(1))
      DATA RNE/3./
      N3=1
  314 CONTINUE
      WBDRE=WBDRF(RNE)
      TW(2)=TG(2)-(TG(1)-TW(1))*WBDRE
      HG(2)=HWF(TW(2))
      DHG=HG(1)-HG(2)
      G=Q(1)/DHG
      RHOA=29./359.*(492./(TG(1)+460.))
      DH1=HG(1)-HWE
      DH2=HG(2)-HWE
      DHAVG=TLAVG(DH1,DH2)
      AFE=G/(500.*60.*RHOA)
      Q1=330.*RNE*AFE*DHAVG
      DQ1=Q(1)-Q1
      ER3=Q(1)*0.005+1.E-20
      CALL NEWTON(N3,NGO3,RNE,DQ1,RNE0,DQ10,.5,20.,ER3)
      GO TO (314,315), NGO3
  315 CONTINUE
      QSENS=G*0.230*(TG(1)-TG(2))
      QLAT=Q(1)-QSENS
      RD=.866/12.
      VOLE=AFE*RNE*RD
      WTE=38.5*VOLE
      COSTE=.76*WTE
      DPE=1.36*.088*RNE**.746
      DPET=4./3.*DPE
      DPET=DPET+.15
      VAE=G/RHOA/3600.
      POWRFE=VAE*DPET*5.20/.738/EFFAN
      CFME=G/RHOA/60.
      CALL FANWTC(CFME,DPE,WTFE,FANCE)
      FANCE=FANCE+10.+32.5*POWRFE/1000.*0.7
      RETURN
C
      SUBROUTINE CONDSR(KCR)
C
      GO TO (161,162,173,174), KCR
  161 CONTINUE
      TG(4)=T(4)-DTHX(3)
      TW(4)=TW(3)
      DT1=T(4)-TG(3)
      DT2=T(4)-TG(4)
      IF(TG(4) .LE. TG(3))GO TO 163
```

Figure A-1. (Continued)

```
      IF((DTI .LE.0.) .OR. (DT2.LE.0.)) GO TO 163
      DTC=TLAVG(DTI,DT2)
      UAC=Q(3)/DTC
      GC=Q(3)/0.23/(TG(4)-TG(3))
      RHOAC=29./359.0*(492./(TG(3)+460.))
      AFC=GC/(500.*60.*RHOAC)
      RNC=UAC/(UAER*AFC)
      RD=.666/12.0
      VOLC=AFC*RNC*RD
      WTCORC=35.*VOLC
      WTC=1.1*WTCORC
      COSTC=0.76*WTC
      DPC=       .088*RNC**.746
      GO TO 164
C     WET CONDENSER CALCS
  162 CONTINUE
      TG(4)=1.E20
      RHUAC=29./359.*(492./(TG(3)+460.))
      TW(4)= T(4)-DTHX(3)
      HG(4)=HWF(TW(4))
      HG(3)=HWF(TW(3))
      DHGC=HG(4)-HG(3)
      GC=Q(3)/DHGC
      AC=Q(3)/1000.
      N4=1
      ERC=.005*Q(3)+1.E-20
  410 ACM=AMAX1(1.,AC/3.)
      ACX=5.*AC
      DTI=Q(3)/300.00/AC
      TWC=T(4)-DTI
      HWC=HWF(TWC)
      DH1C=HWC-HG(3)
      DH2C=HWC-HG(4)
      IF((DHGC.LE.0.).OR.(DH1C.LE.0.).OR.(DH2C.LE.0.))GO TO 163
      DHAVGC=TLAVG(DH1C,DH2C)
      QH=373.*DHAVGC*AC
      DQC=Q(3)-QH
      CALL NEWTON(N4,NGO4,AC,DQC,AC0,DQC0,ACM,ACX,ERC)
      GO TO (410,411), NGO4
  411 CONTINUE
      WTC1=AC*144.*.016*.321
      WTC=WTC1*3.00
      COSTC=.76*WTC
      COSTC=COSTC+40.
      DPC=0.50
      POWRWP=1.E-3*Q(3)
      GO TO 164
  173 CONTINUE
C   HUMIDIFIER USED TO PRECOOL AIR FOR CONDENSER
C
      TG3=TG(3)-.9*(TG(3)-TW(3))
      TG(4)=T(4)-DTHX(3)
      TW(4)=TW(3)
      DT1=T(4)-TG3
      DT2=T(4)-TG(4)
      DTRC=TG(4)-TG3
      IF(DTRC .LE. 0.)  GO TO 163
      IF((DT1.LE.0.).OR.(DT2.LE.0.))GO TO 163
      DTC=TLAVG(DT1,DT2)
      UAC=Q(3)/D C
```

Figure A-1. (Continued)

```
      GC=Q(3)/.230/DTRC
      RHOAC=29./359.*(492./(TG3+460.))
      AFC=GC/(500.*60.*RHOAC)
      RNC=UAC/(UAFR*AFC)
      RD=.866/12.0
      VOLC=AFC*RNC*RD
      WTCORC=35.*VOLC
      WTC=1.1*WTCORC
      COSTC=.76*WTC
      COSTC=COSTC*1.20
      DPC=       .088*RNC**.746+.10
      POWRWP=1.0E-3*Q(3)
      GO TO 164
  174 CONTINUE
C
C   COOLING TOWER EMPLOYED
C
C
      TCOUT =T(4)-DTHX(3)
      WCOOL=Q(3)/(TCOUT-TCIN)
      DT1=T(4)-TCIN
      DT2=T(4)-TCOUT
      IF((DT1.LE.0.).OR.(DT2.LE.0.))GO TO 163
      IF(WCOOL.LE.0.) GO TO 163
      DTC=TLAVG(DT1,DT2)
      HC=200.
      UAC=Q(3)/DTC
      AC=UAC/HC
      WTC1=AC*144.*.016*.321
      WTC=WTC1*2.000
      COSTC=1.53*WTC+1.905E-3*Q(3)+40.
      POWRCT=2.96E-3*Q(3)
      POWRWP=1.9E-3*Q(3)
      RETURN
C
C
C
  163 CONTINUE
      KONT=1
      RETURN
  164 CONTINUE
      DPCT=4./3.*DPC
      DPCT=DPCT+.15
      VAC=GC/RHOAC/3600.
      POWRFC=VAC*DPCT*5.20/.738/EFFAN
      CFMC=VAC*60.
      CALL FANWTC(CFMC,DPCT,WTFC,FANCC)
      FANCC=FANCC+10.+32.5*POWRFC/1000.*.70
      RETURN
C
C
      FUNCTION WBDRF(RN)
      DIMENSION WBDRT(7),RNT(7)
      DATA RNT/0., 1.0, 2., 3., 6., 8., 10./
      DATA WBDRT/1.0, .667, .442, .306, .162, .112, .074/
      CALL LAGIN2(163,RNT,7,2,RN,WBDRF,WBDRT)
      RETURN
C
C
      FUNCTION HWF(TW)
```

Figure A-1. (Continued)

```
      DIMENSION  HWT(11), TWT(11)
      DATA TWT/20., 30.,40.,50.,60.,70.,80., 90., 100., 110., 120./
      DATA HWT/7.1,10.9,15.2,20.3,26.4,34.1,43.7, 55.93, 71.73, 92.34,
     1120.1/
      CALL LAGIN2(164,TWT,11,2,1,T,HWF,HWT)
      RETURN
C
C
      SUBROUTINE FANWTC(CFM,DP,WTF,FANCST)
      DIMENSION DPT(5),DWT(5)
      DATA DPT/.25, .50, 1.0, 2.0, 4.0/
      DATA DWT/122.,106.,85.,50.5,30.4/
      CALL LAGIN2(191, DPT,5,2, DP,DWF,DWT)
      WTF=10.0 +CFM*DWF/4000.
      FANCST=2.20*WTF
      FANCST=FANCST*0.40
      RETURN
      END
```

Figure A-1. (Continued)

```
@ ELT 81190,1,750721, 65748    , 1

C      UCC       81190      FUNCTION XYZMAP    STANG=HUI LIN (AM=303=MR)    10/00/66
       FUNCTION XYZMAP(IND,X,Y,NP,Z,NC,IDX,IDY,BX,BY,ANS)
C
C      FUNCTION XYZMAP HAS THE CAPABILITY OF SUBROUTINES MAPRDY + LAGIN2,
C      ANSWER IS ALSO AVAILABLE AT LOCATION (ANS),
C      ALSO CAPABLE OF HANDLING MAP THAT HAS Z=LINES CROSSING EACH OTHER,
C
C      X=ABSISSA, Y=ORDINATE, Z=THIRD PARAMETER,
C
C      IND=0, Z=F(X,Y), (EQUIVALENT TO SUBROUTINE MAPRDY),
C      IND=1, Y=F(X,Z), (EQUIVALENT TO SUBROUTINE MAPRDY),
C      IND=-1, Y=F(X) ONLY, (EQUIVALENT TO LAGIN2), THEN Z,NC,IDY, AND AY
C       ARE DUMMY VARIABLES THAT ARE NOT NEEDED IN ACTUAL INTERPOLATION,
C
C      XS MUST BE STORED IN ASCENDING ORDER FOR EACH Z,  SIMILARLY,
C      SMALLEST Z BE FED IN AS Z(1), ZS ARE IN ASCENDING ORDER,
C      XS NEED NOT BE THE SAME VALUES FOR VARIOUS ZS,
C      X,Y AND Z ARE TO BE DIMENSIONED IN THE MAIN (OR CALLING) PROGRAM,
C      THEY MUST BE DIMENSIONED NOT LESS THAN *** X(NP*NC), Y(NP*NC) AND
C       Z(NC) *** NOTE NC MAY NOT BE GREATER THAN 20 ***
C
C      NP=NUMBER OF POINTS PER CURVE (OR NUMBER OF X,Y PAIRS FOR EACH Z),
C      NC=NUMBER OF CURVES (OR NUMBER OF ZS), 1 TO A MAXIMUM OF 20
C
C      IDX=POINTS USED FOR INTERPOLATION IN X-DIRECTION,
C      IDY=POINTS USED FOR INTERPOLATION IN Y-DIRECTION (IND=0),
C                                   OR IN Z-DIRECTION (IND=1),
C      IDX OR  IDY CAN EITHER BE 2 OR 3 ONLY,
C      BX=FIRST INDEPENDENT VARIABLE,
C      BY=SECOND INDEPENDENT VARIABLE, (WHEN IND=0 OR 1 ONLY),
C      BY=Y INDEPENDENT VARIABLE, WHEN (IND=0),
C      BY=Z INDEPENDENT VARIABLE, WHEN (IND=1),
C      ANS=DEPENDENT VARIABLE Z(X=AX,Y=AY), WHEN IND=0,
C      ANS=DEPENDENT VARIABLE Y(X=AX,Z=AY), WHEN IND=1,
C      ANS=DEPENDENT VARIABLE Y(X=AX),  WHEN IND=-1,
C
C      NO PRINT OUT, IF DATA OFF THE RANGE OF MAP OR CURVE,
C      THEN, USE 2-POINT INTERPOLATIONS AUTOMATICALLY,
C
C      ****************************************************************
C      XS, YS, AND ZS ARE READ IN IN MAIN PROGRAM RECOMMENDED AS FOLLOWS,
C4100  FORMAT(8I10)                                                *****
C4101  FORMAT(8F10.0)                                              *****
C      ********** FOR IND=0 OR 1 **************************************
C4101  READ (5,4100)         NP,NC                                 *****
C      DO 100 N=1,NC                                               *****
C      READ (5,4101)         Z(N)                                  *****
C      ME=N*NP                                                     *****
C      MS=ME-NP+1                                                  *****
C 100  READ (5,4101)          (X(M),Y(M),M=MS,ME)                  *****
C      ********** FOR IND=-1 ******************************************
C      READ (5,4100)          NP                              ***********
C      READ (5,4101)          (X(M),Y(M),M=1,NP)              ***********
C      ****************************************************************
C
       DIMENSION X(2),Y(2),Z(2),ZZ(20),ZX(20)
       JS=1
       IF (IND) 105,106,103
```

Figure A-1. (Continued)

REPRODUCIBILITY OF THE
ORIGINAL PAGE IS POOR

```
  103 DO 104 I=1,NC
  104 ZX(I)=Z(1)
      GO TO 141
  105 JF=1
      GO TO 108
  106 DO 107 I=1,NC
  107 ZZ(I)=Z(I)
      JE=NC
      NCROSS=0
  108 DO 126 J=JS,JE
      JX2=(J-1)*NP
      DO 109 I=1,NP
      J2=I+JX2
      IF (BX-X(J2)) 114,110,109
  109 CONTINUE
      GO TO 119
  110 ANS=Y(J2)
      GO TO 123
  114 IF(I-IDX)120,120,119
  119 JX2=J2-IDX
  120 IS=JX2+1
      IE=JX2+IDX
      IF(IDX.GT.2) GO TO 122
  121 ANS=(Y(IE)*(BX-X(IS))-Y(IS)*(BX-X(IE)))/(X(IE)-X(IS))
      GO TO 123
  122 IM=IS+1
      G1=(BX-X(IS))/(X(IM)-X(IE))
      G2=(BX-X(IM))/(X(IE)-X(IS))
      G3=(BX-X(IE))/(X(IS)-X(IM))
      ANS=-Y(IS)*G2*G3-G1*(Y(IM)*G3+Y(IE)*G2)
  123 IF (IND) 150,125,124
  124 ZZ(J)=ANS
      GO TO 126
  125 ZX(J)=ANS
      IF(ANS.LT.ZX(1)) NCROSS=1
  126 CONTINUE
      IF(IND.NE.0) GO TO 1151
      IF(NCROSS.EQ.0) GO TO 141
C
      DO 130 K=2,NC
      JMIN=K-1
      DO 129 IP=K,NC
      IF (ZX(IP)-ZX(JMIN)) 128,128,129
  128 JMIN=IP
  129 CONTINUE
      IK=K-1
      C1=ZX(JMIN)
      Z1=ZZ(JMIN)
      ZX(JMIN)=ZX(IK)
      ZZ(JMIN)=ZZ(IK)
      ZX(IK)=C1
  130 ZZ(IK)=Z1
C
  141 ICPY= IDY-1
      DO 142 I=1,NC
      IF(BY-ZX(I))145,144,142
  142 CONTINUE
      JS=NC-ICPY
      GO TO 147
  144 JS=I
```

Figure A-1. (Continued)

```
      IF (IND.EQ.0) GO TO 151
      JE=JS
      GO TO 108
  145 IF(I.LE.IDY) GO TO 147
  146 JS=I-ICPY
  147 JE=JS+ICPY
      IF (IND) 108,1152,108
C     Y=F(X,Z) OR Z=F(X,Y) CALCULATION DEPENDING ON IND=0 OR 1.
  151 ANS=ZZ(JS)
      GO TO 158
 1151 IF(JE.EQ.JS) GO TO 151
 1152 IF(IDY.GT.2) GO TO 153
  152 ANS=(ZZ(JE)*(BY-ZX(JS))-ZZ(JS)*(BY-ZX(JE)))/(ZX(JE)-ZX(JS))
      GO TO 158
  153 JM=JS+1
      G1=(BY-ZX(JS))/(ZX(JM)-ZX(JE))
      G2=(BY-ZX(JM))/(ZX(JE)-ZX(JS))
      G3=(BY-ZX(JE))/(ZX(JS)-ZX(JM))
      ANS=-ZZ(JS)*G2*G3-G1*(ZZ(JM)*G3+ZZ(JE)*G2)
  158 XYZMAP=ANS
      RETURN
      END
```

Figure A-1. (Continued)

```
# ELT TLAVG,1,750721, 65748 , 1

      FUNCTION TLAVG(DT1,DT2)
      TLAVG=(DT1-DT2)/ALOG(DT1/DT2)
      RETURN
      END
```

Figure A-1. (Continued)

```
* ELT VPP,1,750721, 65749        , 1

              FUNCTION VPP(T)
C             FUNCTION TO CALCULATE VAPOR PRESSURE OF WATER AT T
C             T=TEMP  DEG F.
C             VPP=VAPOR PRESS OF WATER  PSIA
              X=647.27-(T+460.)/1.8
              TEMP=X*1.8/(T+460.)*(3.244+5.868E-3*X+1.170E-8*X**3)/(1.+2.188E-3*
             1X)
              VPP=3207./10.**TEMP
              RETURN
              END

  1234 CARDS WRITTEN IN  RANKIN/KCHB    (   718 COMPRESSED CARD IMAGES)
       2 BLOCKS ACQUIRED (KEY = 0101)

END CUR
```

Figure A-1. (Continued)

APPENDIX B

COMPUTER INPUT/OUTPUT DATA FOR SOLAR-POWERED AIR CONDITIONING SYSTEM CONCEPTS

This appendix contains computer input and output data for the four system concepts defined in Figure 1. The data presented are in english units. Input data units are defined in Appendix A, and output data units are given on the printouts. The data are presented as follows:

```
                    SOLAR POWERED AIR CONDITIONING SYSTEM USING     R-11
                    DRY CONDENSER EMPLOYED

$INPUT
VIST   =   .95000000-02,   .10250000-01,   .11000000-01,   .11600000-01,
           .12250000-01,   .12820000-01,   .13400000-01,   .13900000-01,
           .00000000+00,   .00000000+00,   .00000000+00,   .00000000+00,
           .00000000+00,   .00000000+00,   .00000000+00,
TT     =   .00000000+00,   .40000000+02,   .80000000+02,   .12000000+03,
           .16000000+03,   .20000000+03,   .24000000+03,   .28000000+03,
TTH    =  -.40000000+02,  -.20000000+02,   .00000000+00,   .20000000+02,
           .40000000+02,   .60000000+02,   .80000000+02,   .10000000+03,
           .12000000+03,   .14000000+03,   .16000000+03,   .18000000+03,
           .20000000+03,   .22000000+03,   .24000000+03,   .26000000+03,
           .28000000+03,
HVT    =   .87529999+02,   .89949999+02,   .92419999+02,   .94889999+02,
           .97389999+02,   .99679999+02,   .10235999+03,   .10480999+03,
           .10721999+03,   .10958999+03,   .11187999+03,   .11406999+03,
           .11616999+03,   .11812999+03,   .11991999+03,   .12151999+03,
           .12284999+03,
HLT    =   .00000000+00,   .39800000+01,   .79899999+01,   .12030000+02,
           .16120000+02,   .20270000+02,   .24480000+02,   .28750000+02,
           .33080000+02,   .37480000+02,   .41950000+02,   .46470000+02,
           .51070000+02,   .55760000+02,   .60530000+02,   .65459999+02,
           .70569999+02,
PT     =   .73869999+00,   .14190000+01,   .25540000+01,   .43419999+01,
           .70299999+01,   .10910000+02,   .16310000+02,   .23600000+02,
           .33180000+02,   .45500000+02,   .61010000+02,   .80179999+02,
           .10352999+03,   .13158000+03,   .16487000+03,   .20397000+03,
           .24947000+03,
RHOVT  =   .22600000-01,   .41539999-01,   .71709999-01,   .11739999+00,
           .18350000+00,   .27590000+00,   .40100000+00,   .56630000+00,
           .78009999+00,   .10520000+01,   .13920000+01,   .18140000+01,
           .23330000+01,   .29680000+01,   .37440000+01,   .46960000+01,
           .58800000+01,
CP     =   .14000000+00,
GAMMA  =   .11100000+01,
AK     =   .19800000+02,
TR     =   .16500000+03,
TE     =   .40000000+02,
TC     =   .10500000+03,
MW     =   .13740000+03,
DPP    =   .49999999-01,   .49999999-01,   .49999999-01,   .49999999-01,
           .49999999-01,   .49999999-01,   .49999999-01,   .49999999-01,
           .49999999-01,   .49999999-01,   .49999999-01,   .49999999-01,
           .49999999-01,   .49999999-01,   .49999999-01,
EFM    =   .89999999+00,
QR     =   .36000000+05,
RHOL   =   .91000000+02,
EFPUMP =   .50000000+00,
TITLE  =   .25018020+18,
NTB    =              1,
TBT    =   .18500000+03,   .18500000+03,   .18000000+03,   .19000050+03,
           .00000000+00,   .00000000+00,   .00000000+00,   .00000000+00,
NTC    =              1,
TCT    =   .11500000+03,   .11500000+03,   .12000000+03,   .12500000+03,
           .00000000+00,   .00000000+00,   .00000000+00,   .00000000+00,
NTE    =              1,
TET    =   .45000000+02,   .50000000+02,   .00000000+00,   .00000000+00,
```

Figure B-1. Input Data - Concept A

```
                  .00000000+00,   .00000000+00,   .00000000+00,   .00000000+00,
KCR     =                   1,
TBIN    =         .20000000+03,
UAER    =         .11800000+03,
EFFAN   =         .49000000+00,
CPL     =         .21000000+00,
TG      =         .80000000+02,   .00000000+00,   .95000000+02,   .00000000+00,
TW      =         .67000000+02,   .00000000+00,   .75000000+02,   .00000000+00,
NDTE    =                   1,
DTET    =         .10000000+02,   .75000000+01,   .20000000+02,   .12500000+02,
                  .15000000+02,
NDTB    =                   1,
DTBT    =         .75000000+01,   .10000000+02,   .15000000+02,   .00000000+00,
                  .00000000+00,
NDTC    =                   1,
DTCT    =         .10000000+02,   .15000000+02,   .00000000+00,   .00000000+00,
                  .00000000+00,
NTBIN   =                   1,
TBINT   =         .20000000+03,   .00000000+00,   .00000000+00,   .00000000+00,
                  .00000000+00,
NTCIN   =                   2,
TCINT   =         .80000000+02,   .85000000+02,   .00000000+00,   .00000000+00,
                  .00000000+00,
$END
```

Figure B-1 (Continued)

SOLAR POWERED AIR CONDITIONING SYSTEM USING R=11
DRY CONDENSER EMPLOYED RUN ON 28 JUL 75 AT 15:38:47 PAGE 1

STATION/ID	TEMPERATURE DEG F	PRESSURE PSIA	ENTHALPY BTU/LB	FLOW RATE LB/HR	DENSITY LB/CU FT
1	45.0000	8.0000	98.0125	545.3306	.2066
2	164.3327	32.3242	113.4615	545.3306	.7048
3	134.5551	32.3242	109.2925	1881.9654	.7401
4	115.0000	30.7850	31.9975	1881.9654	.0000
5	45.0000	8.4000	31.9975	545.3306	.0000
6	116.1526	90.3184	32.2395	1336.6348	.0000
7	185.0000	86.0175	114.5950	1336.6348	1.9437
8	122.7126	32.3242	107.5917	1336.6348	.7551

HEAT EXCHANGER	HOT FLUID FLO (LB/HR)	HOT FLUID TEMP(F) IN	HOT FLUID TEMP(F) OUT	COLD FLUID FLO (LB/HR)	COLD FLUID TEMP(F) IN	COLD FLUID TEMP(F) OUT	UA (BTU/HR/ (DEG F)	WEIGHT (LB) HX	WEIGHT (LB) FAN	COST (US $) HX	COST (US $) FAN	FAN DP (IN=H2O)	FAN POWER (WATT)	Q (BTU/HR)	WET BULB(F) IN	WET BULB(F) OUT
EVAP	3815.	80.0	55.0	545.	45.0	45.0	.00	35.8	32.6	27.2	42.7	.86	179.0	36000.	67.0	53.4
BOILER	14677.	200.0	192.5	1337.	116.2	185.0	10173.46	75.2	.0	115.1	.0	.00	.0	110079.		
CONDNSR	1882.	134.6	115.0	63246.	95.0	105.0	10082.98	237.4	421.3	180.4	413.7	.41	1445.7	145467.	75.0	75.0

COEF OF PERFORMANCE

POWER COP	.085
REFRIG COP	4.273
SYSTEM COP	.326

TURBO=COMPRESSOR

COMPR DIA(IN)	2.437
COMPR EFF	.708
RPM	58722.
TURBN DIA(IN)	1.822
TURBN EFF	.805

ELECTRIC POWER REQD(WATT)

EVAP FAN	178.997
CONDSR FAN	1445.722
CL TOWER FAN	.000
WATER PUMP	.000
FREON PUMP	94.791
TOTAL	1719.510

SYSTEM COST($)

FACTORY COST	1517.
USER COST	5634.

Figure B-2. Output Data - Concept A

```
                        SOLAR POWERED AIR CONDITIONING SYSTEM USING     R=11
                        PRECOOLER/HUMIDIFIER EMPLOYED

$INPUT
VIST    =    .95000000-02,    .10250000-01,    .11000000-01,    .11600000-01,
             .12250000-01,    .12820000-01,    .13400000-01,    .13900000-01,
             .00000000+00,    .00000000+00,    .00000000+00,    .00000000+00,
             .00000000+00,    .00000000+00,    .00000000+00,
TT      =    .00000000+00,    .40000000+02,    .80000000+02,    .12000000+03,
             .16000000+03,    .20000000+03,    .24000000+03,    .28000000+03,
TTP     =   -.40000000+02,   -.20000000+02,    .00000000+00,    .20000000+02,
             .40000000+02,    .60000000+02,    .80000000+02,    .10000000+03,
             .12000000+03,    .14000000+03,    .16000000+03,    .18000000+03,
             .20000000+03,    .22000000+03,    .24000000+03,    .26000000+03,
             .28000000+03,
HVT     =    .87529999+02,    .89949999+02,    .92419999+02,    .94889999+02,
             .97389999+02,    .99879999+02,    .10235999+03,    .10480999+03,
             .10721999+03,    .10958999+03,    .11187999+03,    .11406999+03,
             .11616999+03,    .11812999+03,    .11991999+03,    .12151999+03,
             .12284999+03,
HLT     =    .00000000+00,    .39800000+01,    .79899999+01,    .12030000+02,
             .16120000+02,    .20270000+02,    .24480000+02,    .28750000+02,
             .33080000+02,    .37480000+02,    .41950000+02,    .46470000+02,
             .51070000+02,    .55760000+02,    .60530000+02,    .65459999+02,
             .70569999+02,
PT      =    .73869999+00,    .14190000+01,    .25540000+01,    .43419999+01,
             .70299999+01,    .10910000+02,    .16310000+02,    .23600000+02,
             .33180000+02,    .45500000+02,    .61010000+02,    .80179999+02,
             .10352999+03,    .13158000+03,    .16487000+03,    .20397000+03,
             .24947000+03,
RHOVT   =    .22600000-01,    .41539999-01,    .71709999-01,    .11739999+00,
             .18350000+00,    .27590000+00,    .40100000+00,    .56630000+00,
             .78009999+00,    .10520000+01,    .13920000+01,    .18140000+01,
             .23330000+01,    .29680000+01,    .37440000+01,    .46960000+01,
             .58800000+01,
CP      =    .14000000+00,
GAMMA   =    .11100000+01,
AK      =    .19800000+02,
TB      =    .16500000+03,
TE      =    .40000000+02,
TC      =    .10500000+03,
MW      =    .13740000+03,
DPP     =    .49999999-01,    .49999999-01,    .49999999-01,    .49999999-01,
             .49999999-01,    .49999999-01,    .49999999-01,    .49999999-01,
             .49999999-01,    .49999999-01,    .49999999-01,    .49999999-01,
             .49999999-01,    .49999999-01,    .49999999-01,
EFF     =    .89999999+00,
QR      =    .34000000+05,
RHOL    =    .91000000+02,
EFPUMP  =    .50000000+00,
TITLE   =    .25618020+18,
NTB     =               1,
TBT     =    .18500000+03,    .18500000+03,    .18000000+03,    .19000000+03,
             .00000000+00,    .00000000+00,    .00000000+00,    .00000000+00,
NTC     =               1,
TCT     =    .10000000+03,    .11500000+03,    .12000000+03,    .12500000+03,
             .00000000+00,    .00000000+00,    .00000000+00,    .00000000+00,
NTE     =               1,
TET     =    .45000000+02,    .50000000+02,    .00000000+00,    .00000000+00,
```

Figure B-3. Input Data - Concept B

```
                .00000000+00,  .00000000+00,  .00000000+00,  .00000000+00,
 KCR     =                3,
 TPIN    =      .20000000+03,
 UAER    =      .11800000+03,
 EFFAN   =      .49000000+00,
 CPL     =      .21000000+00,
 TG      =      .80000000+02,  .55000000+02,  .95000000+02,  .10500000+03,
 TW      =      .67000000+02,  .53368063+02,  .75000000+02,  .75000000+02,
 NDTE    =                1,
 DTET    =      .10000000+02,  .75000000+01,  .20000000+02,  .12500000+02,
                .15000000+02,
 NDTB    =                1,
 DTBT    =      .75000000+01,  .10000000+02,  .15000000+02,  .00000000+00,
                .00000000+00,
 NDTC    =                1,
 DTCT    =      .10000000+02,  .15000000+02,  .00000000+00,  .00000000+00,
                .00000000+00,
 NTBIN   =                1,
 TBINT   =      .20000000+03,  .00000000+00,  .00000000+00,  .00000000+00,
                .00000000+00,
 NTCIN   =                2,
 TCINT   =      .80000000+02,  .85000000+02,  .00000000+00,  .00000000+00,
                .00000000+00,
 $END
```

Figure B-3 (Continued)

SOLAR POWERED AIR CONDITIONING SYSTEM USING R=11
PRECOOLER/HUMIDIFIER EMPLOYED RUN ON 28 JUL 75 AT 15:38:48

STATION/ID	TEMPERATURE DEG F	PRESSURE PSIA	ENTHALPY BTU/LB	FLOW RATE LB/HR	DENSITY LB/CU FT
1	45.0000	8.0000	98.0125	519.7618	.2066
2	137.8871	24.7800	110.0661	519.7618	.5575
3	120.1193	24.7800	107.5787	1327.6260	.5746
4	100.0000	23.6000	28.7500	1327.6260	.0000
5	45.0000	8.4000	28.7500	519.7618	.0000
6	101.2917	90.3184	29.0213	807.8642	.0000
7	185.0000	86.0175	114.5950	807.8642	1.9437
8	111.4324	24.7800	105.9783	807.8642	.5833

HEAT EXCHANGER	HOT FLUID FLO (LB/HR)	HOT FLUID TEMP(F) IN	HOT FLUID TEMP(F) OUT	COLD FLUID FLO (LB/HR)	COLD FLUID TEMP(F) IN	COLD FLUID TEMP(F) OUT	UA (BTU/HR/ (DEG F)	WEIGHT (LB) HX	WEIGHT (LB) FAN	COST (US $) HX	COST (US $) FAN	FAN DP (IN=H2O)	FAN POWER (WATT)	Q (BTU/HR)	WET BULB(F) IN	WET BULB(F) OUT
EVAP	3815.	80.0	55.0	520.	45.0	45.0	.00	35.8	32.6	27.2	42.7	.86	179.0	36000.	67.0	53.4
BOILER	9218.	200.0	192.5	808.	101.3	185.0	6389.15	47.3	.0	72.3	.0	.00	.0	69132.		
CONDNSR	1328.	120.1	100.0	35002.	77.0	90.0	6705.24	157.9	211.5	144.0	221.5	.59	1112.1	104655.	75.0	75.0

COEF OF PERFORMANCE		TURBO=COMPRESSOR		ELECTRIC POWER REQD(WATT)		SYSTEM COST($)	
POWER COP	.101	COMPR DIA(IN)	2.242	EVAP FAN	178.997	FACTORY COST	1069.
REFRIG COP	5.746	COMPR EFF	.725	CONDSR FAN	1112.104		
SYSTEM COP	.519	RPM	61716.	CL TOWER FAN	.000	USER COST	3970.
		TURBN DIA(IN)	1.942	WATER PUMP	104.655		
		TURBN EFF	.789	FREON PUMP	64.206		
				TOTAL	1459.962		

Figure B-4. Output Data - Concept B

```
                    SOLAR POWERED AIR CONDITIONING SYSTEM USING     R-11
                    WET CONDENSER EMPLOYED

$INPUT
VIST     =     .95000000-02,   .10250000-01,   .11000000-01,   .11600000-01,
               .12250000-01,   .12820000-01,   .13400000-01,   .13900000-01,
               .00000000+00,   .00000000+00,   .00000000+00,   .00000000+00,
               .00000000+00,   .00000000+00,   .00000000+00,
TT       =     .00000000+00,   .40000000+02,   .80000000+02,   .12000000+03,
               .16000000+03,   .20000000+03,   .24000000+03,   .28000000+03,
TTH      =    -.40000000+02,  -.20000000+02,   .00000000+00,   .20000000+02,
               .40000000+02,   .60000000+02,   .80000000+02,   .10000000+03,
               .12000000+03,   .14000000+03,   .16000000+03,   .18000000+03,
               .20000000+03,   .22000000+03,   .24000000+03,   .26000000+03,
               .28000000+03,
HVT      =     .87529999+02,   .89949999+02,   .92419999+02,   .94889999+02,
               .97389999+02,   .99879999+02,   .10235999+03,   .10480999+03,
               .10721999+03,   .10958999+03,   .11187999+03,   .11406999+03,
               .11616999+03,   .11812999+03,   .11991999+03,   .12151999+03,
               .12284999+03,
HLT      =     .00000000+00,   .39800000+01,   .79899999+01,   .12030000+02,
               .16120000+02,   .20270000+02,   .24480000+02,   .28750000+02,
               .33080000+02,   .37480000+02,   .41950000+02,   .46470000+02,
               .51070000+02,   .55760000+02,   .60530000+02,   .65459999+02,
               .70569999+02,
PT       =     .73869999+00,   .14190000+01,   .25540000+01,   .43419999+01,
               .70299999+01,   .10910000+02,   .16310000+02,   .23600000+02,
               .33180000+02,   .45500000+02,   .61010000+02,   .80179999+02,
               .10352999+03,   .13158000+03,   .16487000+03,   .20397000+03,
               .24947000+03,
RHOVT    =     .22600000-01,   .41539999-01,   .71709999-01,   .11739999+00,
               .18350000+00,   .27590000+00,   .40100000+00,   .56630000+00,
               .78009999+00,   .10520000+01,   .13920000+01,   .18140000+01,
               .23330000+01,   .29680000+01,   .37440000+01,   .46960000+01,
               .58800000+01,
CP       =     .14000000+00,
GAMMA    =     .11100000+01,
AK       =     .19800000+02,
TB       =     .16500000+03,
TE       =     .40000000+02,
TC       =     .10500000+03,
MW       =     .13740000+03,
DPP      =     .49999999-01,   .49999999-01,   .49999999-01,   .49999999-01,
               .49999999-01,   .49999999-01,   .49999999-01,   .49999999-01,
               .49999999-01,   .49999999-01,   .49999999-01,   .49999999-01,
               .49999999-01,   .49999999-01,   .49999999-01,
EFM      =     .89999999+00,
QR       =     .36000000+05,
RHOL     =     .91000000+02,
EFPUMP   =     .50000000+00,
TITLE    =     .25618020+18,
NTB      =                1,
TBT      =     .18500000+03,   .18500000+03,   .18000000+03,   .19000000+03,
               .00000000+00,   .00000000+00,   .00000000+00,   .00000000+00,
NTC      =                1,
TCT      =     .90000000+02,   .11500000+03,   .12000000+03,   .12500000+03,
               .00000000+00,   .00000000+00,   .00000000+00,   .00000000+00,
NTE      =                1,
TET      =     .45000000+02,   .50000000+02,   .00000000+00,   .00000000+00,
```

Figure B-5. Input Data - Concept C

```
                 .00000000+00,   .00000000+00,   .00000000+00,   .00000000+00,
KCR      =                  2,
TBIN     =       .20000000+03,
UAER     =       .11800000+03,
EFFAN    =       .49000000+00,
CPL      =       .21000000+00,
TG       =       .80000000+02,   .00000000+00,   .95000000+02,   .00000000+00,
TH       =       .67000000+02,   .00000000+00,   .75000000+02,   .00000000+00,
NDTE     =                  1,
DTET     =       .10000000+02,   .75000000+01,   .20000000+02,   .12500000+02,
                 .15000000+02,
NDTB     =                  1,
DTBT     =       .75000000+01,   .10000000+02,   .15000000+02,   .00000000+00,
                 .00000000+00,
NDTC     =                  1,
DTCT     =       .10000000+02,   .15000000+02,   .00000000+00,   .00000000+00,
                 .00000000+00,
NTBIN    =                  1,
TBINT    =       .20000000+03,   .00000000+00,   .00000000+00,   .00000000+00,
                 .00000000+00,
NTCIN    =                  2,
TCINT    =       .80000000+02,   .85000000+02,   .00000000+00,   .00000000+00,
                 .00000000+00,
$END
```

Figure B-5 (Continued)

SOLAR POWERED AIR CONDITIONING SYSTEM USING R-11 RUN ON 28 JUL 75 AT 18:14:38 PAGE 1
WET CONDENSER EMPLOYED

STATION/ID	TEMPERATURE DEG F	PRESSURE PSIA	ENTHALPY BTU/LB	FLOW RATE LB/HR	DENSITY LB/CU FT
1	45.0000	8.0000	98.0125	504.2193	.2066
2	122.1692	20.9527	108.0408	504.2193	.4807
3	110.8671	20.9527	106.4585	1096.6669	.4902
4	90.0000	19.9550	26.6150	1096.6669	.0000
5	45.0000	8.4000	26.6150	504.2193	.0000
6	91.3622	90.3184	26.9011	592.4476	.0000
7	185.0000	86.0175	114.5950	592.4476	1.9437
8	101.2480	20.9527	105.1118	592.4476	.4986

HEAT EXCHANGER	HOT FLUID FLO (LB/HR)	HOT FLUID TEMP(F) IN	HOT FLUID TEMP(F) OUT	COLD FLUID FLO (LB/HR)	COLD FLUID TEMP(F) IN	COLD FLUID TEMP(F) OUT	UA (BTU/HR/(DEG F)	WEIGHT (LB) HX	WEIGHT (LB) FAN	COST (US $) HX	COST (US $) FAN	FAN DP (IN-H2O)	FAN POWER (WATT)	Q (BTU/HR)	WET BULB(F) IN	WET BULB(F) OUT
EVAP	3815.	80.0	55.0	504.	45.0	45.0	.00	35.8	32.6	27.2	42.7	.86	179.0	38000.	67.0	53.4
BOILER	6927.	200.0	192.5	592.	91.4	185.0	4801.57	35.5	.0	54.3	.0	.00	.0	51954.		
CONDNSR	1097.	110.9	90.0	18242.	95.0	******	.00	91.8	108.4	109.8	124.3	.82	831.0	87562.	75.0	80.0

COEF OF PERFORMANCE	
POWER COP	.108
REFRIG COP	7.120
SYSTEM COP	.691

TURBO-COMPRESSOR	
COMPR DIA(IN)	2.176
COMPR EFF	.736
RPM	58444.
TURBN DIA(IN)	2.176
TURBN EFF	.771

ELECTRIC POWER REQD(WATT)	
EVAP FAN	178.997
CONDSR FAN	830.980
CL TOWER FAN	.000
WATER PUMP	87.562
FREON PUMP	49.658
TOTAL	1147.197

SYSTEM COST($)	
FACTORY COST	822.
USER COST	3055.

Figure B-6. Output Data - Concept C

```
SOLAR POWERED AIR CONDITIONING SYSTEM USING    R-11
COOLING TOWER EMPLOYED

$INPUT
VIST   =   .95000000-02,   .10250000-01,   .11000000-01,   .11600000-01,
           .12250000-01,   .12820000-01,   .13400000-01,   .13900000-01,
           .00000000+00,   .00000000+00,   .00000000+00,   .00000000+00,
           .00000000+00,   .00000000+00,   .00000000+00,
TT     =   .00000000+00,   .40000000+02,   .80000000+02,   .12000000+03,
           .16000000+03,   .20000000+03,   .24000000+03,   .28000000+03,
TTH    =  -.40000000+02,  -.20000000+02,   .00000000+00,   .20000000+02,
           .40000000+02,   .60000000+02,   .80000000+02,   .10000000+03,
           .12000000+03,   .14000000+03,   .16000000+03,   .18000000+03,
           .20000000+03,   .22000000+03,   .24000000+03,   .26000000+03,
           .28000000+03,
HVT    =   .87529999+02,   .89949999+02,   .92419999+02,   .94889999+02,
           .97389999+02,   .99879999+02,   .10235999+03,   .10480999+03,
           .10721999+03,   .10958999+03,   .11187999+03,   .11406999+03,
           .11616999+03,   .11812999+03,   .11991999+03,   .12151999+03,
           .12284999+03,
HLT    =   .00000000+00,   .39600000+01,   .79899999+01,   .12030000+02,
           .16120000+02,   .20270000+02,   .24480000+02,   .28750000+02,
           .33080000+02,   .37480000+02,   .41950000+02,   .46470000+02,
           .51070000+02,   .55760000+02,   .60530000+02,   .65459999+02,
           .70569999+02,
PT     =   .73869999+00,   .14190000+01,   .25540000+01,   .43419999+01,
           .70299999+01,   .10910000+02,   .16310000+02,   .23600000+02,
           .33180000+02,   .45500000+02,   .61[illegible]10000+02,   .80179999+02,
           .10352999+03,   .13158000+03,   .16487000+03,   .20397000+03,
           .24947000+03,
RHOVT  =   .22600000-01,   .41539999-01,   .71709999-01,   .11739999+00,
           .18350000+00,   .27590000+00,   .40100000+00,   .56630000+00,
           .78009999+00,   .10520000+01,   .13920000+01,   .18140000+01,
           .23330000+01,   .29680000+01,   .37440000+01,   .46960000+01,
           .58800000+01,
CP     =   .14000000+00,
GAMMA  =   .11100000+01,
AK     =   .19800000+02,
TB     =   .16500000+03,
TE     =   .40000000+02,
TC     =   .10500000+03,
MW     =   .13740000+03,
DPP    =   .49999999-01,   .49999999-01,   .49999999-01,   .49999999-01,
           .49999999-01,   .49999999-01,   .49999999-01,   .49999999-01,
           .49999999-01,   .49999999-01,   .49999999-01,   .49999999-01,
           .49999999-01,   .49999999-01,   .49999999-01,
EFM    =   .89999999+00,
QR     =   .36000000+05,
RHOL   =   .91000000+02,
EFPUMP =   .50000000+00,
TITLE  =   .25618020+18,
NTB    =             1,
TBT    =   .18500000+03,   .18500000+03,   .18000000+03,   .19000000+03,
           .00000000+00,   .00000000+00,   .00000000+00,   .00000000+00,
NTC    =             1,
TCT    =   .90000000+02,   .95000000+02,   .11500000+03,   .12500000+03,
           .00000000+00,   .00000000+00,   .00000000+10,   .00000000+00,
NTE    =             1,
TET    =   .45000000+02,   .50000000+02,   .00000000+00,   .00000000+00,
```

Figure B-7. Input Data - Concept D

```
                  .00000000+00,    .00000000+00,    .00000000+00,    .00000000+00,
 KCR      =                 4,
 TRIN     =       .20000000+03,
 UAER     =       .11600000+03,
 EFFAN    =       .49000000+00,
 CPL      =       .21000000+00,
 TG       =       .80000000+02,    .55000000+02,    .95000000+02,    .95000000+02,
 TH       =       .67000000+02,    .53368063+02,    .75000000+02,    .75000000+02,
 NDTE     =                 1,
 DTET     =       .10000000+02,    .15000000+02,    .20000000+02,    .12500000+02,
                  .15000000+02,
 NDTB     =                 1,
 DTBT     =       .75000000+01,    .10000000+02,    .20000000+02,    .00000000+00,
                  .00000000+00,
 NDTC     =                 1,
 DTCT     =       .50000000+01,    .15000000+02,    .10000000+02,    .15000000+02,
                  .20000000+02,
 NTBIN    =                 1,
 TBINT    =       .20000000+03,    .00000000+00,    .00000000+00,    .00000000+00,
                  .00000000+00,
 NTCIN    =                 1,
 TCINT    =       .80000000+02,    .65000000+02,    .90000000+00,    .00000000+00,
                  .00000000+00,
 $END
```

Figure B-7 (Continued)

SOLAR POWERED AIR CONDITIONING SYSTEM USING R-11
COOLING TOWER EMPLOYED

RUN ON 24 JUL 75 AT 18:20:12

STATION/ID	TEMPERATURE DEG F	PRESSURE PSIA	ENTHALPY BTU/LB	FLOW RATE LB/HR	DENSITY LB/CU FT
1	45.0000	8.0000	98.0125	504.2193	.2066
2	122.1692	20.9527	108.0008	504.2193	.4007
3	110.8670	20.9527	106.4585	1096.6669	.4702
4	90.0000	19.9550	26.6150	1096.6669	[illegible]
5	45.0000	8.4000	26.6150	504.2193	[illegible]
6	91.3622	90.3184	26.9611	592.4476	[illegible]
7	185.0000	86.0175	114.5950	592.4476	1.9437
8	101.2480	20.9527	105.1118	592.4476	.4986

HEAT EXCHANGER	HOT FLUID FLO (LB/HR)	HOT FLUID TEMP(F) IN	HOT FLUID TEMP(F) OUT	COLD FLUID FLO (LB/HR)	COLD FLUID TEMP(F) IN	COLD FLUID TEMP(F) OUT	UA (BTU/HR/DEG F)	WEIGHT (LB) HX	WEIGHT (LB) FAN	COST (US $) HX	COST (US $) FAN	FAN DP (IN-H2O)	FAN POWER (WATT)	Q (BTU/HR)	WET BULB(F) IN	WET BULB(F) OUT
EVAP	3815.	80.0	55.0	504.	45.0	45.0	.00	35.8	32.6	27.2	42.7	.86	179.0	36000.	67.0	53.4
BOILER	6927.	200.0	192.5	592.	91.4	185.0	4801.57	35.5	.0	54.3	.0	.00	.0	51954.		
CONDNSR	1097.	110.9	90.0	17512.	80.0	85.0	12138.63	69.8	.0	344.2	.0	.00	.0	87562.	75.0	75.0

COEF OF PERFORMANCE	
POWER COP	.108
REFRIG COP	7.120
SYSTEM COP	.691

TURBO-COMPRESSOR	
COMPR DIA(IN)	2.176
COMPR EFF	.736
RPM	58444.
TURBN DIA(IN)	2.176
TURBN EFF	.771

ELECTRIC POWER REQD(WATT)	
EVAP FAN	178.997
CONDSR FAN	.000
CL TOWER FAN	259.183
WATER PUMP	166.367
FREON PUMP	49.658
TOTAL	654.205

SYSTEM COST($)	
FACTORY COST	1004.
USER COST	3730.

Figure B-8. Output Data - Concept D