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**DEVELOPMENT OF A HIGH TEMPERATURE SOLAR POWERED WATER
CHILLER. VOLUME 3**

Phase I Technical Progress Report, September 26, 1977—June 1, 1978

By
Richard A. English

June 1978

Work Performed Under Contract No. EG-77-C-03-1590

Energy Systems Division
Carrier Corporation
Syracuse, New York

U.S. Department of Energy

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Solar Energy

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DEVELOPMENT OF A HIGH TEMPERATURE
SOLAR POWERED WATER CHILLER

VOLUME III

PHASE I TECHNICAL PROGRESS REPORT
FOR THE PERIOD
SEPTEMBER 26, 1977 - JUNE 1, 1978

RICHARD A. ENGLISH

JUNE 1978

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Work performed under Contract EG-77-C-03-1590

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1.4 CHILLER DESIGN

Under this task, the overall chiller package was conceptualized. Based on results of the cycle study, Task 1.2, R-113 was chosen as the working fluid and the optimum heat exchange configuration was selected. This work included detailed evaluation of heat exchange and design of all major heat exchangers; evaluation of liquid pumps and detailed design of two alternate approaches; layouts of the total package; estimates of manufacturing costs; control logic; and prediction of the performance of the machine under a variety of conditions.

The resulting conceptual design has been optimized from a cost/performance standpoint. It meets all of the initial product specifications that can be determined at this stage, and holds promise of exceeding the performance goals. Costs appear to be in line with other heat operated equipment.

This section describes the conceptual design rationale and resulting design configuration as well as providing estimates of cost and performance. Because the development of the turbo-compressor design, Task 1.3, paralleled the development of the chiller system design, all of the cost and performance data are based on intermediate turbo compressor performance data, as well as on unoptimized components. Optimized performance was computed at the very end of Phase I, and only a brief comparison is made to show the potential gains available. Updating of all the predicted performance data for the final conceptual design will await Phase II.

1.4.1 HEAT EXCHANGER DESIGN

Development of optimum heat exchangers for this type of equipment is critical to the success of the project. Heat exchangers may account for 90% of the bulk and weight of the unit and over 50% of the cost. The degree of approach to the ideal cycle performance is a direct function of the efficiency of the heat exchange processes. When considering total solar system cost, it can be shown that there is a very favorable cost trade-off with the relatively expensive solar collectors, as the overall COP is increased by improving heat exchanger effectiveness.

Under this task, each heat exchanger was designed in detail. Alternate heat exchange concepts were evaluated to find the most effective for this application. Heat exchange designs were based on data from preliminary R-113 cycle computations with optimized heat exchangers reported in Section 1.2.6.

1.4.1.1 Design of the Condenser

The objective of the following set of calculations was to design a condenser system for gaseous R-113 refrigerant leaving the power cycle turbine and the refrigeration cycle compressor stages. The condenser system is to have a total heat transfer capacity of 28,516 Btuh/ton = 712,900 Btuh under the following physical conditions.

1. 95°F entering air temperature
2. A fan cfm of 1560 cfm/ton = 39,000 cfm
3. A refrigeration cycle load of 14,630 Btuh/ton = 365,750 Btuh
4. A Rankine power cycle load of 13,886 Btuh/ton = 347,150 Btuh
5. A refrigeration cycle refrigerant mass flow rate of 198.861 lb m/hr-ton = 4,971.525 lb m/hr
6. A Rankine cycle refrigerant mass flow rate of 191.279 lb m/hr-ton = 4,781.975 lb m/hr
7. Estimated condensing temperature and pressure for a common condenser = 115.3°F and 14.10 psia (this value was obtained by averaging the condensing temperatures of the optimized series design given in Table 1.2-13, Run 455).

In addition to these parameters, there were a number of very practical design oriented factors that had to receive consideration prior to system formulation. These considerations are listed below.

1. In order to minimize development costs and in an effort to utilize a documented and reliability-proven configuration, effort should be made to build the condenser system into a current Carrier Corporation production machinery system housing. A likely candidate would be to utilize the 30GB 055 condenser housing assembly.
2. In order to design a simple chiller cycle with a minimum of controls and to eliminate the need for shaft seals, a single common condenser will be used instead of the more efficient series condenser cycle described in Section 1.2.4.

3. In order to minimize heat transfer resistance and refrigerant side pressure drop, the condenser coil will be constructed from 1/2 inch O.D. copper tubes with corrugated-geometry aluminum fins at a fin spacing of 14 fins per inch..
4. To facilitate draining of refrigerant condensate, the circuits will be sloped in the downward vertical direction (see Figure 1.4-2).

Once the physical design conditions have been stated, one needs to consider the pertinent heat transfer related considerations. In general, these factors are parameters which directly affect the particular construction geometries employed. These factors are as follows:

1. In order to keep the refrigerant-side pressure drop low, the condenser design should employ a large number of parallel refrigerant circuits.
2. In order to maintain refrigerant load balance between circuits, the vapor should enter in a thermal counterflow arrangement. This fact, combined with Item 1 above, suggests a 4-row deep condenser coil with headers and return bends as illustrated in Figure 1.4-2.

Once the above heat transfer and design factors were considered, the actual heat exchanger sizing procedure was carried out. The first step was to perform an optimization study of the condenser coil face area. This would supply information related to target air flow rates, refrigerant saturation temperature drops and average refrigerant condensing heat transfer coefficients. This was achieved by fixing the entering air temperature, the entering temperature of the condensing vapor, the average linear velocity of the air stream passing through the coil and the tube-fin geometry. Next a series of heat transfer solutions were generated for a variety of coil circuit lengths. The heat transfer perturbation study was performed with the aid of a proprietary Carrier Corporation condenser coil computer simulation program. This program performs incremental tube-by-tube calculations and recognizes variations in air and refrigerant side heat transfer coefficients and temperatures. The program correlations are based on proprietary Carrier Corporation air and refrigerant-side heat transfer data. The results of this analysis are plotted in Figure 1.4-1. This shows the maximum Q/A occurs at a coil length of about 100 inches.

Once the optimum circuit length was achieved, computer solutions allowing perturbations on coil height (and thus the number of circuits), face velocity and mean condensing temperature were performed, with the aforementioned proprietary program. This time the system restraint was a capacity of 712,900 Btuh. The results of this study can be found in Figure 1.4-3 and are summarized in Table 1.4-1.

FIGURE 1.4 - 1

R-113 CONDENSER-REFRIGERANT CYCLE

TWO PASS, COUNTERFLOW

ENT. AIR TEMP. = 95.0 °F

AIR FACE VELOCITY = 342.1 FT/MIN
1/2 O.D., 1.25 X 1.08, 14 FPI

FIGURE 1.4 - 2
CONDENSER COIL CIRCUITING ARRANGEMENT

FIGURE 1.4 - 3

R-113 CONDENSER

TWO PASS, COUNTERFLOW
 ENT. AIR TEMP. = 95.0°F
 1/2" O.D. TUBES, 1.25 X 1.08, 14 FPI
 Q = 712,900 BTU/H
 LENGTH BETWEEN TUBE SHEETS = 100 IN.

TABLE 1.4-1 CCNDENSER DESIGN SUMMARY TABLE

Condensing Temperature	Tc = 116°F				Tc = 117°F				UNITS
Face Velocity	300	320	340	360	300	320	340	360	ft/min
Heat Flux(q/A)	5,579	5,844	6,099	6,344	5,845	6,122	6,390	6,647	BTU/hr-ft ²
Surface Area	127.8	122.0	116.9	112.4	122.0	116.4	111.6	107.3	ft ²
Air Flow Rate	38,330	39,040	39,740	40,450	36,590	37,260	37,930	38,610	ft ³ /min
Heat Flux/tube	4,843	5,073	5,294	5,507	5,074	5,315	5,547	5,770	BTU/hr-tube
Total Number of Tubes	147	140	135	130	140	134	128	124	
Air Pressure Drop	.216	.242	.268	.296	.217	.242	.268	.296	psi

The final coil selection used for preliminary costing and layout was based on 128 tubes in the face, and 101 inches length. This resulted in a face velocity of 347.5 fpm and an estimated condensing temperature of 116.7°F. This point is shown on Figure 1.4-3. Figure 1.4-4 shows this preliminary coil design which was costed. It is based on a layout that divides the coil into four equal slabs. Data on tube and fin pattern are given.

1.4.1.2 Design of the Water Cooler

The objective of these calculations was to design an R-113-to-water 25 ton (300,000 Btuh) water cooler (refrigerant evaporator) under the following design conditions.

1. Temperature of H_2O leaving cooler = 45°F
2. Water flow rate = $\dot{m}_{H_2O}$ = 2.5 gpm/ton = 62.5 gpm
3. R-113 flow rate = $\dot{m}_{ref}$ = 198.861 lb m/hr-ton = 4971.525 lb m/hr
4. Refrigerant evaporation (suction) temperature = 40°F

In addition to these parameters, there were a number of practical design considerations. The major design considerations are listed below.

1. The differences between cooler temperature and maximum shell temperature were such that differential tube-to-shell transient thermal expansion could be neglected. Therefore, a fixed tube sheet shell-and-tube heat exchanger design was selected.
2. At the temperature conditions expected, copper tubing and steel shell materials can be safely used with R-113.
3. Preliminary calculations showed that significant reduction in overall heat transfer resistance could be achieved by employing tubing that possessed both external and internal fins for heat transfer enhancement.
4. In order to avoid refrigerant circuit distribution problems, a flooded cooler design was chosen. Hence the R-113 is the shell-side fluid with H_2O flowing in the tubes. This configuration also expedites cleaning of the heat exchanger surfaces.
5. The candidate cooler designs should avoid large pressure drops on the water side and minimize submergence effects on the refrigerant side.
6. The preliminary calculations for the cooler (and the boiler) assumed that the R-113 leaves the shell as saturated vapor. This implies that the R-113 temperature change through the cooler is practically zero. As can be found in Holman (H-2), this implies that the log mean temperature correction factor, F_G , is unity. This approximation would result in a maximum log mean temperature difference (LMTD) error of 4 percent. This difference is more than compensated by oversizing and by the use of conservative fouling factors and boiling coefficients.

FIGURE 1.4 - 4

PRELIMINARY CONDENSER SLAB DESIGN

TUBES: (34) IN FACE, (136) TOTAL, $1/2$ " O.D. COPPER

TUBE PATTERN: $1.25" \times 1.08"$

FINS: (1414) CORRUGATED PLATE ~ ALUMINUM .0065" THICK, 14 FINS/INCH

CIRCUITS: (68)

Finally, a number of heat transfer and fluid flow considerations had to be made. In general these deal with the methods of approach to heat exchanger sizing and specific assumptions made to calculate the pertinent heat transfer coefficients. These considerations are listed below.

1. The heat exchanger tubes are internally finned, which makes it difficult to evaluate the magnitude of tube-side fouling.

A value of

$$0.0005 \frac{\text{hr-ft}^2\text{-}^\circ\text{F}}{\text{Btu}}$$

was chosen for the fouling resistance, based on the internal tube surface area.

2. The candidate tubing is manufactured by the Wolverine Division of UOP, Inc. It is a nominal 0.75 inch O.D. tube with fins per inch of 0.035 inch fin height on the external surface. Additionally, there are fins on the tube I.D. The metal resistance associated with this tubing is approximately

$$0.000035 \frac{\text{hr-ft}^2\text{-}^\circ\text{F}}{\text{Btu}}$$

3. The heat transfer coefficient and pressure drop of the water flowing in the tubes is obtained from the following general equations.

For the water-side heat transfer coefficient, a modified Dittus-Boelter equation of the following general form was used.

$$h = \frac{A (1 + 0.11 T_{H_2O}) V^{0.8}}{D^{0.2}} \quad (1)$$

Where V is linear water velocity in ft/sec., T_{H_2O} is the mean water temperature in $^\circ\text{F}$, D is the interior tube diameter in inches and A is a constant determined from tube performance tests.

The water-side pressure drop is determined by the following equation.

$$\Delta P = f \left(\frac{L}{D} \right) \left(\frac{\rho}{g_c} \right) \left(\frac{V^2}{2} \right) \quad (2)$$

Where Δp is the pressure drop, L is tube length, ρ is the fluid density and f is the friction factor which can be expressed as:

$$f = A/R_e^B \quad (3)$$

Where R_e is the fluid Reynolds number and A and B are empirical coefficients used to characterize the particular internal tube geometry.

4. Theoretical predictions of heat transfer coefficients for nucleate boiling are difficult to formulate. This problem is complicated by the use of externally finned tubing. To avoid these problems, boiling coefficients were obtained from physical test data of R-113 on a finned-tube bundle (at specific reference temperatures) as a function of cooler load. Saturation pressure correction factors are used to extrapolate these coefficients from the reference temperatures to the actual suction conditions.
5. Since R-113 is on the shell side, it is assumed that the shell side fouling resistance is negligible.
6. To minimize pressure drop and fouling resistance, the linear tube-side velocity should lie between 3 and 10 ft/sec.

Once the above factors were established, the actual heat exchanger performance could be computed. The equation employed to size the water cooler was of the classical form,

$$q = U_o A_o F_G (\text{LMTD}) \quad (4)$$

where q is the heat transfer rate in BTUH, A is the external tube surface area, U_o is the overall heat transfer coefficient, F_G is the LMTD correction factor (assumed to equal 1.0 per previous discussion) and LMTD is the log mean temperature difference.

The overall heat transfer coefficient, U_o , was calculated as follows:

$$U_o = \frac{1}{\left(\frac{1}{h_{H_2O}} + R_m + R_f \right) SR + \frac{1}{h_{ref}}} \quad (5)$$

In this equation, the term in parentheses is the sum of the water-side heat transfer convection coefficient, the metal resistance and the fouling resistance all based on the internal surface area. The term SR is the ratio of external to internal surface area, and h_{ref} is the refrigerant boiling coefficient.

The value of LMTD is calculated by the following equation.

$$\text{LMTD} = \frac{(T_{H_2O_2} - T_{ref}) - (T_{H_2O_1} - T_{ref})}{\ln \left[\frac{(T_{H_2O_2} - T_{ref})}{(T_{H_2O_1} - T_{ref})} \right]} \quad (6)$$

Here, $T_{H_2O_2}$ and $T_{H_2O_1}$ are leaving and entering water temperatures, respectively, and T_{ref} is the refrigerant suction temperature.

With these factors, the required external tube surface area simply becomes:

$$A_o = q/V_o \text{ (LMTD) } \quad (7)$$

Using the above information, several preliminary cooler designs were evolved and are listed in Table 1.4-2.

The selection used for preliminary costing and layout was the 20 circuit, 4 pass design in Table 1.4-2. The selection was made to achieve a combined effect of low water side pressure drop, short shell length, and low tube bundle size for minimum submergence effect. The preliminary design which was costed and used in the layout is shown on Figure 1.4-5.

1.4.1.3 Design of the Liquid Preheater

Approximately 50% of the heat necessary to boil the working fluid in the R-113 Rankine cycle is supplied as sensible heat to raise the liquid to the boiling point. The most effective way to accomplish the total boiling process is to separate the sensible heating and the boiling processes and design heat exchangers specifically for each. The preheater was designed to sensibly heat the liquid R-113 returning to the boiler to its boiling point.

The preliminary design studies of the solar system which would be used in the simulation program indicated that the heat transfer medium might well be a glycol solution. It is known that the thermophysical properties of glycol result in lower heat transfer coefficients than obtained with plain water. The preheater and boiler have been designed for a 40% ethylene glycol solution to be most realistic. This results in a conservative design for water. Following is the list of specific preheater design conditions.

1. Temperature of R-113 leaving the preheater = 290°F
2. Temperature of ethylene glycol solution entering preheater = 294.4°F
3. Glycol flow rate, $\dot{m}_{\text{glycol}} = 39,475 \text{ lb m/hr}$
4. Refrigerant mass flow rate, $\dot{m}_{\text{ref}} = 4781.975 \text{ lb m/hr}$
5. Heat transfer rate = 153,126 Btu/hr

There are a number of heat exchanger designs that are capable of meeting these conditions. Selection of the specific heat exchanger type is dependent upon the following design factors.

1. At the high temperature levels achieved in the preheater, the thermal stability of R-113 will be acceptable if all of the surfaces it contacts are steel.
2. It is very important to design a preheater that exhibits minimal refrigerant pressure drop: Therefore, initial construction considerations will focus on a shell-and-tube heat exchanger with R-113 on the shell side.

TABLE 1.4-2

PRELIMINARY EVAPORATOR DESIGN						
NO. OF CIRCUITS	15		20			UNITS
No. of Passes	6	6	4	6	8	
External Surface Area	360	360	390	390	390	ft ² - outside
Length/Circuit Pass	6.73	5.05	8.2	5.47	4.10	ft - length
Total Linear Ft. Tube	606	606	656	656	656	ft - length
Water-Side Pressure Drop (Tubing)	6.05	6.05	2.99	2.99	2.99	psi
Water-Side Pressure Drop (Reversals)	1.28	1.71	.42	.72	.96	psi
Water-Side Total Pressure Drop	7.33	7.76	3.41	3.71	3.95	psi
No. of Tubes	90	120	80	120	160	
Water-Side Coefficient	1826	1826	1451	1451	1451	$\frac{\text{BTU}}{\text{hr} - \text{ft}^2_{\text{i.s.}} - ^\circ\text{F}}$
Refrigerant-Side Coefficient	165	165	154	154	154	$\frac{\text{BTU}}{\text{hr} - \text{ft}^2_{\text{o.s.}} - ^\circ\text{F}}$
Fouling Factor	.0005	.0005	.0005	.0005	.0005	$\frac{\text{BTU}}{\text{hr} - \text{ft}^2_{\text{o.s.}} - ^\circ\text{F}}$
R-113 percent of Resistance	60	60	59	59	59	
Linear Water Velocity	4.60	4.60	3.45	3.45	3.45	$\frac{\text{ft}}{\text{sec}}$

FIGURE 1.4 - 5

PRELIMINARY WATER COOLER DESIGN

3. Shell-side sensible heat transfer coefficients are significantly smaller than their nucleate boiling counterparts. As a result, there is no reason to employ internally finned tubing to boost the water side heat transfer coefficient.
4. The maximum difference between the ambient temperature and the operating temperature is on the order of 200°F. Under transient start-up or shut-down conditions, differential thermal expansion would ruin normal end sheet to tube connections. Therefore, the design conditions call for a floating u-tube construction technique.
5. To maximize external surface area per unit length of tubing, externally finned tubing was chosen. The aforementioned computer program was restricted to a maximum of 19 external fins/inch. Therefore, all sizing calculations are based on this tubing.
6. All values of constants and physical properties associated with a 40 percent ethylene-glycol solution are extracted from the ASHRAE Handbook of Fundamentals (A-6).
7. There is some uncertainty as to the proper value for the internal surface fouling factor for glycol at 290 to 300°F. A value of $0.001 \frac{\text{hr-ft}^2\text{-}^\circ\text{F}}{\text{Btu}}$ is recommended. As pointed out in section 1.4.1.4, this value may be conservative.

Heat transfer and pressure drop calculations for all liquid shell-side flow are very approximate. Therefore, to most accurately obtain this information, a proprietary shell-and-tube heat exchanger computer program written by Heat Transfer Research Inc. (HTRI) was employed in this design.

The basic program operation uses the hot and cold fluid process conditions (i.e., temperature and flow rates) to calculate a thermal load duty. Upon specifying a variety of geometric parameters related to construction techniques, the program calculates overall heat transfer coefficients, LMTD's, effective temperature difference correction factors, and eventually, the ultimate required external surface area. In theory this is very similar to the approach outlined in Section 1.4.1.2. Realistically it has the advantage of employing very small step sizes and iterative calculations to account for the physical property variations that can appear with the R-113 process flow in the preheater. The program also clearly specifies the necessary construction details and allows for perturbation analyses of both physical and geometrical factors. Results of such perturbation studies are listed in Table 1.4-3. At this point, it should be reiterated that the choice of a shell-and-tube preheater was to minimize the R-113 pressure drop. This does not prohibit the consideration of alternately constructed exchangers, such as tube-in-tube, in later phases of the program.

The two shell, series design, with tubes spaced 1/8" apart (center column, Table 1.4-3) was chosen for cost estimating. The two shell, series design, more closely approximates counterflow heat exchange than a single tube-in-shell heat exchanger, and results in about 1/3 the surface. The

TABLE 1.4-3

PRELIMINARY PREHEATER DESIGN DATA

(Employing 19 fpi, internally smooth steel tubing)

PREHEATER TYPE	SINGLE SHELL	SERIES SHELLS	SERIES SHELLS	UNITS
Shell code (type)	TEMA E	TEMA E	TEMA E	
No. of Shells	1	2	2	
No. of Shell passes/shell	1	1	1	
No. of u-tube passes/shell	2	2	2	
No. of vertical baffles/shell	10	6	6	
No. of tube sheet holes/shell	54	54	48	
Length of tube ligament	3/16	1/8	3/16	inches
Tube pitch	.938	.875	.938	
Linear glycol velocity	3.75	3.75	4.22	ft/sec
Tube-side pressure drop				
Shell	1.24	.595	.78	psi
Shell-side velocity	.06	.13	.12	ft/sec
Shell-side pressure drop/shell	.35	.40	.35	psi
U_o (fouled)	31.31	33.54	32.32	BTU
				hr-ft ² o.s. - °F
Total surface area/shell	219	69	70.5	ft ²

TABLE 1.4-3

(PAGE 2)

PRELIMINARY PREHEATER DESIGN DATA

(Employing 19 fpi, internally smooth steel tubing)

PREHEATER TYPE	SINGLE SHELL	SERIES SHELLS	SERIES SHELLS	UNITS
Shell length/shell	8.071	2.617	3.017	ft
Shell diameter	10.071	8.071	8.071	inches
Shell side % of resistance	77.38	75.77	77.61	%
Fouling factor	.001	.001	.001	$\frac{\text{hr-ft}^2\text{.s.} - ^\circ\text{F}}{\text{BTU}}$
Thermal counter flow effectiveness	.62	.98	.98	
Overdesign factor	19	20	12	%

1/8" ligament tube placement is a Carrier standard production technique and results in a more compact design. Figure 1.4-6 shows the preliminary design of one of the two series heat exchangers which was used for costing and layout.

1.4.1.4 Design of the Boiler

The objective of these calculations was to design the heat exchanger to boil R-113, which enters from the preheater at its boiling point. As explained in Section 1.4.1.3, the boiler has been designed for a 40% ethylene glycol solution to be realistic. The boiler design conditions are listed below:

1. The glycol solution enters the boiler at 300°F.
2. The glycol mass flow rate is $m_g = 39,475$ lb m/hr
3. The R-113 flow rate is, $m_{ref} = 8 \times 4781.975$ lb m/hr
4. The saturated boiling temperature is 294.41°F
5. The boiler heat transfer rate is 209,632 Btuh

A number of practical design considerations are listed below.

1. The difference between the ambient temperature and the operating shell temperature is on the order of 200°F. Under transient start-up or shut-down conditions, differential thermal expansion would ruin normal end sheet to tube connections. Therefore, the design conditions call for a floating u-tube construction technique.
2. At the elevated temperatures expected in the boiler, the thermal stability of R-113 will be acceptable if all of the surfaces it contacts are steel.
3. Preliminary calculations showed that significant reduction in tubing length could be achieved by employing tubing that possesses both external and internal fins for heat transfer enhancement.
4. The bend area will be left unfinned to facilitate small radius bends.
5. As was the case of the water cooler, a flooded boiler design was chosen to minimize fouling and refrigerant distribution problems.
6. The boiler was designed to minimize glycol pressure drop.
7. The heat exchanger LMTD correction factor was assumed to equal 1.0 (see item 6 of the design considerations in Section 1.4.1.2 for a more detailed discussion of this approximation).

- 9 TUBES ON 2.880" CENTERS
 8 TUBES ON 3.718" CENTERS
 7 TUBES ON 4.178" CENTERS
 3 TUBES ON 6.636" CENTERS

TUBE SHEET 13 1/2" O.D. x 1" THK.

TUBE DATA: 3/4" O.D. WILCOX (SQUARE 18)
 19 FINS PER INCH
 27 CIRCUITS

FIGURE 1.4 - 6
 PRELIMINARY PREHEATER DESIGN

CARRIER CORPORATION	
ENERGY SYSTEMS DIVISION	
DATE: 5/21/78	BY: J. R. GALT
PREHEATER	
DRAWING NO. 04-0003	

In addition to physical design factors, there were a number of heat transfer parameters pertinent to the ultimate design of the flooded boiler. Values of constants and physical properties for the tube side 40% ethylene glycol solution were taken from the ASHRAE Handbook of Fundamentals (A-6). The important heat transfer design factors are as follows:

1. There is some uncertainty as to the value of the glycol fouling factor for a 300°F solution in internally finned tubes. The majority of heat transfer design texts recommended a value of $0.001 \frac{\text{hr-ft}^2\text{-}^\circ\text{F}}{\text{Btu}}$ based on internal tube surface area. The large fouling factor is primarily due to the kinetic fouling processes associated with high temperature tube flow, and is generally considered to be slightly conservative.
2. The candidate tubing is similar to the internally and externally finned chiller design except that it will be steel. The metal resistance associated with the change to steel is $0.00035 \frac{\text{hr-ft}^2\text{-}^\circ\text{F}}{\text{Btu}}$ when based on internal tube surface area.
3. The general form of the equations employed to calculate the tube-side (glycol) heat transfer coefficient and pressure drop are similar to those employed in the chiller design. The expression for obtaining the convective heat transfer coefficient is the more general form of the equation employed in the chiller design, and has a wider (temperature related) application range. This equation for turbulent pipe flow, is listed below.

$$\text{Nu} = \frac{hd}{K} = A \text{Re}^B \text{Pr}^C \quad (1)$$

where Nu is the Nusselt number, d is the internal tube diameter, K is the fluid thermal conductivity, Re is the Reynolds number, Pr is the glycol Prandtl number and A, B and C are empirical coefficients chosen to best fit the internally finned tube performance.

The equation employed to calculate the tubing pressure drop was the previously defined (see Section 1.4.1.2):

$$p = f \frac{L}{D} \frac{\rho}{g_c} \frac{v^2}{z} \quad (2)$$

where $f = A/\text{Re}^B$

Additional pressure drop will be experienced in the bend region where work must be done to turn and mix the flow. In order to approximate bend losses, an equivalent straight tube length of 50 l/d was used to simulate the bend pressure drop. (S-9).

4. Again, in order to determine reasonable values for the nucleate boiling heat transfer coefficient, reference data for R-113 boiling on a tube bundle was employed, with a correction for pressure effects. The reference data were obtained in a system where the refrigerant entered the shell as a two-phase mixture,

thus, including a certain amount of heat transfer enhancement due to bubble agitation. While this condition duplicates chiller performance, the refrigerant is expected to enter the boiler as single phase liquid. A correction factor, based on physical test data obtained at Carrier Corporation, was employed to account for this effect.

5. The shell side fouling resistance was assumed to be negligible.
6. To minimize fouling resistance and pressure drop, the linear tube-side velocity should lie between 3 and 10 ft/sec.

The method used to size the boiler heat exchanger is identical to the one used in the chiller design, section 1.4.1.2. These equations will be repeated, in brief form, for the sake of continuity.

$$q = U_o A_o \text{ LMTD} \quad (1)$$

$$U_o = \frac{1}{(1/h_{\text{glycol}} + R_f + R_m) \text{ SR} + 1/h_{\text{ref}}} \quad (2)$$

$$\text{LMTD} = \frac{(T_{\text{glycol}_2} - T_{\text{ref}}) - (T_{\text{glycol}_1} - T_{\text{ref}})}{\ln \left[\frac{(T_{\text{glycol}_2} - T_{\text{ref}})}{(T_{\text{glycol}_1} - T_{\text{ref}})} \right]} \quad (3)$$

$$\text{and } A_o = q / (U_o (\text{LMTD})) .$$

With the above procedures and design considerations in mind, the following preliminary boiler designs were obtained, and are listed in Table 1.4-4.

The 30 circuit design was chosen to limit the physical length and the pressure drop in the glycol circuits. The preliminary design which was costed and used in the layout is shown in Figure 1.4-7.

1.4.1.5 Design of The Liquid-Suction Heat Exchanger

The purpose and advantages of a liquid suction heat exchanger (LSHX) were shown in Section 1.2.4. When the liquid suction heat exchanger was designed, it was proposed to heat exchange suction vapor with the total liquid flow from the condenser in order to increase the NPSH available to the liquid pump. It was later shown that such an approach results in a serious loss in cycle efficiency, and the idea was dropped. However, the design discussed below and the subsequent cost estimates and layouts are based on the original concept of total flow.

The purpose of these calculations is to design a liquid suction heat exchanger (LSHX) which will transfer a minimum of 44,350 Btuh to the suction gas. The controlling conditions are listed below.

TABLE 1.4-4
PRELIMINARY BOILER DESIGN DATA

No. of Circuits (Tubes)	15	20	30	Units
Total External Surface Area	260	280	300	ft ²
Length of tube/ circuit	29.16	23.55	16.68	ft
Total Linear ft. of tube	437	471	500	ft
Glycol Pressure drop(tubing & bend)	4.86	2.45	.94	psi
Mean linear glycol velocity	6.05	4.54	3.025	ft/sec
Tube-side heat transfer coeffi- cient	3247	2580	1865	BTU/hr-ft ² i.s. - °F
R-113 boiling coefficient	504	484	461	BTU/hr-ft ² o.s. - °F
R-113 % of total resistance	24.4	24.2	23.64	%

15 TUBES ON 8.880" CENTERS
 10 TUBES ON 3.718" CENTERS
 9 TUBES ON 3.171" CENTERS

FIGURE 1.4 - 7

PRELIMINARY BOILER DESIGN

TUBE DATA: 30 CIRCUMF
 15.7 FT. TOTAL TUBE LENGTH PER CIRCUIT
 471 FT. TOTAL TUBE LENGTH
 1/4 O.D. TUBESHEET
 90 FINS PER INCH WITH 10 START 10

CARRIER CORPORATION	
ENERGY SYSTEMS DIVISION	
DATE: 3-15-78	BY: R. R. R.
BOILER	
SHEET 1 OF 1	

1. Vapor mass flow rate = 4972 lb m/hr
2. Saturated vapor inlet temperature = 43.5°F, pressure = 2.907 psia.
3. Total condenser liquid flow rate = 9568 lb m/hr
4. Liquid inlet temperature = 116.4°F

Once the thermodynamic target points were fixed, a number of design and heat transfer factors had to be considered. These are listed below:

1. Preliminary cycle analyses show that it is necessary to limit the vapor pressure drop to a maximum value of 0.05 psi.
2. A target thermodynamic effectiveness (vapor temperature change divided by maximum temperature difference in exchanger) of $E = 0.80$ was established.
3. Due to the extremely low vapor side pressure drop and the high effectiveness required, a plate fin heat exchanger with the vapor on the fin side was required. This also means that existing Carrier manufacturing techniques can be employed.
4. From a practical viewpoint, the drop in saturation temperature due to condenser fluid pressure drop through the LSHX should be substantially less than the initial amount of subcooling available. Therefore, the liquid pressure drop should be restricted to the range of 1 or 2 psi. In a practical design it is difficult to minimize the liquid pressure drop and still maximize the liquid convective heat transfer coefficient. Therefore, the final design will have to involve a slight compromise of these two factors.

The general method for sizing the LSHX proceeded as follows:

1. Proprietary Carrier Corporation data are available for air flow over a corrugated plate fin coil in the form of a heat transfer j -factor vs. Reynolds number and a friction factor vs. Reynolds number. Since these plots are dimensionless in nature, they were applied to gaseous R-113 flows by employing the correct R-113 thermophysical properties.
2. Heat transfer for the liquid tube-side flow was calculated with the use of a Dittus-Boelter type convection flow heat transfer coefficient for R-113 liquid. This equation, in the form employed, can be found in the ASHRAE Handbook of Fundamentals (A-6). For simplicity it is repeated below.

$$h = c \frac{G^{0.8}}{D^{0.2}}$$

$$= 0.00201 \frac{G^{0.8}}{D^{0.2}} \quad \text{for R-113 liquid} \quad (1)$$

where G is the liquid mass flow rate (lb m/hr-ft²) and D is the internal tube diameter in feet.

3. The liquid pressure drop was calculated with the use of the Fanning Friction Factor as follows.

$$p = f (L/D) (\rho/g_c) (v^2/z) \quad (2)$$

where $f = A/Re^B$

In this equation an equivalent length of 50 l/d was employed to calculate the pressure drop in return bends.

4. With the above information an iteration procedure on coil size, circuiting, and geometry was performed. Thus corrugated fin coil geometries were selected which would allow a 0.05 psi vapor-side pressure drop and a 1.76 psi liquid pressure drop.
5. Once the candidate geometry was selected, it was possible to calculate heat transfer surface areas and overall heat transfer coefficients, U_o , between the liquid and vapor streams.
6. The final LSHX capacity was completed by the effectiveness NTU approach as outlined in Reference H-2. For a thermal counterflow fluid arrangement, the effectiveness, E , was calculated by the following equation.

$$E = \frac{1 - \exp [-NTU (1 - C_{\min}/C_{\max})]}{1 - \frac{C_{\min}}{C_{\max}} \exp [-NTU (1 - C_{\min}/C_{\max})]} \quad (3)$$

where $NTU = \frac{V A_o}{C_{\min}}$ and $C_{\min}$ and $C_{\max}$ are the minimum and maximum values of the product of fluid mass flow rate and the fluid specific heat at constant pressure.

7. With E it was possible to calculate the actual leaving vapor and liquid temperatures, and the LSHX capacity as:

$$\begin{aligned} T_{V2} &= T_{V1} + E (T_{L1} - T_{V1}) \\ \text{and } T_{L2} &= T_{L1} + \frac{C_{\min}}{C_{\max}} (T_{V2} - T_{V1}) \\ \text{and } q &= C_{\min} (T_{V2} - T_{V1}) \end{aligned} \quad (4)$$

where T_{V1} and T_{V2} are the entering and leaving vapor temperatures and T_{L1} and T_{L2} are the entering and leaving liquid temperatures, respectively.

The above approach was used to generate a preliminary LSHX design, which was costed and used in the layout. The LSHX parameters are summarized in Table 1.4-5, and the design is shown in Figure 1.4-8.

1.4.1.6 Design of the Regenerator

The purpose of the regenerator heat exchanger is to use hot discharge gas leaving the turbine and the compressor to heat liquid refrigerant returning to the boiler/preheater. The objective of the calculations in this section was to design a regenerator heat exchanger capable of transferring 57,000 Btu/h between the vapor and liquid streams. The controlling conditions are listed below.

1. Vapor mass flow rate = 9753 lb m/hr
2. Vapor inlet temperature = 179.2°F
3. Liquid mass flow rate = 4782 lb m/hr
4. Liquid inlet temperature = 115°F
5. Vapor inlet mixed enthalpy = 106.026 Btu/lb m
6. Liquid outlet temperature and pressure = 166.4°F and 157.18 psia.

Once the above conditions were established, a number of practical design considerations had to be made. These factors and constraints are listed below.

1. The exchanger design is limited to a maximum vapor-side pressure drop of 0.5 psi (from cycle optimization studies).
2. It is desired that the regenerator have a target effectiveness of 0.80.
3. Due to the low pressure drop and high effectiveness required, a plate fin heat exchanger with vapor on the fin-side and liquid on the tube-side was required. This also means that existing Carrier manufacturing techniques can be employed.
4. The vapor-side pressure drop restriction required a low vapor face velocity. This low face velocity in turn results in a low vapor-side convective heat transfer coefficient. The end result is that the candidate coil design requires a large amount of surface area per unit width. This factor, coupled with the requirement for high heat transfer effectiveness results in an extremely deep coil.

The low vapor pressure drop requires a small fin density (on the order of 8 fins per inch).

5. In order to make a coil with the sufficient number of tube rows, it is necessary to build a very tall coil, and then turn it on its side such that the vapor enters the coil from the bottom end. Thus, the vapor flow is along the fin corrugations instead of across them. Vapor-side coefficients for this end flow configuration were assumed to be approximately equal to coefficients for flow in the transverse direction.

TABLE 1.4-5

DESIGN DATA FOR LIQUID SUCTION HEAT EXCHANGER

Face Area, in ²	205	
LBTS, in	25.6	
Height, in	8.0	
Depth, in	13.9	
Depth, rows	16	
Tubes in face	8	
Total No. Tubes	128	
No. Liquid Circuits	16	
Type Fins	Corrugated Plate	
Fins per Inch	8.0	
Tube Diam, No., in	3/8	
Tube I.D., expanded in	.3619	
Tube Spacing	1.00 x .866	
Vapor Face Velocity, ft/sec	9.52	
Vapor H.T. Coeff, BTU/hr ft ² F	9.28	
Vapor Press. Drop, psi	.0500	
Liquid Velocity, ft/sec	2.61	
Liquid H.T. Coeff, BTU/hrft ² F	235	
Liquid Press. Drop, psi	1.76	
Liquid Reynolds No.	20,200	
Overall H. T. Coeff, BTU/hrft ² F	69.7	(based on inside area)
Mean Temp. Diff, °F	26.2	
Total Ext. Surf., ft ²	293.7	
Total Length Tube, ft	273.4	
Capacity, BTUh	47,200	
Effectiveness	.851	$= \Delta t_v / (t_{L1} - t_{V1})$
Vapor Flow Rate, lb/hr	4,972	
Vapor Inlet Temp., °F	43.5	
Vapor Outlet Temp., °F	105.5	
Liquid Flow Rate, lb/hr	9,568	
Liquid Inlet Temp., °F	116.4	
Liquid Outlet Temp., °F	95.1	

FIGURE 1.4 - 8

PRELIMINARY LIQUID SUCTION HEAT EXCHANGER DESIGN

With the above conditions in hand, it was possible to proceed with the actual sizing of the regenerator. As in the case of the LSHX, proprietary nondimensional data for transverse gaseous flow over a corrugated fin-tube coil was employed. This allowed for computation of single-phase convection heat transfer coefficients and pressure drops.

The heat transfer coefficients and pressure drop for the tube-side liquid flow were calculated by use of the modified Dittus-Boelter and the Fanning Friction Factor equations. These equations can be found in Section 1.4.1.5.

An iterative procedure was employed to determine the final coil size. This procedure was as follows:

1. From $E = 0.80$ and the maximum and minimum fluid capacitance rate values, calculate the required exchanger NTU.
2. Estimate the overall heat transfer coefficient, U_o , to calculate an initial required surface area.
3. With this value of surface area, calculate the fluid stream pressure drops and heat transfer coefficients at fixed velocities.
4. With the information in 3, recalculate the vapor face velocity and the coil circuiting to obtain a new U_o .
5. Return to Step 2 and iterate until the desired capacity is obtained with a vapor-side pressure drop less than 0.5 psi and a liquid-side pressure drop less than 2 psi.

The results of this iterative procedure are summarized in Table 1.4-6. This design was used for costing and layout, and is shown in Figure 1.4-9.

TABLE 1.4-6

DESIGN DATA FOR THE REGENERATOR

Face Area, in ²	93.2	
LBTS, in	13.4	
Height, in	6.93	
Depth, in	28.0	
Depth, rows	56	
Tubes in Face	4	
Total No. Tubes	224	
No. Liquid Circuits	8	
Type Fins	Corrugated Plate	
Fins per inch	8.0	
Tube Diam, nominal, in	3/8	
Tube I.D., Expanded, in	.3619	
Tube Spacing, in	1.712 x .500	(perpendicular to 1 x .866)
Vapor Face Velocity, ft/sec	10.0	
Vapor H.T. Coeff, BTU/hrft ² F	26.4	
Vapor Press. drop, psi	.315	
Liquid Velocity, Ft/sec	2.52	
Liquid H.T. Coeff, BTU/hrft ² F	238	
Liquid Press Drop, psi	3.86	
Liquid Reynolds No.	23,300	
Overall H.T. Coeff, BTU/hr ft ² F	122.7	(based on inside area)
mean temp. diff., °F	19.57	
Total Ext. Surf., ft ²	268.9	
Total Length Tube, ft	250.5	
Capacity, BTUH	57,000	
Effectiveness	.800	$= \Delta t_L / (t_{V1} - t_{L1})$
Vapor Flow Rate, lb/hr	9,753	
Vapor inlet temp, °F	179.2	
Vapor outlet temp, °F	143.4	
Liquid Flow Rate, lb/hr	4,782	
Liquid Inlet Temp, °F	115.0	
Liquid Outlet Temp, °F	166.4

FIGURE 1.4 - 9

PRELIMINARY REGENERATOR DESIGN

TUBE DATA: 274 - $\frac{5}{8}$ " CORNER (AL190B) EXPANDED
 TYPE 21
 10 CIRCUMPS
 FINS 1 (107) CORRUGATED PLATE
 ALUMINUM .010" THICK
 8 FINS PER INCH

CARRIER CORPORATION	
ENERGY SYSTEMS DIVISION	
DATE: 2-21-78	DESIGNED BY: R. R. RUT
REGENERATOR	
PROJECT NUMBER: E3006-04-0004	

1.4.2 PUMP DESIGN

Development of a boiler feed pump or condensate pump for the Rankine power cycle has been recognized as a major task in this project. The objectives of the design are:

- Hermetic design
- Liquid refrigerant lubrication
- Variable capacity
- Medium high delivered pressure
- Low NPSH
- High efficiency
- Low cost
- Long life

There are no pumps on the market which will meet all of these objectives. Indeed, there are no organic Rankine powered systems in commercial production for which such a pump has been designed. Developments in this field which are described in the literature generally do not give any details of the pump design. Experimental systems have taken the expedient route of using commercially available pumps which only meet the pumping requirements. For example, the pump used in the Carrier 2-1/2 ton Rankine air conditioner was an open drive gerotor pump designed for hydraulic applications. It met the pumping requirements, but was not hermetic, low cost, or highly efficient. We also know that smaller pumps used in ammonia-water gas air conditioners, with similar requirements, had to be developed by that industry.

The pump design task was undertaken through parallel programs. Specifications were sent to the pump industry to search for commercially available pumps which would meet the requirements or manufacturers who were interested in special designs. The other approach was to review past Carrier experience in this field and create a conceptual design of our own. As described below, we were unable to find a commercially available design which met all the requirements, and proceeded to develop conceptual designs of two alternate types for in-house manufacture.

The initial pump specifications were determined to be:

- A. Liquid pumped: Refrigerant R-113 (oil free)
- B. Nominal flow: 96.3 lbs/min (7.92 gal/min)
- C. Inlet pressure: 16.7 psia
- D. Inlet liquid temperature: 115°F
- E. Static Inlet head: 2 ft of refrigerant
- F. Net positive suction head: 4.1 ft of refrigerant
- G. Outlet pressure: 146.7 psia
- H. Differential pressure: 130 psi (206 ft of refrigerant)
- I. Liquid Temperature range: 50 to 135°F
- J. Duty cycle: Varies up to 24 hours/day
- K. Liquid Viscosity: 1.849 lbs/ft-hr at 60°F
1.588 lbs/ft-hr at 80°F
1.379 lbs/ft-hr at 100°F
1.209 lbs/ft-hr at 120°F
- L. Maximum leak rate: 2 ounces/year
- M. Efficiency: Overall pump and motor 50%.

1.4.2.1 Pump Search

Letters requesting condensate pump proposals based on the above specifications were sent to 23 pump manufacturers.

The following is a list of manufacturers with their proposals who were able to offer equipment meeting most of the requirements.

- A. Airesearch Manufacturing Company of California, Division of the Garrett Corporation. Hermetic motor driven vane type pump, refrigerant cooled, 40-45% overall efficiency. Two sample pumps plus development costs - \$46,292. Production pump costs not determined.
- B. Buffalo Forge Company. Hermetic centrifugal pump. 9-10% overall efficiency. Approximate costs production pump \$3,400.
- C. Chempump Division, Crane Company. Hermetic Refrigerant cooled centrifugal pump. 10% overall efficiency. Approximate costs for production pumps \$2,233.
- D. Roth Pump Company, Subsidiary of Roy E. Roth Company. Hermetic refrigerant cooled turbine pump. Low efficiency about same as centrifugal pumps. Approximate costs for production pumps \$15,314.

The following pumps used or investigated by Carrier for other similar applications were reviewed:

- A. Whirlpool Corporation. Canned magnetic drive piston pump with a centrifugal booster. Pump capacity of original sample too small; however, concept good and could be scaled up. Would require complete pump development program. Overall efficiencies should be about the same as a vane or gerotor pump - 50%.
- B. Carrier Corporation. RDC designed motor driven piston pump. A large, slow acting pump requiring a low speed motor. Concept needs to be further developed for use in this program.
- C. The Weatherhead Company. Hydraulic diaphragm pump. The diaphragm is actuated by a motor driven hydraulic piston pump. Pumps used by Carrier in 7-1/2 and 10 ton ammonia water air conditioners. Reliability of the diaphragm was not satisfactory.
- D. W. H. Nichols Company. Gerotor type pump - open direct or belt drive. Used by Carrier on the Florida solar residence Rankine system. Cavitation may be a problem and would require careful design consideration. No work has been done by Nichols or Carrier on a hermetic motor-pump combination. Carrier must develop gerotor housings and running gear. Overall efficiencies expected to be in the range of 50 to 60%.

- E. Carrier Corporation. Trap pump. This is a free piston (plunger) double acting liquid pump driven by high pressure refrigerant vapor from the boiler. It has the advantage of using solar rather than electrical energy to drive the pump. This would be similar to a steam pump. A full development program would be required.

1.4.2.2 Pump Concept Selection and Design Analysis

Many factors must be considered in selecting a pump concept for the Rankine cycle. Probably, the two most important to consider are reliability and efficiency. These would be followed by lubrication (ability to operate without oil) characteristics, refrigerant leakage and noise. A low NPSH requirement is a must. Pump, motor and drive costs and required development cost must also be considered.

For a centrifugal pump to operate in the 50 to 60% efficiency area, a specific speed of 1200 to 1600 is necessary. The specific speed of this application is approximately 200 for a 3600 rpm pump speed. A pump speed of 25,200 rpm would be required for good efficiency. A centrifugal pump operating at 25,000 rpm would require a NPSH of 10.5 ft which is much greater than the NPSH of 4.1 ft available for this application. For NPSH of 4.1 ft, the maximum speed is 12,800 rpm.

The low specific speed of this application suggests the use of a positive displacement pump. Metal bellows and diaphragm pumps were ruled out of consideration for this application because of expected reliability problems, size, noise and low speed drives. However, two other types or classifications of positive displacement pumps offer possibilities of meeting the application requirements. First, Alternate 1 is a high pressure refrigerant vapor driven double acting free piston (plunger) type pump; and secondly, Alternate 2 is an electrically driven rotary pump of either the vane or gerotor type.

The volumetric efficiency for rotary pumps increases with pump speed; however, as speed increases the required NPSH also increases. The low NPSH available will cause problems for a 3500 rpm rotary pump and may cause problems for a 1750 rpm pump. It may be possible to eliminate cavitation (low NPSH) problems of gerotor or vane pumps by using a centrifugal pump in tandem to increase the inlet liquid pressure (NPSH) to 10 or 12 ft. This has been tried by Carrier on a piston type pump. The specific speed of this tandem or booster centrifugal pump would be about 1500 based on 3500 rpm and have an efficiency of 45 to 55%. Preliminary calculations indicate it would be best to operate a rotary pump at 1750 rpm, and avoid the use of a centrifugal booster pump by careful design of the pump inlet. If, through prototype tests, the need for a centrifugal booster pump is proven, then the gerotor and booster pump should be operated at 3500 rpm, to increase the overall efficiency of the combination. Section 1.4.2.4 of this report describes a 1750 rpm gerotor pump. The rotary pump would be the lower cost to develop than the free piston pump, because it uses a motor and gerotors that are in production for other applications. Development work would center around lubrication, bearing selection, motor cooling, pump capacity and efficiency.

1.4.2.3 Pump Design - Alternate 1, Free Piston "Trap" Pump

Although the free piston "trap" pump would be more expensive to develop, it has the major advantage of eliminating the need for electrical energy to operate the pump except for a small amount required to operate the valves. Preliminary calculations indicate about the same cost effectiveness for this type of pump as for the solar driven refrigeration compressor. See Section 1.4.2.3 of this report for this design concept. To operate with this type of pump would require about 5% more solar energy (collectors) and more condenser surface than with an electrically driven pump. The "trap" does not take advantage of the available work of the high pressure vapor as it expands to the lower pressure of the condenser through the exhaust valves at the end of the pumping stroke and, therefore, would be larger in size than a Rankine engine for the same condensate pumping capacity. However, some of the heat in the exhaust gas will be gained back with the regenerator.

It would be possible to design the system as a free piston Rankine engine driven pump, but the design of a trap pump is simpler because the starting and inertia problems of a free piston Rankine engine are avoided.

1.4.2.3.1 Layout

Figure 1.4-10 shows a preliminary layout of a free piston "trap" pump. The pump and engine shown have been proportioned for a maximum flow of 8 gpm of liquid refrigerant R-113. To handle peak refrigerant flow rates, the final trap pump would be larger.

High pressure vapor from the boiler admitted to one cylinder through one of the motor driven vapor valves located at each end of the pump forces the piston to move. Because of the difference in piston area, liquid refrigerant is pushed from the pump cylinder to the boiler. At the same time, the other pump cylinder fills, and the other vapor cylinder completes its exhaust to the condenser. At the end of the stroke, the vapor valves are reversed by the control motor. The high pressure vapor in the cylinder is exhausted to the condenser while more high pressure vapor admitted to the opposite end of the pump causes the piston to reverse direction and repeat the cycle. The high pressure vapor is allowed to expand to the condensing pressure level through the exhaust valves at the end of each stroke. As stated above, this design does not take advantage of the expansion work available in the high pressure vapor. Instead, it uses a volume of high pressure vapor equal to the volume of the liquid pumped times the area ratio. The energy supplied to the process is the heat necessary to evaporate this volume of vapor in the boiler.

1.4.2.3.2 Cost Estimate

A free piston "trap" pump sized for a maximum flow rate of 12 gpm is estimated to cost between \$200 and \$220 in annual production quantities of 500 units. This cost is the sum of material costs, direct labor costs and manufacturing overhead costs.

FIGURE 1.4 -10
PRELIMINARY FREE-PISTON TRAP PUMP LAYOUT

CARRIER CORPORATION			
ENERGY SYSTEMS DIVISION			
SCALE: 1" = 1'-0"	DESIGNED BY	DRAWN BY: E.S. 12-78	
FREE-PISTON LIQUID PUMP			
APPROVED BY:		D	

1.4.2.3.3 Control Concepts

An advantage of the "trap" type pump is that its pumping rate can match the system requirements by changing the speed of the valve control motor which determines the number of pump strokes per minute. This type of pump control conserves the available solar energy, since it is required to operate only when the boiler level control is calling for more liquid refrigerant. As the boiler level drops, the speed of the pump would be increased. Under conditions of low flow requirements the piston could reach the end of its stroke before the motor reverses the control valves. To maintain an even liquid refrigerant flow through the regenerator and preheater under this condition, a small surge tank must be installed at the pump outlet.

The control system consists of a boiler liquid level control (variable resistor) and a motor speed control. The motor speed control operates a gear reduction motor which in turn drives the engine inlet and exhaust valves. A low level limit mounted on the liquid receiver is wired to prevent operation of the pump unless there is a satisfactory supply of liquid refrigerant in the receiver. This is to prevent damage to the pump and engine because of refrigerant leaks - loss of charge protection.

1.4.2.3.4 Performance Predictions

The free piston "trap" pump would require an additional refrigerant flow rate of 5.5 to 6.0%. This means that the collectors, regenerator and condenser all must be increased in size over an electrically driven pump system. The 5.5 to 6.0% increased flow rate is based on a mechanical advantage of 1.18 (piston area ratio) and a volumetric efficiency of 90%. If the exhaust gases from the trap pump are passed through a regenerator, then the extra amount of solar energy required to drive this pump would be about 4 to 4.5% more than a Rankine cycle chiller system with an electrically driven pump.

1.4.2.4 Pump Design - Alternate 2, Gerotor Pump

1.4.2.4.1 Layout

Figure 1.4-11 shows a preliminary layout of a hermetic motor driven gerotor type rotary condensate pump. The design shown is for a 12 gpm liquid refrigerant R-113 flow rate. The final design may need to be modified to match system peak flow requirements.

The pumping element is a standard W. H. Nichols Company gerotor, Model 6170. The gerotor housing, shaft, bearings and motor must be developed. The moving parts are lubricated with refrigerant. A small portion of liquid refrigerant from the pump discharge is used for lubrication and to cool the motor.

1.4.2.4.2 Cost Estimate

Cost estimates were prepared for an open drive gerotor pump and also for a semi-hermetic pump. The fabrication cost (sum of material, direct labor and manufacturing overhead) for each pump based on a maximum annual requirement of 500 units is \$159 for the open-drive pump and \$193 for the semi-hermetic pump. The open drive pump would require pulleys, V-belt and motor to be equivalent to the semi-hermetic pump.

1.4.2.4.3 Control Concepts

The pumping rate of the gerotor pump can be matched to the system requirements by two methods. Either a fixed speed pump motor or a variable speed pump motor could be used. The two curves on Figure 1.4-12 show the difference in pump horsepower required by the system for each of these two methods. Either method is controlled by a liquid level controller which either speeds up the pump (variable displacement) or closes off a bypass (fixed displacement) as the boiler level drops. Since the curve on Figure 1.4-12 does not include the effect of changing motor efficiency on the variable speed pump, the difference in power may not be as great as shown.

A low level limit control located at the liquid receiver is required to prevent operation of the pump unless there is a satisfactory supply of liquid refrigerant available in the receiver. This is to prevent damage to the pump in case of refrigerant leaks -- loss of charge protection.

1.4.2.4.4 Performance Predictions

A pump speed of 1750 rpm was selected because of the low available NPSH of 4.1 ft of refrigerant. The mechanical efficiency of the proposed gerotor pump operating at 1750 is expected to be 80 to 85%. The calculated volumetric efficiency is 88%, and the actual may be assumed to be 85 to 88%. Assuming a motor efficiency of 85%, then the overall pump efficiency could be expected to range from 57 to 64% at design conditions.

FIGURE 1.4-12

PUMP HORSEPOWER REQUIRED BY CYCLE

1.4.3 CHILLER PACKAGE DESIGN

The overall chiller design integrates the Rankine driven vapor compression cycle and controls into a unitary package which meets the specifications and objectives set forth in Section 1.1.2. A layout has been made using the heat exchangers which were designed to the initial cycle specifications, and the turbo/compressor layout supplied by MTI.

Controls were evolved which integrated the normal chiller controls with the special requirements of the Rankine cycle. Motor power supply controls are not included at this time.

Those codes and standards which affect the conceptual design up to this point are addressed.

1.4.3.1 Layout

Figures 1.4-13 and 1.4-14 illustrate one possible arrangement of system components into a single package 16' 4" long by 7' 6" wide by 7' 2" high.

In this arrangement, cooling air enters the condenser coils through both of the unit's longer sides. Condenser air flow is maintained by four individually powered direct drive fan-motor assemblies. The floor of the condenser entering air compartment serves as a roof to protect all other components of the system. The condenser air discharge is vertically upward which reduces possibilities of condenser cooling air recirculation.

The turbine-compressor, boiler, cooler and other heat exchangers are mounted under the condenser on a channel iron frame. These components are located and connected together with pipe sized to obtain optimum performance.

This lower section is enclosed with weatherproof side panels removable for service operations.

All water (solar collection system and chilled water) connections are at one end of the unit. Removal of all heat exchangers except for the regenerator are also at this end of the package. Service of the turbine-compressor unit and condensate pump is at the opposite end. Electrical controls are located on the side of the unit with electrical power supply connections made at the turbine-compressor end.

This arrangement of components results in a uniform distribution of unit weight (operating) over the floor area of the package. This makes it easier to install the package especially for roof mounted installations.

The size and shape of the complete package allows transportation by either truck or rail. Its shape and balance facilitate rigging.

FIGURE 1.4 -13

NOTE:
3 FT. OF OPEN AREA IS REQUIRED AROUND PERIMETER OF UNIT FOR AIRFLOW AND SERVICING. AN ADDITIONAL 6 FT. IS REQUIRED AT WATER CONNECTION END FOR REMOVAL OF HEAT EXCHANGERS.

CARRIER CORPORATION			
ENERGY SYSTEMS DIVISION			
DATE	1-1-70	DESIGNED BY	W. R. GALT
REV.	1-20-70	BY	
SOLAR WATER CHILLER - AIR COOLED (LAYOUT)			
DRAWN BY			D. J. BROWN

FIGURE 1.4 -14

CARRIER CORPORATION ENERGY SYSTEMS DIVISION			
MODEL 1-1-10	REVISED BY	DATE BY	DATE
DATE 4-20-78			
SOLAR WATER CHILLER - AIR COOLED - ASSEMBLY			
DRAWN BY			DATE
D			1/20/80

1.4.3.2 Chiller Controls

The control system of the packaged chiller will contain all of the controls necessary to operate the system under the normal range of expected conditions and to protect it against abnormal conditions or failures. The controls are comprised of four functional groups represented on the control logic diagram, Figure 1.4-15, as the start-up sequence, chiller controls, Rankine cycle controls, and motor compressor controls.

The start-up sequence is required to correct anticipated refrigerant charge migration problems which may occur after prolonged shutdown. Referring to Figure 1.4-16, refrigerant may migrate out of the boiler or cooler due to leakage through the vapor line shutoff valve, V4, or the liquid line check valve. When start-up is initiated, the start-up sequencer will operate the feed pump to transfer refrigerant between the vessels until the satisfactory inventory situation is established. Once the start-up sequence control is satisfied, the chiller start-up can proceed. Level sensors provided in the boiler, receiver and cooler will be used to determine the corrective action required, according to the logic of Figure 1.4-17.

The chiller control group comprises essentially those safety and operating controls which are found in commercially available air cooled chillers, excluding the compressor controls. Figure 1.4-18 is a logic diagram of these controls. Safety switches prove chilled water flow, vapor compression system pressures, and chilled water temperature. Also, the condenser fan staging controls are included for low ambient condenser pressure control (one stage is shown, but more may be required to satisfactorily cover the desired ambient range). If the chiller control group is satisfied, start-up proceeds, and the compressor is activated.

The compressor drive mode, Rankine or motor, is determined by the solar control limit switch, which compares outdoor ambient air temperature and solar water temperatures to determine whether the solar water is hot enough to drive the compressor. If so, the Rankine cycle control group is activated. If not, the compressor motor control group is activated.

The Rankine cycle control group provides the solar control limit switch function as well as the Rankine cycle operating control (the limiting relationship between the solar water temperature and the air temperature is shown as line A on Figure 1.4-44 and the theory is discussed in Section 1.4.5.4). If the solar water temperature is greater than line A for a given ambient air temperature, then the turbine will have enough power to operate the compressor above the surge speed, and the Rankine cycle can be energized to provide the cooling demanded. However, if the solar temperature is too much hotter than line A at the given ambient temperature, the compressor may overspeed into the choke flow region with subsequent loss in efficiency. This can be avoided by controlling the water temperature into the boiler to the value required by line A. This can be accomplished in a practical fashion by a modulating mixing valve at the entrance to the boiler, which mixes hotter supply water with cooler recirculated water to achieve the desired entering temperature.

FIGURE 1.4 -15
LOGIC OF MAJOR FUNCTIONAL CONTROL GROUPS

FIGURE 1.4-16
REFRIGERANT LEVEL CONTROL SYSTEM

L1 - BOILER HIGH LEVEL LIMIT
L2 - RECEIVER LOW LEVEL LIMIT
L3 - COOLER HIGH LEVEL LIMIT

FIGURE 1.4 -17

LOGIC FOR START-UP LEVEL CONTROL SEQUENCE

FIGURE 1.4 -18

CHILLER CONTROL GROUP LOGIC

Figure 1.4-19 is a logic diagram of this control system. In addition to the modulating water valve, the control opens the valve in the vapor line between the boiler and turbine to power the turbine if solar water flow is proved and a minimum boiler pressure is exceeded. To avoid inordinately long start-up lags on short cycles, the combination of the vapor shut-off valve, the liquid line check valve, and insulation will maintain the boiler pressure and temperature over short periods for a quick start.

During Rankine cycle operation, the refrigerant pumped into the boiler will be controlled from level sensors L1 and L2. The pump capacity will be set to exceed the largest mass flow of vapor possible through the turbine. During other conditions of operation, the excess mass flow into the boiler will cause the level to rise, eventually turning on L1, and opening V3. The flow from the pump to the receiver through V3 will be limited by an orifice, with the remaining flow to the boiler set to a value less than the smallest flow of vapor possible through the turbine. Thus, the cycling of V3 by L1 will regulate the mass flow into the boiler to match the mass flow out through the turbine. If the level in the receiver should fall below L2 during Rankine cycle operation, a situation which can only result from a malfunction, the system will be shut down. Flow of refrigerant to the cooler is regulated by a float valve or an electrical, level controlled expansion valve; thus during operation, the cooler level will be closely controlled and would not be expected to reach L3. If it does reach L3, the system will shut down, proper levels would be restored after restarting by the start-up sequencer.

The motor control group regulates the motor speed in relation to the saturated temperature difference between the evaporating and condensing refrigerant (this control function is shown on Figure 1.4-46 and the theory is described in Section 1.4.5.5). In addition, the motor control group provides thermal and current overload protection for the electric motor and power pack. The motor control logic is shown in Figure 1.4-20. Because the motor is the default operating mode for the Rankine cycle, the chiller will shut down if the motor thermal protection switch opens.

1.4.3.3 Code Compliance

The only codes which apply at this stage of the conceptual design relate to the design of the pressure containing vessels and piping. The wall thickness of these vessels was designed in accordance with the B9.1 code (A-9) and the ASME Boiler Code (A-14) in order to develop realistic cost estimates.

According to the B9.1 code, R-113 is a group 1 refrigerant. The proposed chiller system is considered to be an indirect closed-surface system, and will meet Section C.2 of the code when the refrigerant containing components are located outdoors (as proposed) or in a machinery room.

The design pressure has been selected to include all operating or stand-by conditions according to Section 8.2. For a heat operated cycle, the possibility that all vessels may be exposed to maximum cycle pressure must be considered. Therefore, all vessels have been designed for the maximum boiler pressure. A pressure limiting device, and a relief valve will be

FIGURE 1.4 -19

RANKINE CYCLE CONTROL LOGIC

FIGURE 1.4 -20

MOTOR CONTROL LOGIC

incorporated. The maximum design operating temperature will be 320°F, corresponding to an equilibrium saturated pressure for R-113 of 211 psia. The pressure limiting device will be set at 235 psia which becomes the design working pressure according to section UG-21 of the boiler code. The relief valve pressure must not be more than $235 \text{ psia} / 0.90 = 261$, according to section UG-125. The ultimate strength of all vessels should exceed 2-1/2 times that value, or 653 psia.

For shells and tubing, a low carbon fusion welded steel tubing, such as ASTM A-53, Grade B, will be specified. The following formula is used to determine the wall thickness, t , from section UG-27 of the boiler code.

$$t = \frac{P R_o}{S E + 0.4 P}$$

where R_o = tube outside radius, inches

t = wall thickness, inches

P = design pressure, psi

S = maximum allowable stress, psi

E = joint efficiency

The maximum allowable stress for A-53 up to 650°F is 10,200 psi, per Table UCS-23. A joint efficiency of 1.0 is recommended by Section UW-12. In all cases, schedule 40 pipe wall thicknesses will exceed the minimum thickness calculated by this formula. Accordingly, schedule 40 pipe was specified for all heat exchanger shells. Equivalent thickness elliptical heads were specified.

1.4.4 MANUFACTURING COST ESTIMATES

An estimate of the cost to manufacture an air-cooled solar driven Rankine cycle engine and water chiller package was completed. The cost estimate was based on a nominal 25 ton water cooling capacity. Production quantities were assumed to be in the order of 100 to 500 packages per year (see Appendix A1.4). Whenever possible, cost estimates were based on actual production costs of similar components with minor adjustments to cover size and function variations. For costs of components where no similar part was available, a detailed cost estimate was made by estimating the required time for each process required to manufacture this particular component. Costs for raw materials and purchased parts were obtained and used with current labor and burden rates to arrive at a reasonable manufacturing cost.

1.4.4.1 Component Cost Estimates

Table 1.4-7 below summarizes the estimated manufacturing costs by major components. To help in analyzing these costs, the total package costs were split into two parts. The cost of components related to the cooler are listed in the first column while those related to the Rankine engine are listed in the second column. The refrigerant piping diagram, Figure 1.4-21, shows the various components and their locations with respect to the entire system.

1.4.4.2 Chiller Package Cost

The estimated manufacturing cost (labor, material, and burden) of the proposed air-cooled solar chiller is \$14,893. This includes a stand-by electric motor, power supply, controller and shipping package. To this figure must be added engineering, tooling, sales and administration, and mark-ups for the manufacturer, distributor and dealer to obtain the selling price.

1.4.4.3 Equipment Price Comparisons

The estimated equipment prices were compared to other established prices in three ways. First the selling prices of a line of air-cooled reciprocating water chiller were compared to the estimated selling price of the cooling-related components of the chiller. Next the estimated selling price of the Rankine engine and related components was compared to the prices of solar-powered organic Rankine cycle engines presented in an article by Barber (B-16). Thirdly, the estimated selling price of the complete chiller was compared against a commercially available solar powered, air cooled absorption chiller. To make the absorption chiller functionally equal, the prices of a cooling tower and a stand-by electric reciprocating water chiller were added. Assuming that auxiliary electric energy is required, an electric chiller will be needed to provide cooling in the event that solar energy is not available to power the absorption machine. The proposed Rankine unit includes an electric motor drive system for auxiliary electric energy back-up.

TABLE 1.4-7

<u>COMPONENTS</u>	<u>MANUFACTURING COSTS</u>		
	<u>Cooler Related Components</u>	<u>Rankine Engine Components</u>	<u>Total Package Rankine Engine & Cooler</u>
Heat Exchangers			
Boiler	-----	\$1625	\$ 1625
Condenser & Fans	\$ 932	\$1000	\$ 1932
Preheater	-----	\$ 974	\$ 974
Regenerator	-----	\$ 550	\$ 550
Liquid Suction Exchanger	-----	\$ 600	\$6 600
Cooler	\$1579	-----	\$ 1579
Base	\$ 450	\$ 535	\$ 985
Turbine Compressor	\$ 400	\$ 400	\$ 800
Controls	\$ 221	\$ 724	\$ 945
Receiver	\$ 75	\$ 78	\$ 153
Pump	-----	\$ 500	\$ 500
Covers, Frame & Associated Labor	\$ 450	\$ 462	\$ 912
Piping	\$ 444	\$ 445	\$ 889
Refrigerant	\$ 150	\$ 160	\$ 310
Misc.	<u>\$ 69</u>	<u>\$ 70</u>	<u>\$ 139</u>
Sub Totals	\$4770	\$8123	\$12893
Hermetic Motor	\$ 387	-----	\$ 387
Speed Control & Power Supply	<u>\$1613</u>	-----	<u>\$ 1613</u>
Total Package	\$6770	\$8123	\$14893

FIGURE 1.4 - 21

REFRIGERANT PIPING DIAGRAM

Figure 1.4-22 is a curve of typical selling prices of conventional air-cooled water chillers. From this curve the selling price per ton of a conventional chiller would be \$563 for a 25 ton package. A conventional air cooled water chiller performs the same function as the cooler part of the proposed equipment; however, some means of driving the refrigeration compressor must be provided for an equal comparison. Conventional air conditioning and refrigeration equipment marketed through dealers and distributors usually costs the customer two and a half to three times the manufacturer's costs, depending on manufacturing facilities and tooling required for the product. On equipment sold directly to the customer by the manufacturer, this "mark-up" would be somewhat less.

Table 1.4-8 below outlines the cooler-related components selling price comparison. This shows that the cooler portion of the proposed design is in line with the selling prices on conventional equipment.

TABLE 1.4-8

COST-PRICE COMPARISON
25 TON AIR-COOLED WATER CHILLERS

	<u>Conventional Reciprocating Water Chiller</u>	<u>Solar Driven Chiller Package Cooler Components Only</u>
Manufacturing Costs	--	\$4770
Addition of Motor and Drive (Conventional)	--	\$ 200
Selling Price Mark-Up	--	2.5
Selling Price Per Ton	\$563	\$ 497
Selling Price	\$14,075	\$12,425

The curve in Figure 1.4-23 giving the equipment costs for solar-powered Rankine engines operated at 300°F (150°C) was developed from data reported by Barber (B-16)

At 17.3 kW, which is the shaft output of the proposed Rankine machine, Barber's data gives an equipment cost or selling price of \$650/kW. Table 1.4-9 compares this selling price against the selling price estimate for the Rankine cycle components of the proposed design.

A mark-up of 2.5 results in a selling price about 3% higher than Mr. Barber's data. For equal performance and functions the comparison is close, particularly considering the mark-up contained in the \$11,245 selling price is not known. Also, the equipment content is not known in detail, but assumed to be water cooled and designed for indoor applications. Condenser water pumps necessary for operation are not included, whereas the cooling air fans were included in the Carrier estimate.

FIGURE 1.4 - 22

COOLER SELLING PRICE COMPARISONS

FIGURE 1.4 - 23

RANKINE ENGINE AND SYSTEM PRICE COMPARISON

TABLE 1.4-9
COST-PRICE COMPARISON
RANKINE CYCLE EQUIPMENT

	<u>Barber's Data ** 17.3 KW Output</u>	<u>Carrier's Estimate 17.3 KW Output (25 Ton Package)</u>
Manufacturing Costs	-----	\$ 8123
Adjustments		
Less Liquid Suction Exchanger	-----	\$ 600
Less Receiver	-----	\$ 78
Less Covers	-----	\$ 500
Less Part of Controls	-----	<u>\$ 300</u>
Total Manufacturing Costs	-----	\$ 6645
Selling Price Mark-Up	-----	2.5
Selling Price - Rankine Engine	\$11245	\$16613
Selling Price - Cooling Tower	\$ 2429	Air-Cooled
Selling Price - Cooling Tower - Piping Installa- tion Labor	<u>\$ 2429</u>	<u>Air-Cooled</u>
Total	\$16103	\$16613

**REF. B-16

Table 1.4-10 is a comparison of the proposed solar driven Rankine cycle chiller with a commercially available water-cooled solar driven absorption chiller. A cooling tower and some means of stand-by water chilling equipment such as a conventional electrically powered reciprocating chiller must be provided for the absorption unit to be functionally equal to the proposed equipment.

With a mark-up of 2.5 the selling price for the Rankine cycle equipment would be competitive. The absorption pricing does not include added installation costs for the tower and chiller. Another point to consider is that the Rankine machine has been designed with a weatherproof cabinet suitable for outdoor installation whereas only the cooling tower of the absorption system is installed outdoors. Indoor installations use valuable space and may cause vibration or noise problems.

1.4.4.4 Cost Reduction Possibilities

An initial review of the component cost estimates indicates the following areas for possible cost reductions.

1. The two liquid refrigerant preheaters could be combined, or an even greater savings might be obtained by use of a "tube-in-tube," multi-pass heat exchanger with water in the inner tubes and liquid refrigerant in the annular spaces. See Figure 1.4-24 for the proposed design. In addition to a savings in preheater costs, the proposed design would reduce the amount of refrigerant charge required in the system.
2. The liquid suction heat exchanger could be mounted within the cooler shell. Figure 1.4-25 illustrates this proposal.
3. It may be possible to reduce the regenerator cost by using a longer 10" pipe shell. While the shell would cost more, the use of smaller diameter dished heads or possibly flat plate heads would result in a net savings.
4. The size of the package used to determine cost estimates was over 1200 cubic feet. By rearranging components, it may be possible to reduce the size of the package to 800 or 900 cubic feet. This could save both material and shipping expense. The package described in Section 1.4.3.1 of this report is 878 cubic feet.

1.4.4.5 Cost Factors

Cost factors for each heat exchanger were developed in order to evaluate chiller cost/performance relationships (section 1.4.5.4). The cost factor is the change in cost for an incremental change in area of a heat exchanger.

TABLE 1.4-10

COST-PRICE COMPARISON

Conventional Solar Driven Absorption
vs. Proposed Rankine Cycle (25 Tons)

	<u>Conventional Solar Driven Absorption</u>	<u>Rankine Cycle Proposed Package</u>
Manufacturing Costs	----	\$14893
Mark-Up	<u>----</u>	<u>2.5</u>
Selling Price	\$23821	\$37232
Adjustments		
(1) For Stand-by Cooling 25 Ton Reciprocating Cooler	\$14075	----
(2) Cooling Tower	<u>\$ 2429</u>	<u>----</u>
Total	\$40325	\$37232

FIGURE 1.4 - 24

TUBE-IN-TUBE PREHEATER
3 PASS SHOWN

FIGURE 1.4 - 25

COMBINED COOLER AND LIQUID-SUCTION
HEAT EXCHANGER

Each heat exchanger design was examined carefully to determine how to change the surface area without changing the overall coefficient of heat transfer, U . In most cases, this could be accomplished by changing the length of the shell and tube bundle, without changing the number of tubes, diameter, or number of circuits. The only other effect of this type of change was a proportional increment in tube side pressure drop. It was assumed that for this type of change, only the material cost changed. This type of change applied to the boiler, cooler, preheater, LSHX, and regenerator. However, in the case of the LSHX and regenerator, the step size of the change was limited to 2 rows of tubes at a time, and the vapor side (shell side) pressure drop was also proportionally affected.

It was determined that the only way to conveniently scale the condenser for a constant U value was to include the air handler. Therefore, the air handler was assumed to change its size in proportion to the surface area. The total material, labor, and burden cost of the coil and air handler was changed in proportion to the size change.

The estimated cost of refrigerant charge in this design is significant. Therefore, estimates were made of the refrigerant charge in each vessel and incremental costs were included in the cost factors. The receiver size is related to the refrigerant charge capacity of the boiler and the cooler. It was assumed to vary with these heat exchangers in the ratio of 58% with the boiler and 42% with the cooler, and these incremental costs were also included in the boiler and cooler cost factors.

The resulting cost data are given on Table 1.4-11. The variable cost is that portion of the total heat exchanger cost which varies with surface area in the manner defined above. The cost factor, which is the variable cost per unit of area, can be most conveniently derived from the variable cost change for a percentage change in UA (assuming U holds constant).

TABLE 1.4-11

COST ANALYSIS DATA OF BASE DESIGN

	<u>Total Cost</u>	<u>Variable Cost*</u>	<u>UA Btu/NRF</u>	<u>Pressure Drop Inside</u>	<u>Outside</u>
Boiler	\$1625	\$1138	32,700		
Preheater	1370	660	5,064		
Cooler	1579	708	35,260		
Condenser	3049	2830	63,000		0.1
LSHX	706	103	1,805	1.76	0.05
Regenerator	<u>655</u>	<u>145</u>	<u>2,908</u>		0.315
Subtotal	\$8984	\$5584			
Other Costs	<u>8643</u>				
TOTAL	\$17,627				

* includes refrigerant.

1.4.5 PREDICTED PERFORMANCE

Performance of the conceptual design was calculated over a range of off-design conditions in several iterations. The first results were with a radial bladed, vaneless diffuser compressor. The turbo-compressor performance balance point was determined graphically. Because the cycle computer program (RANKØ6) used did not simulate fixed heat exchangers, the resulting data represented a family of units ideally designed for each condition selected. However, some valuable insights were developed in areas of turbo-compressor matching techniques, turbo-compressor operating limitations, motor requirements, and off-design limitations. These preliminary results are not reported herein.

For the next iteration, the computer program was modified to calculate cycle conditions with fixed heat exchangers at various selections of entering solar water, entering condenser air and leaving chilled water. A performance map of a backward curved compressor with inlet guide vanes and a vaneless diffuser was available for manually calculating a cycle balance point. It was this iteration which defined the basic off-design performance of the system and provided insight into control strategy, as well as provided the off-design performance map used in the system simulation evaluation. These results are reported in detail below.

The final iteration under Phase I utilized the cycle program RANK1Ø, which used curve-fit turbo-compressor data to automatically determine the system performance under any input conditions. This program confirmed the data calculated manually in the second iteration. Then, using variable cost data for each heat exchanger, the relative sizes of the heat exchangers were rebalanced to achieve the maximum COP at a constant cost. Finally, using compressor vaned diffuser performance data, the performance potential of the conceptual design was predicted.

1.4.5.1 Computer Program

The previous computer model of the cycle, RANKØ6, was revised to allow calculation of off-design performance of the cycle based on inputted UA values and source and sink fluid stream characteristics. In this program called RANK1Ø, actual heat exchanger designs were modeled assuming a constant UA value over the cycle's operating range. A copy of RANK1Ø is listed in Appendix A1.4. The iterative method used to analyze the heat exchangers was based on a comparison of enthalpy changes for the fluid streams involved with the heat transfer between them (using log-mean or average temperature differences). The example of an iterative solution shown in Figure 1.4-26 is for the evaporator, but is similar to the convergence method used for each of the other heat exchangers.

For assumed values of compressor and turbine efficiency, this convergence method will calculate a balance point for each of the heat exchangers separately. However, since the input temperature(s) for each of the heat exchanger subroutines (i.e., evaporating temperature) is dependent on the characteristics of the other components, convergence of these input temperatures must also be insured. This was accomplished by converging on the condensing, boiling and evaporating temperatures within specified tolerances in the manner shown in Figure 1.4-27.

FIGURE 1.4-26 ENERGY BALANCE ON EVAPORATOR

FIGURE 1.4-27 TEMPERATURE CONVERGENCE FOR EVAPORATOR, BOILER AND CONDENSER.

FIGURE 1.4 - 28
R-113 VANELESS DIFFUSER COMPRESSOR MAP

FIGURE 1.4 - 29
R-113 TURBINE MAP

FIGURE 1.4 - 30

R-113 VANED DIFFUSER MAP

The estimation of off-design operating performance for the combined Rankine-vapor compression cooling system required actual turbine and compressor efficiency values at the anticipated cycle operating states and flow conditions. Performance maps of this nature were supplied by Mechanical Technology, Inc. (MTI) for a turbine and both a vaned and vaneless diffuser compressor using the refrigerant R-113. These turbine and compressor maps are shown as Figures 1.4-28, 1.4-29 and 1.4-30.

Equations were curve fit to these maps using the proprietary computer program RDCI9040 MULREG, which is a multiple regression analysis routine developed by the Carrier Research Division. MULREG has the flexibility of being able to curve fit a function of up to 50 variables based on up to 300 observations. MULREG also offers transform functions (allowing the combination of two or more variables) which were required for the complex turbine and compressor maps. The form of the equations that were developed were as follows:

1. Shaft Speed (SPEED) as a function of compressor pressure ratio (PRC) and flow factor (TPHIC).
2. Compressor adiabatic efficiency (ETACOM) as a function of shaft speed and compressor flow factor.
3. Turbine flow factor (TPHIT) as a function of turbine expansion ratio (EXPRAT) and shaft speed.
4. Turbine adiabatic efficiency (ETAEXP) as a function of shaft speed and turbine expansion ratio.
5. Compressor surge line flow factor (FLOS) as a function of shaft speed.
6. Compressor choke line flow factor (FLOC) as a function of shaft speed.

The compressor surge and choke lines were represented as a function of only one variable, the shaft speed, since surge and choke represent the limiting lines on the map and are not a surface.

The derived multiple regression equations for the MTI turbo-compressor using R-113 as a working fluid are as follows:

VANELESS DIFFUSER:

$$\begin{aligned} \text{SPEED} = & 49.640854 (\text{COMRAT}) - 6.8966733 (\text{COMRAT})^2 + 4.8063214 (\text{TPHIC}) \\ & + 0.36910227 (\text{COMRAT})^3 + 0.43645545 \times 10^{-2} (\text{TPHIC})^3 - 3.9063458 (\text{TPHIC})(\text{COMRAT}) \\ & + 0.65464123 (\text{TPHIC})(\text{COMRAT})^2 - 0.035841316 (\text{TPHIC})(\text{COMRAT})^3 \\ & + 0.19012101 (\text{TPHIC})^2 - 0.020666998 (\text{COMRAT})(\text{TPHIC})^2 - 11.188171 \end{aligned}$$

$$\text{Standard error of SPEED} = 0.276$$

$$\begin{aligned} \text{ETACOM} = & 0.36115868 \times 10^{-3} (\text{SPEED}) - 0.80214901 \times 10^{-6} (\text{SPEED})^3 \\ & - 0.44141320 \times 10^{-2} (\text{TPHIC})^2 - 0.038236016 (\text{TPHIC}) \\ & + 0.15711980 \times 10^{-2} (\text{SPEED}) \text{ TPHIC} + 0.73226555 \end{aligned}$$

$$\text{Standard error of ETACOM} = 0.006$$

$$\text{FLOS} = 0.37678571 \times 10^{-2} (\text{SPEED})^2 - 0.43225000 (\text{SPEED}) + 15.542857$$

$$\text{Standard error of FLOS} = 0.052$$

$$\text{FLOC} = 0.21428571 \times 10^{-3} (\text{SPEED})^2 + 0.1490000 (\text{SPEED}) - 2.9257143$$

$$\text{Standard error of FLOC} = 0.124$$

VANED DIFFUSER:

$$\begin{aligned} \text{SPEED} = & -2.1890225 (\text{COMRAT})(\text{TPHIC}) - 0.01877252 (\text{COMRAT})^2 (\text{TPHIC})^2 \\ & + 0.14957566 \times 10^{-3} (\text{TPHIC})^3 - 3.1701447 (\text{COMRAT})^2 \\ & - 0.21440066 (\text{COMRAT})(\text{TPHIC})^2 + 0.71320934 (\text{TPHIC}) (\text{COMRAT})^2 \\ & + 0.79918471 \times 10^{-2} (\text{TPHIC})^3 (\text{COMRAT}) - 0.13495259 (\text{COMRAT})^3 \\ & + 29.6013 (\text{COMRAT}) + 0.52496209 (\text{TPHIC})^2 \\ & + 2.3811324 (\text{TPHIC}) + 10.682561 \end{aligned}$$

$$\text{Standard error of SPEED} = 0.278$$

$$\begin{aligned} \text{ETACOM} = & 0.13776620 \times 10^{-3} (\text{TPHIC})^5 + 1123.0762 \text{ LN}(\text{TPHIC}) \\ & - 0.27962557 \times 10^{-8} (\text{SPEED})^5 - 13.526584 (\text{SPEED}) \text{ LN}(\text{TPHIC}) \\ & + 0.11515365 (\text{SPEED} - 100)^2 - 11.0779 (\text{TPHIC})^2 \\ & - 0.54027705 \times 10^{-3} (\text{TPHIC}) (\text{SPEED})^2 + 30.433321 (\text{SPEED}) \sqrt{\text{TPHIC}} \\ & + 0.10224163 (\text{SPEED}) (\text{TPHIC})^2 - 0.12349022 (\text{SPEED})^2 \\ & + 0.68265586 \times 10^{-6} (\text{SPEED})^4 + 448.83442 (\text{TPHIC}) \\ & - 0.15341009 \times 10^{-2} (\text{SPEED}) (\text{TPHIC})^3 - 0.75810083 \times 10^{-2} (\text{TPHIC})^4 \\ & - 2614.6447 \sqrt{\text{TPHIC}} - 4.9345875 (\text{TPHIC}) (\text{SPEED}) \\ & + 0.30588810 (\text{TPHIC})^3 + 863.57416 \end{aligned}$$

$$\text{Standard error of ETACOM} = 0.006$$

$$\begin{aligned} \text{FLOS} = & 0.35089286 \times 10^{-2} (\text{SPEED})^2 - 0.381375 (\text{SPEED}) \\ & + 13.041429 \end{aligned}$$

$$\text{Standard error of FLOS} = 0.048$$

$$\begin{aligned} \text{FLOC} = & 0.12625 \times 10^{-2} (\text{SPEED})^2 - 0.040367857 (\text{SPEED}) \\ & + 4.6631429 \end{aligned}$$

$$\text{Standard error of FLOC} = 0.033$$

FOR BOTH TYPES OF COMPRESSORS:

$$\begin{aligned} \text{TPHIT} = & 0.065851817 (\text{EXPRAT}) - 0.70576625 \times 10^{-6} (\text{SPEED})^3 \\ & - 0.37994856 \times 10^{-3} (\text{SPEED})(\text{EXPRAT}) - 0.010958649 (\text{EXPRAT})^2 \\ & + 0.88975668 \times 10^{-3} (\text{EXPRAT})^3 - 0.24278706 \times 10^{-4} (\text{EXPRAT})^4 \\ & + 0.44840648 \times 10^{-5} (\text{EXPRAT})(\text{SPEED})^2 - 0.11235189 \times 10^{-6} (\text{SPEED})^2 (\text{EXPRAT})^2 \\ & + 0.15297192 \times 10^{-3} (\text{SPEED})^2 - 0.012337200 (\text{SPEED}) + 0.57105446 \end{aligned}$$

Standard error of TPHIT = 0.003

$$\begin{aligned} \text{ETAEXP} = & -0.45256894 \times 10^{-4} (\text{EXPRAT})^4 + 0.11998145 \times 10^{-4} (\text{SPEED})(\text{EXPRAT})^3 \\ & + 0.34251545 \times 10^{-7} (\text{SPEED})^4 + 0.14180358 \times 10^{-2} (\text{SPEED})^2 \\ & + 0.89071936 \times 10^{-3} (\text{EXPRAT})^3 + 0.58462175 \times 10^{-5} (\text{EXPRAT})(\text{SPEED})^2 \\ & + 0.79280863 \times 10^{-2} (\text{EXPRAT})^2 + 0.42738345 \times 10^{-2} (\text{SPEED})(\text{EXPRAT}) \\ & - 0.44081447 \times 10^{-3} (\text{SPEED})(\text{EXPRAT})^2 - 0.22211429 (\text{EXPRAT}) \\ & - 0.12202305 \times 10^{-4} (\text{SPEED})^3 - 0.077552288 (\text{SPEED}) + 2.5145707 \end{aligned}$$

Standard error of ETAEXP = 0.003

A relatively simple iteration scheme, presented in Figure 1.4-31 was developed to converge on values of ETACOM, ETAEXP and TONS using these equations. The accuracy of this routine is dependent on inputted convergence tolerances, DELT, TDEL and DELTON

1.4.5.2 Predicted R-113 Chiller Performance Map

Predicted performance data were calculated for the range of operating conditions expected, using the cycle program RANK10. Heat exchangers were sized using the "UA" and pressure drop data calculated under Section 1.4.1, and summarized in Table 1.4-12. Compressor performance for a 40° backward curved, 20° inlet guide vane, vaneless diffuser design was used. Compressor and turbine maps are shown on Figures 1.4-28 and 1.4-29. The ranges of operating conditions explored were:

	<u>Min.</u>	<u>Rated</u>	<u>Max.</u>
Leaving Chilled Water	40°F	44°F	50°F
Entering Solar Water	265°F	291°F	310°F
Entering Condenser Air	70°F	95°F	115°F

The rated conditions were determined first, by varying entering solar water until unit cooling capacity was equal to 25 tons at 95°F entering condenser air and 44°F leaving chilled water. A computer printout of this run, shown in Figure 1.4-32, gives all of the heat quantities and cycle state points. The design produces 25 tons at an OCOP of 0.68 with 291°F entering solar water. Shaft speed was 20,300 rpm (100.9%). The turbine was operating in a choked flow condition at a flow factor, ϕ , of 0.325. The flow factor for either the turbine or the compressor is defined as:

FIGURE 1.4-31 OFF - DESIGN CONVERGENCE ROUTINE

TABLE 1.4-12

HEAT TRANSFER DATA USED IN CYCLE COMPUTATIONS

<u>Heat Exchanger</u>	<u>U</u>	<u>UA</u>	<u>Pressure Drop</u>	
	<u>Btuh ft² F⁻¹</u>	<u>Btuh F⁻¹</u>	<u>Outside, psi</u>	<u>Inside, psi</u>
Boiler	109	32,700	0	0.94
Preheater	31.85	5,064	0.7	1.27
Cooler	90.4	35,260	0	0.72
Condenser	3.44	58,426	0.01	0.09
Regenerator	122.7	2,908	0.32	3.86
LSHX	69.7	1,805	0.05	1.76

FIGURE 1.4-32

BY SYSTEMS DIVISION
RUN NO.5743 DATE 3/27/78

RANK10 DUAL LOOP RANKINE TURBO-COMPRESSOR CYCLE WITH COMMON CONDENSER, REGENERATOR AND LSHX

INPUT PARAMETERS: REFRIGERANT R 113 CONVERGENCE TOLERANCES 0.01 %
0.010 F

TSOL	GPMS	TAIR	CFM	ECNT	GPMON
291.000	75.000	95.000	42050.000	53.600	62.500
TSUB	TSUP	TSUPHT	ETAPUM	ALOSS	
1.000	1.000	0.0	0.500	0.020	
PREG	PULSH	PLLSH			
0.315	0.050	1.760			
UAHX	UALSHX	UAPRHT	UABOIL	UACR	UAUEAP
2908.0000	1805.000	5064.000	32700.000	63000.000	35260.000

INITIAL VALUES:

TB	TC	TE	ETREXP	ETACOM
280.000	116.500	40.000	0.830	0.750
ETALSH	ETAHX	TONS		
0.850	0.800	25.000		

CALCULATED PERFORMANCE:

RANKINE CYCLE:

QBOILR	QBOIL	QPRHT	FPRHT	QHX	ETAHX
273387.375	443436.000	170048.500	38.348	72906.000	0.726
WPC	EXPRAT	ETAT2	QPUMP	DISPLP	PHIT
239.676	9.358	0.165	1.290	0.042	0.0129

COMPRESSION CYCLE:

QEVAP	TONS	QLSHX	QCOMP	COMRAT	PHIC
300000.000	25.000	44166.187	28.101	5.823	0.5028
COP	WRF	ETALSH			
4.125	100.201	0.706			

OVERALL CYCLE:

QCOND	QCOF	TB	TC	TE
743434.937	0.677	278.589	116.857	39.408
TSOL2	TSOL3	CNT	TA	
283.789	279.175	44.000	111.370	

ITERATION COUNTERS:

CL	CH	CC	CR	CF	CT	CLS
42.0	110.0	395.0	144.0	44.0	37.0	36.0

CYCLE STATE POINTS:

NUMBER	TEMPERATURE	PRESSURE	ENTHALPY	VOLUME	ENTROPY
1	168.277	139.753	44.706		0.051
14	278.589	139.753	73.086	0.013	0.093
2	278.589	139.753	118.712	0.242	0.189
4	167.398	14.934	104.044		0.189
5	117.214	14.619	95.968	2.181	0.176
6	102.526	14.934	106.538	2.397	0.193
7	115.857	14.522	32.539		0.032
71	75.635	12.762	23.536	0.0	
8	115.857	139.753	32.539		0.031
9	39.408	2.615	23.536	0.0	0.050
101	40.418	2.615	84.688	10.858	0.173
10	99.690	2.565	93.691	12.410	0.190
11	170.934	14.934	104.625		0.190
12	194.686	14.934	108.561	2.446	0.196
13	188.012	14.934	107.449	2.419	0.194
15	147.147	14.619	100.759	2.305	0.184

READY

$$\phi = W * \sqrt{T/P}$$

where: W = mass flow, lb/sec
T = absolute entering temperature, R
P = entering pressure, psia.

Figure 1.4-33 shows the effect of varying entering solar water on performance. As water temperature increases, cooling capacity increases and OCOP decreases ($\phi = 0.325$). The reason for this is that as the boiling temperature is raised, the work output of the turbine increases, which in turn increases shaft speed, compressor flow, and resulting cooling capacity. However, the heat rejection must also increase, raising the condensing temperature, and the evaporator temperature must drop. Thus, the vapor compression cycle operates at a higher lift and a lower COP, more than offsetting the gain in the Rankine cycle, and dropping the OCOP.

The turbine size can be reduced for higher entering temperatures, to hold the shaft speed constant. The vapor compression cycle will then remain fixed, and the OCOP will increase as the Rankine cycle improves. Figure 1.4-33 also shows the performance of a machine with a turbine sized to a flow factor of 0.246. This sizing produces 25 tons at 310°F entering water with an OCOP of 0.71. The only difference between these two designs is the turbine size; i.e., the area of the nozzles and the admission area of the wheel. Figure 1.4-34 further explores the effect of turbine sizing. The OCOP increases with entering solar water temperature as the turbine size, ϕ , is decreased to hold cooling capacity at 25 tons.

The performance was explored at off-design conditions using the original turbine size, $\phi = 0.325$, which gives rated capacity at 291°F. Figure 1.4-35 shows the predicted capacity and OCOP over a range of chilled water temperatures and entering condenser air temperatures at 291°F entering solar water. It can be observed that the maximum condenser air temperature is limited by the surge line of the compressor, while the lower limit is determined when the compressor flow chokes, with subsequent increase in speed and decrease in efficiency. Surge and choke limits are shown as dotted lines. Thus, the useful operating range of condenser temperature for fixed solar temperatures is limited to about 16°F for this compressor design. Variations in chilled water temperature have a modest effect on this range.

To complete the off-design performance, Figures 1.4-36 and 1.4-37 show predicted performance at 265°F and 310°F respectively. Again, the useful range of condenser air temperatures remains about 16°F; however the temperature level shifts directly with solar temperature.

The strong dependence of the usable envelope as related to condenser air temperature and entering solar temperature is better shown on Figure 1.4-38. These data are for 44°F leaving chilled water and would shift $\pm 1\frac{1}{2}$ tons over the range of 40 to 50°F chilled water temperatures. The implications of this rather limited performance envelope are significant in regard to controlling the machine over the range of conditions expected in the field. Obviously, to operate at higher temperatures requires hotter water. Even at 320°F water, this design would be limited to slightly over 105°F air,

FIGURE 1.4 - 33

R-113 RANKINE CYCLE

COMPARATIVE PERFORMANCE WITH TWO TURBINE SELECTIONS

FIGURE 1.4 - 34
R-113 RANKINE CYCLE
PERFORMANCE AS A FUNCTION OF TURBINE SELECTION
FOR 25 TON CAPACITY

FIGURE 1.4 - 35
R-113 PREDICTED SYSTEM PERFORMANCE
AT 291 F SOLAR WATER

FIGURE 1.4 -36
R-113 PREDICTED SYSTEM PERFORMANCE
AT 265 F SOLAR WATER

FIGURE 1.4 - 37
R-113 PREDICTED SYSTEM PERFORMANCE
AT 310 F SOLAR WATER

FIGURE 1.4 - 38

R-113 PREDICTED SYSTEM PERFORMANCE AS A FUNCTION OF AMBIENT AIR TEMP.
(FOR 44 F LEAVING CHILLED WATER)

while at the lower end, water temperatures must be reduced to sustain operation within the surge choke limits. This suggests that the supply water temperature or flow will have to be modulated to operate across the maximum range of outdoor air temperatures. A constant head pressure control would provide a lower limit to this requirement. The maximum ambient could be raised by a number of strategies:

- A larger condenser
- A larger turbine and lower design water temperature
- Revised compressor characteristics for a wide surge choke range.

Another approach to operation at higher ambients is to switch from turbine to electric motor operation whenever solar water temperatures are not high enough to handle the ambient conditions. While this seems to be self-defeating, ambients above 105°F are peaking conditions which occur briefly during relatively few days in some locations. If solar heat is stored during motor operation, solar operation could resume when ambients return to the operating region, and solar utilization should be unaffected.

A final approach to higher ambients would be to switch on the electric motor and load share with the turbine. This will be thoroughly investigated when part load motor efficiencies are available for analysis.

1.4.5.3 Comparison of R-113 and R-114 Machines

Limited performance data on R-114 were calculated using the same heat exchanger "UA" and pressure drop data used in the R-113 computations in the preceding section. This provides a comparison of equal cost machines. The compressor and turbine maps for R-114 are shown in Figures 1.4-39 and 1.4-40. The compressor is a 40° backward curved wheel with 20° inlet guide vanes, and a vaneless diffuser. The entering solar water temperature which produced 25 tons at 95°F air and 44°F leaving chilled water was found to be 281.9°F, with an OCOP of 0.62. The range of ambient from surge to choke conditions was found to be 12.5°F (103°F to 90.5°F), slightly less than for R-113. These data are compared to R-113 in Figure 1.4-41. The computer printout for the rating point is Figure 1.4-42, which gives heat quantities, and state points.

As pointed out in the conclusions of Section 1.3, the matching of the turbine and compressor is better for R-114 than for R-113. This is born out by comparing the turbine and compressor efficiencies of R-114 in Figure 1.4-42, 0.856 and 0.784 respectively, with the efficiencies of R-113 in Run 684, 0.831 and 0.758. The 7% lower efficiency product of R-113 is more than offset by its superior thermodynamic cycle performance which results in an OCOP of 0.68.

Another salient point is that the pump horsepower (QPUMP) is 5.5 hp for R-114 compared with 1.29 hp for R-113. Using the Figure of Merit developed under Section 1.2.1.4, the two refrigerants compare as follows:

- R-113 Figure of Merit = 0.83
- R-114 Figure of Merit = 0.66

FIGURE 1.4 - 39

R-114 VANELESS DIFFUSER COMPRESSOR MAP

FIGURE 1.4 - 40

R-114 TURBINE MAP

FIGURE 1.4 - 41
COMPARISON OF PERFORMANCE
WITH R-113 AND R-114 WORKING FLUID

FIGURE 1.4 - 42

ENERGY SYSTEMS DIVISION
RUN NO.7511 DATE 5/ 2/78

RANK10 DUAL LOOP RANKINE TURBO-COMPRESSOR CYCLE WITH COMMON CONDENSER, REGENERATOR AND LSHX

INPUT PARAMETERS: REFRIGERANT R 114 CONVERGENCE TOLERANCES 0.01 %

0.010 F 0.010 TONS

TSOL	GPMS	TAIR	CFM	CWT	GPMCW
291.900	75.000	95.000	42050.000	44.000	62.500
TSUB	TSUP	TSUPHT	ETAPUM	ALOSS	
1.000	1.000	0.0	0.500	0.020	
PREG	PULSH	PLLSH			
0.315	0.050	1.760			
UAHX	UALSHX	UAPRHT	UABOIL	UACR	UAEUAP
2908.000	1805.000	5064.000	32700.000	63000.000	35260.000

INITIAL VALUES:

TB	TC	TE	ETAEXP	ETACOM
270.000	119.000	39.500	0.000	0.750
ETALSH	ETAHX	TONS		
0.740	0.000	25.000		

CALCULATED PERFORMANCE:
RANKINE CYCLE:

QBOILER	QBOIL	QPRHT	FPRHT	QHX	ETAHX
208429.187	483733.562	275304.250	56.912	59641.301	0.623
WRC	EXPRAT	ETAT2	QPUMP	ETAEXP	TPHIT
320.245	6.064	0.151	5.020	0.856	0.1613

COMPRESSION CYCLE:

QEUAP	TONS	QLSHX	QCOMP	COMRAT	TPHIC
300000.000	25.000	55958.312	28.076	4.140	3.8055
COP	WRF	ETALSH	ETACOM	ATPHIC	
4.199	245.595	0.702	0.784	2.706	

OVERALL CYCLE:

QCOND	QCOMP	TB	TC	TE
783738.312	0.620	270.101	118.215	39.400
TSOL2	TSOL3	ECWT	TA	
276.342	269.000	53.600	112.258	
SPEED	TPHIC	FLOS	FLOC	
100.092	3.806	3.122	4.137	

ITERATION COUNTERS:

CL	CH	CC	CB	CE	CT	CLS
56.0	74.0	285.0	265.0	7.0	51.0	48.0

CYCLE STATE POINTS:

NUMBER	TEMPERATURE	PRESSURE	ENTHALPY	VOLUME	ENTROPY
1	145.960	372.593	43.588		0.086
14	270.101	372.593	77.975	0.0	0.138
2	270.101	372.593	104.009	0.066	0.171
4	150.750	61.444	93.385		0.171
5	118.324	61.129	87.368	0.526	0.161
6	158.837	61.444	94.906	0.575	0.174
7	117.215	61.029	36.139		0.073
71	81.268	59.269	27.025	0.011	0.056
8	117.215	372.593	36.139		0.073
9	39.408	14.891	27.025	0.0	0.057
101	40.408	14.891	75.886	2.024	0.155
10	94.344	14.841	85.000	2.276	0.172
11	154.531	61.444	94.095		0.172
12	169.237	61.444	96.870	0.588	0.177
13	163.361	61.444	95.758	0.581	0.175
15	140.832	61.129	91.542	0.553	0.168

READY

1.4.5.4 Load Lines and Control Strategy

Analysis of the predicted system performance map, Figure 1.4-38 has given some insight into a control strategy for the system when operating on solar power; i.e., turbine drive. Because of the limited range of outdoor ambients between surge and choke for a given solar temperature, the strategy required to operate at outdoor ambients below the choke limit is to throttle the flow or modulate the temperature of the solar water into the boiler in relation to the outdoor ambient. Under this type of control, the system will operate along some load line superimposed on the performance map. Figure 1.4-43 shows the map with two arbitrarily drawn load lines. Any suitable line could be drawn on the map. Load line A is a constant cooling capacity line, while load line B varies capacity with outside ambient to match the building load. The intercept of these lines with the solar temperature lines gives the required control function of supply water vs outdoor ambient, shown on Figure 1.4-44.

Therefore, the set-up and operation of this machine will be very flexible. Constant capacity may be advantageous with cold storage, while variable capacity may be advantageous with hot storage to minimize on/off cycling of the unit by matching the building load line. These are features that can only be evaluated through system simulation.

Operation at ambients below the head pressure control setting would produce a constant capacity because the air flow over the condenser is staged or modulated to hold the condensing temperature constant. As long as the load line does not cross into the choke or surge regions, the constant temperature setting, and consequently the minimum solar water temperature, can be as low as practical. The limiting condition would probably be when refrigerant flow to the cooler could not be maintained due to low pressure difference. When the temperature of the solar water is below the minimum for a given ambient, and cooling is required, the electric motor will be turned on to drive the compressor with the vapor supply to the turbine closed. Although load sharing between the motor and turbine is an obvious possibility, the economic feasibility on this system cannot be determined until final motor efficiency characteristics are known.

1.4.5.5 Electric Motor Performance Requirements and Control

Shaft power requirements of the compressor at rated and off-design conditions were predicted using the cycle program RANK10, modified to delete the Rankine cycle operation. The compressor performance map used in this program is shown as Figure 1.4-45 with the 25 ton rating point indicated.

Pressure ratios for off-design operating conditions range from 2.57 at 75°F ambient air to 7.96 at 115°F ambient air. Examination of Figure 1.4-45 clearly shows that variable speed operation is required to cover these conditions. The pressure ratio range of the compressor at a constant speed only covers one-fifth of this range.

FIGURE 1.4 - 43

R-113 PREDICTED SYSTEM PERFORMANCE WITH LOAD LINES
(AT 44°F LEAVING CHILLED WATER)

FIGURE 1.4 - 44

CONTROL FUNCTIONS

FIGURE 1.4 - 45

R-113 VANELESS DIFFUSER COMPRESSOR MAP
WITH OPERATING LINE FOR MOTOR SPEED CONTROL

An obvious control strategy would be to vary the speed in relation to the compression ratio or the compressor flow in a manner which avoids the surge or choke limits. However, from a practical standpoint, accurate measurement of either of these variables within a control system would be very difficult and most likely expensive. It was found that the difference between saturated condensing and evaporating temperatures relate directly to compressor speed, as shown on Figure 1.4-46. These data translate to the operating line indicated on the compressor map, Figure 1.4-45. It is felt that modulating the motor speed as a function of temperature difference measurement can readily be accomplished and would provide a practical control system.

Using this control line, electrically driven performance was predicted at various ambient air and chilled water temperatures. Figure 1.4-47 compares shaft horsepower, cooling capacity and speed as a function of entering condenser air temperature (ambient), over the range from 75°F to 115°F. For temperatures below 75°F, a condenser head pressure control would modulate fans to maintain a fixed condensing temperature. Thus, the capacity speed and horsepower would remain constant below 75°F.

The compressor motor watts cannot be predicted at this time, because final motor characteristics are not available. The motor efficiency goal at rated conditions is 85%. On this basis, the 22.75 compressor hp translates to 19.97 kW, which is an EER* of 15.0. This compares favorably with rated performance of commercially available air cooled reciprocating water chiller compressors. EER's of these machines range from the 7.5 minimum established by ASHRAE Standard 90 up to approximately 10.0. The fact that there will be some increase in EER's during the next decade should be taken into consideration when making this comparison. The reason for the superior performance of the proposed design under motor operation is the oversized condenser available to the compressor when the Rankine power cycle is not energized. At rated conditions, the predicted condensing temperature is 105.6°F under motor power versus 116.8°F under Rankine turbine power. The EER difference is not as great when condenser fan power is included. Table 1.4-13 compares the two designs on an equivalent basis, with condenser fan power included.

Speed torque specifications for the motor were developed from the data in Figure 1.4-47 and are presented in Figure 1.4-48. This shows the envelope of operation anticipated, with the limits established by operating controls. There will be some operation at entering chilled water temperatures above 60°F during start-up or pull-down of chilled water temperature. Normal operation should be between 50°F and 60°F.

This type of machine exhibits the interesting characteristic of increasing capacity with outdoor ambient. This characteristic results from the need to increase speed as the pressure ratio increases. Conventional, fixed speed machines decrease in capacity as the ambient increases.

* EER = Energy Efficiency Ratio = Cooling Capacity, Btu/hr/watts input.

FIGURE 1.4 - 46

SPEED CONTROL PARAMETER FOR MOTOR ONLY OPERATION

FIGURE 1.4 - 47

MOTOR DRIVEN COMPRESSOR SYSTEM PERFORMANCE
HORSEPOWER, SPEED & CAPACITY

TABLE 1.4-13

COMPARISON OF ELECTRICALLY DRIVEN DESIGN WITH A
CONVENTIONAL AIR COOLED CHILLER

	<u>Rankine</u>	<u>Conventional</u>
Condensing Temp., °F	105.6	116.8
Condenser Air Flow, cfm	42,400	25,000
Condenser Fan Power, kW	5.1	2.6
Compressor Power, kW	20.0	28.0
Total Power, kW	25.1	30.6
Capacity, MBTUH	300	300
EER	11.9	9.8

FIGURE 1.4 - 48

25 TON R-113 CENTRIFUGAL COMPRESSOR
MOTOR SPEED - TORQUE REQUIREMENTS

The increase in capacity with ambient would tend to match the natural load line of a building, which would be advantageous in reducing equipment cycling and providing better control of conditions.

The disadvantage of this characteristic is that the peak horsepower required from the motor is much larger than for a conventional 25 ton unit. A variable speed motor is essentially a constant torque motor sized for the maximum horsepower required. Therefore, referring to Figure 1.4-48, the motor must be sized to produce a torque of 9.2 ft-lbs at 112% of speed, or a peak power of 40 hp, at 115°F ambient and 50°F entering chilled water. Horsepower delivered on conventional fixed speed machines at maximum ambient conditions may not be significantly larger than at rated conditions, allowing a smaller motor and power supply for a given capacity.

1.4.5.6 Overall System Performance Maps for Simulations

Predicted performance data for motor operation and turbine operation were combined in tabular form, for use in the system simulation program. These tables covered the range of solar water, outdoor air, and entering chilled water temperature anticipated under simulated conditions. Data for both pump watts and condenser fan watts were included.

Figure 1.4-49 shows the combined cooling capacity and O COP data used for these tables. The tabular data are presented in Tables 1.4-14, pages 1-5, where page 5 is for motor only operation.

1.4.5.7 Cost/Performance Optimization

Optimization of the various components for the Rankine/Vapor Compression Chiller is essential due to high costs of heat exchangers and solar collectors. As with other types of power and vapor compression systems, there is a point at which it becomes uneconomical to increase heat exchanger size for small increases in cycle efficiency. When these systems are used in conjunction with solar collectors however, the high cost of solar collectors as compared to conventional heat exchangers (boilers, condensers, evaporators, etc.) requires seemingly oversized heat exchangers to be used for a cost effective design. For this reason, the optimization method used was to size the heat exchangers based on preliminary computer simulations; obtain a cost estimate of both fixed and variable costs for each heat exchanger; and then to adjust heat exchanger sizes to obtain maximum performance at a fixed cost. After the heat exchangers were optimized, the turbo-compressor wheel sizes were readjusted for optimum efficiency. Turbo-compressor costs, unlike heat exchanger costs, were assumed constant over the optimization range.

Using preliminary computer runs of the RANKØ6 program, initial heat exchanger UA values, pressure drops and costs were estimated based on expected costs for manufacturing materials and refrigerant charge. As shown in Table 1.4-11, the final costs were broken up into fixed and variable costs. The variable costs were assumed to change linearly with UA, making it possible to use different UA values and still have an accurate estimate of system costs.

FIGURE 1.4 - 49

PREDICTED SYSTEM PERFORMANCE
CONSTANT CAPACITY CONTROL

TSOL 310

TABLE 1.4-14
R-113 RANKINE CYCLE PREDICTED PERFORMANCE

Page 1 of 5

	ECWT		50		KW				55				60			
	TAIR	** TONS	OCOP	W _M	* W _P	W _F	TONS	OCOP	W _M	W _P	W _F	TONS	OCOP	W _M	W _P	W _F
HEAD PRESS CONTROL ↓	25	23.8	.928	0	.99	1.33	25.6	.997	0	.99	1.33	27.5	1.06	0	.99	1.33
	50	23.8	.928	0	.99	2.68	25.6	.997	0	.99	2.68	27.5	1.06	0	.99	2.68
	75	23.8	.928	0	.99	4.0	25.6	.997	0	.99	4.0	27.5	1.06	0	.99	4.0
	75	23.8	.928	0	.99	4.0	25.6	.997	0	.99	4.0	27.5	1.06	0	.99	4.0
TURBINE ONLY ↙	80	23.8	.85	0	1.12		25.6	.916	0	1.12		27.4	.97	0	1.12	
	85	23.8	.775	0	1.24		25.5	.834	0	1.24		27.2	.89	0	1.24	
	90	23.7	.705	0	1.39		25.5	.757	0	1.39		27.1	.81	0	1.39	
	95	23.7	.64	0	1.55		25.4	.685	0	1.55		27.0	.73	0	1.35	
	100	23.6	.58	0	1.74		25.4	.615	0	1.74		26.9	.665	0	1.74	
	105	23.6	.525	0	1.99		25.3	.56	0	1.99		26.8	.61	0	1.99	
MOTOR ONLY ↑	105	25.7	--	27.7	0		27.0	--	26.3	0		28.5	--	24.1	0	
	110	26.5	--	31.3	0		27.7	--	29.8	0		29.25	--	28.4	0	
	115	27.0	--	35.0	0	4.0	28.5	--	33.4	0	4.0	30.0	--	31.7	0	4.0
HIGH TEMPERA- TURE CUT OFF →	115	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0

NOTE:

GPM SOLAR WATER -- 75.0

GPM CHILLED WATER-- 62.5

CFM COND. AIR --- 42000

MIN TEMP. ECWT --- 50 F --- OFF FOR FREEZE PROTECTION

MAX-AMBIENT --- 115 F

LOAD LINE---CONSTANT COOLING CAPACITY AT 291 F TSOL

W_P @ MOTOR EFF'Y = .75; FIXED DISPLACEMENT PUMP
 * W_M @ " " = .85

** {TURBINE & COMPRESSOR MAPS DATED 2/22/78 & 2/24/78(resp.)
 {RANKIØ PROGRAM, RUNS 574-642

ESD
 3/31/78

TSOL 290

TABLE 1.4-14 (Cont.)

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	ECWT						50						55						60					
	TAIR	T	OCOP	W _M	W _P	W _F	T	O	W _M	W _P	W _F	T	O	W _M	W _P	W _F								
HEAD PRESS CONTROL	25	23.8	.928	0	.99	1.33	25.6	.997	0	.99	1.33	27.5	1.06	0	.99	1.33								
	50	23.8	.928	}	.99	2.68	25.6	.997	}	.99	2.68	27.5	1.06	}	.99	2.68								
	75	23.8	.928		.99	4.0	25.6	.997		.99	4.0	27.5	1.06		.99	4.0								
	75	23.8	.928		.99	4.0	25.6	.997		.99	4.0	27.5	1.06		.99	4.0								
80	23.8	.85	}		1.12	}	25.6	.916		}	1.12	}	27.4		.97	}	1.12	}						
TURBINE ONLY	85	23.8		.775	1.24		25.5	.834	1.24		27.2		.89	1.24										
	90	23.8		.705	1.39		25.5	.757	1.39		27.1		.81	1.39										
	95	23.8		.64	0		0	25.4	.685		0		0	27.0	.73		0		0					
	95	24.1	—	21.1	}	25.4	—	20.0	}	26.6	—	19.0	}	27.5	—	22.0	}							
MOTOR ONLY	100	25.0	—	23.9		}	26.25	—		23.1	}	27.5		—	22.0	}								
	105	25.7	—	27.7			}	27.0		—		26.3		}	28.5			—	24.1	}				
	110	26.5	—	31.3				}		27.7		—			29.8			}	29.5		—	28.4	}	
	115	27.0	—	35.0	0				4.0	28.5		—	33.4		0		4.0		30.0		—	31.7		0
CUT-OFF	115	0																						

TABLE 1.4-14 (Cont.)

TSOL 270		50					55					60								
ECWT		T	O	WM	WP	WF	T	O	WM	WP	WF	T	O	WM	WP	WF				
↓	HEAD	25	23.8	.928	0	.99	1.33	25.6	.997	0	.99	1.33	27.5	1.06	0	.99	1.33			
	PRESS	50	23.8	.928		.99	2.68	25.6	.997		.99	2.68	27.5	1.06		.99	2.68			
	CONTROL	75	23.8	.928	{	.99	4.0	25.6	.997	{	.99	4.0	27.5	1.06	{	.99	4.0			
		75	23.8	.928		.99	4.0	25.6	.997		.99	4.0	27.5	1.06		.99	4.0			
	TURBINE	80	23.8	.85	{	1.12	{	25.6	.916	{	1.12	{	27.4	.97	{	1.12	{			
	ONLY	85	23.8	.775		0		1.24	25.5		.834		0	1.24		27.2		.89	0	1.24
		85	22.2	-	15.4	0	{	23.5	-	14.5	0	{	24.5	-	13.6	0	{			
		90	23.2	-	18.1	{		24.5	-	16.9	{		25.6	-	16.2	{				
		95	24.1	-	21.1			{	25.4	-			20.0	{	26.6			-	19.0	{
	MOTOR	100	25.0	-	23.9				{	26.25			-		23.1			{	27.5	
ONLY	105	25.7	-	27.7	{	27.0	-	26.3		{	28.5	-	24.1	{						
	110	26.5	-	31.3		{	27.7	-			29.8	{	29.25		-	28.4	{			
	115	27.0	-	35.0			0	4.0			28.5		-		33.4	0			4.0	30.0
CUT-OFF →		115	0																	

TABLE 1.4-14 (Cont.)

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TSOL 250
ECWT

50

55

60

		T	O	WM	WP	WF	T	O	WM	WP	WF	T	O	WM	WP	WF
TURBINE & HEAD PRESS CONTROL	25	23.8	.928	0	.99	1.33	25.6	.997	0	.99	1.33	27.5	1.06	0	.99	1.33
	50	23.8	.928		.99	2.68	25.6	.997	5	.99	2.68	27.5	1.06	0	.99	2.68
	75	23.8	.928	0	.99	4.0	25.6	.997	0	.99	4.0	27.5	1.06	0	.99	4.0
	75	20.0	-	10.5		4.0	21.0	-	9.9		4.0	22.0	-	9.4		4.0
	80	21.1	-	12.8	0		22.3	-	12.1	0		23.3	-	11.5	0	
	85	22.2	-	15.4	0		23.5	-	14.5	0		24.5	-	13.6	0	
	90	23.2	-	18.1			24.5	-	16.9			25.6	-	16.2		
	95	24.1	-	21.1			25.4	-	20.0			26.6	-	19.0		
MOTOR ONLY	100	25.0	-	23.9			26.25	-	23.1			27.5	-	22.0		
	105	25.7	-	27.7			27.0	-	26.3			28.5	-	24.1		
	110	26.5	-	31.3			27.7	-	29.8			29.25	-	28.4		
	115	27.0	-	35.0	0	4.0	28.5	-	33.4	0	4.0	30.0	-	31.7	0	4.0
CUT-OFF → 115		0														

TABLE 1.4-14 (Cont.)

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MOTOR ONLY ECWT		50					55					60				
TAIR	T	OCOP	W _M	W _P	W _F		T	O	W _M	W _P	W _F	T	O	W _M	W _P	W _F
HEAD PRESS CONTROL ↓	25	20.0	—	10.5	0	1.33	21.0	—	9.9		1.33	22.0	—	9.4	0	1.33
	50	20.0	}	10.5	}	2.68	21.0	}	9.9		2.68	22.0	}	9.4		2.68
	75	20.0		10.5		4.0	21.0		9.9		4.0	22.0		9.4		4.0
	75	20.0		10.5		4.0	21.0		9.9		4.0	22.0		9.4		4.0
	80	21.1		12.8			22.3		12.0			23.3		11.4		
	85	22.2		15.4			23.5		14.5			24.5		13.6		
	90	23.2		18.1			24.5		16.9			25.6		16.2		
	95	24.1		21.1			25.4		20.0			26.6		19.0		
	100	25.0		23.9			26.25		23.1			27.5		22.0		
	105	25.7		27.7			27.0		26.3			28.5		24.1		
	110	26.5		31.3			27.7		29.8			29.25		28.4		
	115	27.0	—	35.0	0	4.0	28.5	—	33.4		4.0	30.0	—	31.7	0	4.0
115	0															

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To increase chiller performance at a constant total system cost, the size of each of the heat exchangers was first varied independently to find its effect on overall system performance. The results of these runs gave an estimate of which components could give the greatest increase in OCOP and tons of cooling for the least cost.

The incremental values of $\Delta \text{COP} / \Delta \$$ and $\Delta \text{tons} / \Delta \$$ are given in Table 1.4-15 for each heat exchanger. The boiler and preheater were lumped together for this analysis. These incremental values will change with each new selection of heat exchanger sizes, but the trends, represented by the arrows in Table 1.4-15, should be valid over a wide range of size selections. It can be seen that for all components except the boiler, the OCOP increases with increasing UA. It can also be seen that the tons increases with increasing UA for all components except the LSHX and regenerator. These variations were traded off to increase the OCOP at constant tons and relatively constant cost. Figure 1.4-50 shows the optimization path resulting from the trade-offs. The data for each run is shown in Table 1.4-16. As an example of the trade-off process, the boiler/preheater size was reduced from Run 697 to Run 699 by 14% for a savings of \$236, with a reduction in capacity of only 0.3 tons. This savings was offset from Run 699 to Run 702 by increasing the LSHX, condenser, and regenerator by a total cost of \$202. Run 702 was then only \$34 less than Run 697, but produced the same cooling capacity at an OCOP which was better by 3.3%.

The OCOP of the final Run, 712, was better than the first Run, 684, by 13.2% at the same capacity and cost. This process was ended by knowledge of practical considerations. The major change was embodied in a boiler/preheater size decrease of 55%, offset by increases in the other heat exchangers. As explained previously, an increase in boiler size would tend to increase tons and to reduce OCOP unless the turbine size was adjusted downward to hold tons constant.

The optimization was continued by adjusting the turbine size to explore larger boiler/preheater sizes. Figure 1.4-51 shows how the boiler/preheater UA was varied in relation to turbine ϕ . The cost/performance results, shown on Figure 1.4-52 with data on Table 1.4-16, indicate that returning to the original boiler size would result in a further OCOP improvement of 7.8%, but would add \$900 to the original cost estimate.

The results of this optimization are an increase in OCOP of 22%, from 0.682 to 0.832 for a total cost increase of the chiller estimated at only \$900. This process can be continued by trading off against the solar collector system, but it is obvious from the shape of the curve on Figure 1.4-52 that there will be a rapidly diminishing return. Further optimization is not planned until heat exchanger designs have been reviewed and system cost data are available. Further runs under Phase I have been made with a boiler size between Runs 724 and 729, where Figure 1.4-52 crosses the constant cost line.

Table 1.4-15

Effects of Heat Exchanger Size
Variation on Overall Cycle Performance

Component	UA	TONS	OCOP	VARIABLE COSTS (\$)	$\frac{\Delta \text{COP}}{\Delta \$}$	$\frac{\Delta \text{TONS}}{\Delta \$}$
BOILER AND PREHEATER	↑	↑	↓	BOILER: 1138 PRE- HEATER 660	-5.56×10^{-6}	$+9.070 \times 10^{-4}$
EVAPORATOR	↑	↑	↑	708	$+9.887 \times 10^{-5}$	$+3.898 \times 10^{-3}$
CONDENSER	↑	↑	↑	2830.50	$+7.030 \times 10^{-5}$	$+2.961 \times 10^{-3}$
LIQUID SUCTION HEAT EXCHANGER	↑	↓	↑	103	$+9.7087 \times 10^{-5}$	-1.9417×10^{-4}
REGENERATOR	↑	↓	↑	145	$+1.9317 \times 10^{-4}$	-1.030×10^{-3}

FIGURE 1.4 - 50

HEAT EXCHANGER OPTIMIZATION RUNS FOR:

25 TON CAPACITY

289.45°F SOLAR WATER @ 75 GPM

44°F CHILLER OUTLET TEMP. @ 625 GPM

95°F AMBIENT AIR TEMP.

TABLE 1.4-16

RANKINE SYSTEM OPTIMIZATION SUMMARY

RUN #	OCOP	TONS	COST Δ \$	UA (BTU·HR ⁻¹ ·F ⁻¹)					
				BOILER	PREHEATER	CONDENSER	COOLER	REGENERATOR	LSHX
684	.682	25.0	0	32700	5064	63000	35260	2908	1805
695	.688	24.8	0	30826	4474	"	"	"	3610
696	.689	24.2	0	31412	4865	"	38786	"	1805
697	.696	25.3	0	30125	4665	"	42312	"	"
698	.697	25.1	-145	27488	4257	"	"	"	"
699	.698	25.0	-236	25829	4000	"	"	"	"
700	.703	25.0	-185	"	"	"	"	"	2708
701	.712	25.4	- 71	"	"	65533	"	"	"
702	.719	25.3	- 34	"	"	"	"	3635	"
703	.720	25.2	-106	24525	3798	"	"	"	"
704	.721	25.0	-214	22563	3494	"	"	"	"
705	.734	25.5	- 45	"	"	69300	"	"	"
706	.736	25.0	-207	19620	3038	"	"	"	"
707	.745	25.4	- 65	"	"	"	49364	"	"
708	.747	24.8	-209	17004	2633	"	"	"	"
709	.751	25.0	-152	"	"	70560	"	"	"

TABLE 1.4-16 (Cont.)
RANKINE SYSTEM OPTIMIZATION SUMMARY

RUN #	OCOP	TONS	COST $\Delta \$$	UA (BTU·HR ⁻¹ ·F ⁻¹)					
				BOILER	PREHEATER	CONDENSER	COOLER	REGENERATOR	LSHX
710	.766	25.6	+ 74	"	"	75600	"	"	"
711	.768	24.8	- 53	14715	2279	"	"	"	"
712	.772	25.1	+ 4	"	"	76860	"	"	"
719	.774	25.0	-269	9728	1506	"	"	"	"
724	.800	25.0	- 94	12916	2000	"	"	"	"
729	.819	25.0	274	19620	3038	"	"	"	"
734	.832	25.0	993	32700	5064	"	"	"	"

FIGURE 1.4 - 51

CORRESPONDING VALUES OF UA AND ϕ_t
FOR THE TURBINE AND BOILER/PREHEATER
OPTIMIZATION AT CONSTANT 25 TONS

FIGURE 1.4-52
TURBINE AND BOILER/PREHEATER OPTIMIZATION
AT CONSTANT 25 TONS

1.4.5.8 Seasonal COP Analysis

The predicted OCOP of the original proposed design at the rating point was 0.68. This is approximately equal to commercially available water cooled absorption performance. However, from Figure 1.4-49, it is apparent that this machine should give a seasonal OCOP substantially higher than the rated OCOP, because most of the seasonal operation will be at ambients below the 95°F value used for rating. It is known that absorption equipment does not exhibit as large an improvement in performance at lower ambients. Figure 1.4-53 compares the capacity and OCOP characteristics of the air cooled Rankine unit with typical 25 ton water cooled absorption chiller. In order to put the water cooled absorption data on the same basis as the air cooled Rankine data, it was assumed that the supply water would be equal to the ambient dry bulb less 10°F.

Bin data of outdoor air temperatures for Phoenix, Arizona and Columbia, Missouri were determined. Using OCOP's corresponding to each temperature bin, and operating time fractions for each bin, a weighted seasonal OCOP was determined for this Rankine design and for a typical absorption unit. These results follow and details are given in Tables 1.4-17 and 1.4-18.

	<u>Seasonal OCOP</u>	
	<u>Columbia</u>	<u>Phoenix</u>
Rankine	0.906	0.812
Absorption	0.704	0.686

These results show a significant advantage for the Rankine unit on a seasonal basis, which improves in northern latitudes. However, part of this advantage is lost because of reduced collector efficiencies at the Rankine operating temperature level. This effect has been evaluated for an evacuated collector. Figure 1.4-54 shows the all-day efficiency of a typical evacuated collector as a function of solar water temperature, computed at 40°N latitude and 30° slope on an hour-by-hour basis with clear day radiation. Assuming solar cooling system performance is approximately proportional to the product of the daily collection efficiency and the OCOP, the combined system results follow:

	<u>Solar System Overall Seasonal Performance</u>	
	<u>Columbia</u>	<u>Phoenix</u>
Rankine	0.48	0.43
Absorption	0.42	0.42

Because the Rankine unit will operate a significant portion of the time at solar temperatures below 300°F, the relative seasonal performance of the Rankine solar system may be as much as 12 to 22% better than the absorption system. This will translate into lower collector costs for the Rankine unit.

FIGURE 1.4 - 53

COMPARATIVE PERFORMANCE OF RANKINE WITH ABSORPTION

TABLE 1.4-17

COMPARISON OF RANKINE AND ABSORPTION ON SEASONAL OCOP - COLUMBIA, MISSOURI

	<u>Band</u>	<u>Freq.</u> <u>%</u>	<u>Hrs.</u>	<u>Mean</u> <u>Load</u> <u>1000(s)</u>	<u>Tons</u> <u>@ 55</u>	<u>CCOP</u>	<u>Load</u> <u>Fract.</u>	<u>Oper.</u> <u>Time</u>	<u>Ton</u> <u>Hours</u>	<u>Input</u> <u>Ton</u> <u>Hours</u>	<u>T_{sol}</u>	<u>η_c</u>
Rankine	50-60	18	1577	31	26.2	1.0	0.1	155	4,074	4,074	245	0.57
	60-70	16	1402	105	26.2	1.0	0.33	468	12,268	12,268	245	0.57
	70-80	9	788	175	26.2	1.0	0.556	438	11,492	11,492	245	0.57
	80-90	5.3	464	245	25.2	0.83	0.81	376	9,473	11,414	270	0.55
	90-100	2.4	210	310	25.2	0.68	100	210	5,292	7,782	290	0.54
Weighted Avg. $\eta_c = 0.56$												
Total								1647	42,599	47,030	Avg. COP = 0.906	
Absorption	50-60				30.7	0.707	0.084	133	4,074	5,762		
	60-70				30.7	0.707	0.285	400	12,268	17,351		
	70-80				30.7	0.707	0.475	374	11,492	16,254	@195°F Solar Water	
	80-90				30.7	0.707	0.665	309	9,473	13,399	45°F Chilled Water	
	90-100				25.5	0.682	100	210	5,355	7,852		
Total									42,662	60,618	Avg. COP = 0.704	

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TABLE 1.4-18

COMPARISON OF RANKINE AND ABSORPTION ON SEASONAL OCOP - PHOENIX, ARIZONA

	<u>Band</u>	<u>Freq.</u> <u>%</u>	<u>Hrs.</u>	<u>Load</u> <u>1000 (s)</u>	<u>Tons</u>	<u>OCOP</u>	<u>Load</u> <u>Fract.</u>	<u>Oper.</u> <u>Time</u>	<u>Cap'y</u> <u>Ton-Hr</u>	<u>Input</u> <u>Ton-Hr</u>	
Rankine	50-60	15	1314	26.2	26.2	1.0	0.08	109	2,868	2,868	
	60-70	20	1752	80	26.2	1.0	0.254	446	11,680	11,680	
	70-80	21	1840	133.4	26.2	1.0	0.424	781	20,455	20,455	317°F Solar Water
	80-90	15.5	1360	187	25.4	0.83	0.614	834	21,193	25,534	
	90-100	9.2	806	240.6	25.2	0.68	0.796	641	16,160	23,765	
	100-110	4	350	294.2	25.2	0.56	0.973	341	8,581	15,323	
Total									80,937	99,625	0.812 Avg. COP
Absorption	50-60				30.7	0.707	0.07	93	2,869	4,058	
	60-70						0.22	380	11,680	16,521	
	70-80						0.362	666	20,455	28,931	
	80-90				30.7	0.707	0.51	690	21,193	29,976	195°F Solar Water
	90-100				28.6	0.7	0.7	565	16,160	23,086	
	100-110				19.8	0.65	1.00	433	8,581	15,323	
Total									80,938	117,895	0.686 Avg. COP

FIGURE 1.4 - 54

G.E. TC100 COLLECTOR

DAILY COLLECTOR PERFORMANCE

AT 40° N. LATITUDE WITH 30° SLOPE

1.4.5.9 Addendum to Predicted Performance

All of the previous discussions and calculated results were based on an R-113 compressor with a vaneless diffuser whose performance map is given in Figure 1.4-28. Towards the end of Phase I, a compressor performance map with a vaned diffuser was calculated, Figure 1.4-30. It was available too late in the program to revise all of the chiller performance maps, but was used to compare the performance of the two compressors at the surge, choke, and rating points. It was necessary to re-match the turbine and compressor to achieve the 25 ton rating condition by scaling the turbine downwards 9% and scaling the compressor upwards 25%.

A comparison of the chiller performance with the vaned and vaneless compressors is shown in Figure 1.4-57. A computer printout at the rating point is given in Figure 1.4-56. At the rating condition, the OCOP of the chiller with the vaned compressor is 0.82, up from 0.78 with the vaneless compressor. In addition, the operating ambient range of the chiller between surge and choke points seems to have been extended about 10°F with the vaned diffuser compressor at a constant solar water temperature. Across this range of conditions, the compressor speed varies only slightly. Examination of Figures 1.4-28 and 1.4-30 shows that although the flow range between surge and choke of the vaned diffuser compressor is less at constant speed, the pressure ratio range is greater, hence the greater ambient range of the chiller.

FIGURE 1.4 - 56
ENERGY SYSTEMS DIVISION
RUN NO.765; DATE 5/ 4/78

RANK10 DUAL LOOP RANKINE TURBO-COMPRESSOR CYCLE
WITH COMMON CONDENSER AND REGENERATOR
ALSO A VARIED DIFFUSER COMPRESSOR

INPUT PARAMETERS: REFRIGERANT R 113 CONVERGENCE TOLERANCES 0.01 %
0.010 F 0.030 TONS

TSOL	GPMS	TAIR	CFM	CWT	GPMCH
290.000	75.000	95.000	51301.000	44.000	62.500
TSUB	TSUP	TSUPHT	ETAPUM	ALOSS	
1.000	1.000	0.0	0.500	0.020	
PREG	PULSH	PLLSH			
0.394	0.075	2.640			
UAHX	UALSHX	UAPRHT	UABOIL	UACR	UAEVAP
3635.000	2707.500	2278.800	14715.000	76860.000	49364.000

INITIAL VALUES:

TB	TC	TE	ETREXP	ETACOM
269.000	111.000	41.400	0.830	0.800
ETALSH	ETAHX	TONS		
0.740	0.800	25.000		

CALCULATED PERFORMANCE:
RANKINE CYCLE:

QBOILR	QBOIL	QPRHT	FPRHT	QHX	ETAHX
236236.500	365599.937	129363.375	35.384	67962.000	0.844
WRC	EXPRAT	ETAT2	QPUMP	ETREXP	TPHIT
200.856	9.363	0.167	0.973	0.830	0.2980

COMPRESSION CYCLE:

QEVAP	TONS	QLSHX	QCOMP	COMRAT	IPHIC
300161.625	25.013	43014.594	23.459	5.029	9.7657
COP	WRF	ETALSH	ETACOM	ATPHIC	
5.027	101.175	0.876	0.799	11.719	

OVERALL CYCLE:

QCOND	OCOP	TB	TC	TE
665764.750	0.821	269.196	110.969	41.508
TSOL2	TSOL3	ECWT	TA	
283.700	280.250	53.605	107.016	
SPEED	TPHIC	FLQS	FLQC	
93.631	9.766	8.095	11.951	

ITERATION COUNTERS:

CL	CH	CC	CB	CE	CT	CLS
47.0	861.0	215.0	227.0	18.0	54.0	45.0

CYCLE STATE POINTS:

NUMBER	TEMPERATURE	PRESSURE	ENTHALPY	VOLUME	ENTROPY
1	168.434	126.466	44.744		0.085
14	269.196	126.466	70.493	0.0	0.123
2	269.196	126.466	117.513	0.269	0.188
4	160.176	13.508	102.914		0.188
5	111.453	13.114	95.118	2.414	0.175
6	175.299	13.508	105.393	2.626	0.192
7	109.969	12.992	31.217		0.063
71	69.511	10.352	22.222	0.010	0.047
8	109.969	126.466	31.217		0.063
9	41.508	2.761	22.222	0.0	0.047
101	42.508	2.761	84.992	10.320	0.173
10	101.622	2.686	93.987	11.887	0.190
11	166.812	13.508	103.998		0.190
12	183.343	13.508	106.721	2.661	0.194
13	179.228	13.508	106.041	2.643	0.193
15	136.565	13.114	99.110	2.530	0.182

READY

FIGURE 1.4 - 57

COMPARISON OF VANED AND VANELESS DIFFUSER
WITH R-113 COMPRESSOR ON CHILLER PERFORMANCE