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**DEVELOPMENT OF A HIGH TEMPERATURE SOLAR POWERED WATER
CHILLER. VOLUME 1**

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

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DEVELOPMENT OF A HIGH TEMPERATURE
SOLAR POWERED WATER CHILLER

VOLUME I

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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ABSTRACT

Phase I work performed under DOE Contract EG-77-03-C-1590 presents a conceptual design of a high temperature solar powered water chiller. State-of-the-art work in the field of organic Rankine cycles is reviewed. Various Rankine cycle driven vapor compression configurations are evaluated, as well as the effect of design parameters and working fluid selection. R-113 is the recommended fluid for both cycles. A conceptual design of a hermetic high speed turbine driven centrifugal compressor is presented. Several alternate variable speed electric motor drives are described, with the high frequency induction motor which is mounted on the shaft between the turbine and compressor favored. Various aerodynamic impeller designs are evaluated to optimize the single stage R-113 compressor impeller. The performance of the chiller system over a wide range of expected operating conditions is predicted. Control strategies for the turbo/compressor/motor unit are described. System simulation of the proposed chiller for a commercial building application in two cities is described, with preliminary results. A preliminary appraisal of owner's costs is given, as well as an assessment of the environmental impact of the chiller.

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1.0.1 PROGRAM DEFINITION

This program has been undertaken in response to the Department of Energy RFP on Solar Activated Cooling Projects for Heating and Cooling Applications; June 6, 1977. Contract EG-77-C-03-1590 was subsequently awarded to the Energy Systems Division of Carrier Corporation to develop a high-temperature solar-powered water chiller in the 15 to 25 ton range, utilizing 250-300°F solar hot water. The project is to study, design and test a prototype heat engine/vapor compression air cooled water chiller which utilizes auxiliary electric energy. Energy Systems Division has subcontracted to Mechanical Technology, Inc., the study, design and construction of the turbo/compressor/motor unit.

This report covers Phase I, the conceptual design study.

1.0.2 PHASE I OBJECTIVES

1. To establish system design objectives and preliminary specifications for an air cooled, high temperature, solar powered water chiller using electric auxiliary energy.
2. To prepare a conceptual design of the turbo machinery and the total chiller package, with predicted performance and costs.
3. To evaluate the overall cost/performance of a solar heating and cooling system comprised of the solar powered chiller and selected solar system components, in three locations and two types of buildings.
4. To prepare a preliminary evaluation of the resulting design with regard to environmental impact, maintenance costs, and equipment life.

1.0.3 APPROACH

1.0.3.1 Chiller Design

The air cooled Rankine driven vapor compression cycle is an excellent choice for a solar powered water chiller for several reasons. It can develop an overall COP that is equal to or higher than presently available from water cooled absorption equipment when source temperatures are in the 250-300°F range. Evacuated collectors now on the market can efficiently provide heat in this temperature range. The Rankine system can be made to readily interface with electric power through the addition of an electric motor drive to the power shaft. A single halocarbon working fluid can be used in both cycles to simplify the design of a hermetic system.

The prime mover to be developed is a high speed turbine which directly drives a centrifugal compressor. A cycle diagram of a simple dual loop cycle with a single working fluid, Figure 1.0-1, shows the component relationships and state points basic to this development. Figure 1.0-2 is a pressure enthalpy diagram of a typical fluorinated hydrocarbon of the drying type showing the thermodynamic Rankine power cycle and Rankine vapor compression cycle with the same state points.

Vapor leaving the boiler at high pressure (2) flows to the turbine. As the high pressure vapor expands through the turbine to the lower pressure condenser (6), work is imparted to the turbine, which directly drives the compressor. The compressor removes vapor from the low pressure evaporator (10), and compresses it to the condenser pressure (12). Combined vapor from the turbine exhaust and compressor discharge is condensed to a liquid in the condenser (7). The condenser rejects the total heat from both cycles. The condensed liquid then splits into two streams, with some being pumped to the boiler (8) and the remainder expanding into the evaporator (9) where it evaporates at low pressure, removing heat (cooling) from the load.

The chiller is to be designed as an outdoor unitary package which utilizes, to the extent possible, components and currently available manufacturing technology of present outdoor hermetic air cooled packaged chillers. Table 1.0-1 lists the design features which will be goals of this program. The unitary design minimizes field installation costs and can utilize service and installation techniques and personnel currently available in the trade.

The turbo/compressor is to be designed as a hermetic unit. Use of gas bearings eliminates the need for oil within the system, simplifying design, improving heat transfer and improving the thermal stability of the working fluid. The hermetic design eliminates the need for shaft seals which could be a major service problem. Use of a refrigerant common to both cycles eliminates the need for internal seals or separation processes.

The most critical goal of the design is to achieve maximum solar system cost effectiveness. A high chiller COP will minimize collector requirements and, therefore, system costs. Although the high Rankine cycle temperature level, up to 300°F, maximizes ideal cycle efficiency potential, it reduces collector efficiency. The design goal is to optimize the collector and chiller system in this regard. Achieving a high overall cycle COP (OCOP)

FIGURE 1.0-1

SCHEMATIC DIAGRAM OF SIMPLE DUAL LOOP
RANKINE / VAPOR COMPRESSION CYCLE

FIGURE 1.0-2

PRESSURE-ENTHALPY DIAGRAM OF SIMPLE DUAL LOOP
RANKINE VAPOR COMPRESSION CYCLE

TABLE 1.0-1

HIGH TEMPERATURE SOLAR POWERED CHILLER DESIGN GOALS

1. Unitary Packaged Chiller Concept
2. Maximize Solar System Cost Effectiveness
3. High Overall Cycle COP
4. Low Parasitic Power
5. Electric Auxiliary Power
6. Air Cooled
7. 220-300°F Solar Water Supply
8. Water Chiller
9. Meets Applicable Standards
10. Meets Applicable Codes
11. Hermetic Design

requires designing a highly efficient turbine/compressor unit, designing heat exchangers which achieve very small temperature approaches at reasonable cost, selecting an optimum working fluid, and devising a cycle heat exchanger configuration which maximizes the availability of energy in the cycle.

1.0.3.2 Program Structure

In order to achieve the equipment design goals, the program has been laid out to reiterate the design process as more definitive design factors are developed. Figure 1.0-3 shows the logical flow of information from the establishment of preliminary system objectives (Task 1.1) to the final evaluation (Task 1.6), with the opportunity for reiterative design as necessary.

System objectives were established at the outset as the result of two activities; first, a literature search and state of the art assessment; and secondly, a review of the current commercial water chiller business by the Commercialization Review Board.

The initial cycle study (Task 1.2) used a simple dual loop Rankine cycle computer program to identify cycle performance as a function of operating conditions, working fluids, and heat exchanger configuration, and provided basic data for preliminary heat exchanger, and turbo/compressor design. From this data, a refrigerant selection matrix was developed to screen candidate halocarbon refrigerants as working fluids. The field was narrowed to two refrigerants, which were carried into turbo/compressor design (Task 1.3) and chiller design (Task 1.4). As information became more definitive, the Rankine cycle program was upgraded to include fixed heat exchangers, pressure drops, and finally turbo/compressor performance maps, with successively more realistic cycle data (under Task 1.2) being generated and forwarded to Tasks 1.3 and 1.4.

Once the predicted off-design performance of the chiller package and its cost estimate were known, the project moved into the simulation phase to determine the best solar system design for use with a chiller, and to compute annual operating energy usage for the system in a number of locations and building types.

1.0.3.3 Program Schedule

The period of Phase I was to be seven (7) months. The schedule for the entire program is shown on two complementary milestone charts. The first, Figure 1.0-4, shows the schedule of major tasks for the prime contractor, Energy Systems Division of Carrier Corporation. The second, Figure 1.0-5, shows the matching schedule for the turbo/compressor design tasks to be performed by Mechanical Technology, Inc.

FIGURE 1.0-3
PROGRAM STRUCTURE

FIGURE I.O-4 MILESTONE CHART

ENERGY SYSTEMS DIVISION
CARRIER CORPORATION

FIGURE 1.0-5
MILESTONE CHART

MECHANICAL TECHNOLOGY INCORPORATED

1.1 PRELIMINARY SPECIFICATIONS

The initial task was to establish system design objectives and preliminary product specifications. A literature search was made to assess the state of the art of heat engine driven air conditioning systems, as well as obtain comparative design data on conventional vapor compression water chillers and absorption chillers.

A Commercialization Review Board was convened with members from Carrier Air Conditioning Company Marketing, Engineering and Service Departments. This board considered the traditional factors related to bringing this type of product into the market, and helped establish the preliminary product specifications.

1.1.1 STATE OF THE ART SURVEY

A literature review was conducted to:

- A. Develop an understanding of Rankine powered air conditioning systems under development over the past decade.
- B. To characterize the systems, components and working fluids as they may apply to the proposed system.
- C. To develop comparative data on commercially available absorption and conventional water chillers for use in gauging the performance of the proposed system.

1.1.1.1 Summary

As of this writing, commercially available heat operated air conditioning equipment suitable for solar applications is limited to lithium bromide absorption machines ranging in size from three through several hundred tons. The smaller units, i.e., 3 and 25 tons, are recently available from one manufacturer to operate at supply water temperatures of 195°F, which is compatible with current flat plate collector technology. The larger machines, which are available from three manufacturers, were developed for steam operation. When used on hot water, they require supply temperatures of 240°F to produce rated capacity, thus requiring higher performance collectors such as evacuated flat plate or medium concentration types. However, many solar installations with these larger units have been made (M-6) by derating and operating on water at less than 200°F.

As of this writing, there are no heat engine powered air conditioners manufactured and marketed in the United States. One 20 ton Rankine powered heat pump rated for 221°F water is manufactured in Japan. A number of Rankine powered air conditioners are currently under development or installed in demonstration sites, ranging from 3 tons to 100 tons in capacity with supply temperatures from 190 to 300°F.

Several important studies (G-4, W-6, T-4, C-3, C-4, M-6) have been conducted which review the performance potential of Rankine cycles and their economic application in solar air conditioning systems. Conclusions are conflicting, primarily due to the assumptions made by the investigators. Solar system life cycle costing and payback varies by an order of magnitude due to projected solar collector costs, equipment costs, fuel costs, and efficiencies of equipment. The design of the Rankine air conditioner has a major impact on the solar collector cost, which is by far the major cost item in the system. Choice of operating temperature level determines the type of collector and the Rankine cycle efficiency, thus establishing the required collector area. As high performance collector technology has developed recently, the interest in higher temperature, high efficiency Rankine cycles has begun to emerge. The relative OCOP* of absorption and Rankine systems and their dependence upon the source temperature is shown in Figure 1.1-1 (from C-6).

* OCOP (the overall COP of the heat powered air conditioner) is defined herein as the $\frac{\text{cooling capacity}}{\text{heat input}}$ and is distinguished from the COP of the vapor compression cycle, which is defined as $\frac{\text{cooling capacity}}{\text{heat equivalent energy supplied to the vapor compression cycle}}$

FIGURE 1.1-1

COEFFICIENT OF PERFORMANCE COMPARISON FOR
WATER COOLED RANKINE ENGINE/VAPOR COMPRESSION
COOLING UNITS AND ABSORPTION COOLING UNITS
(FROM REF. C-6, PAGE 31, FIGURE 2)

1.1.1.2 Rankine Systems

Thorough reviews of Rankine systems have been published recently (C-3, R-5, M-6, C-6). Rankine power systems under design, demonstration, or in production were cataloged by Curran (C-3) in 1974. Most were designed for electrical power generation or automotive propulsion from relatively high grade thermal energy and outputs range from 0.13 to 200 hp. Expander efficiencies above 10 hp range from 62 to 80%, indicating that a goal of 80% is technically feasible. Overall efficiencies, which are dependent upon the source and sink temperatures, range from 55 to 65% of Carnot.

Since the Curran report, a number of solar air conditioner projects using the Rankine cycle have been initiated under the Department of Energy. Table 1.1-1 lists the major contractors. Up-to-date specifications and design data are not available for several of the projects, and can only be estimated from past work reported in the literature. Following are brief synopses of current and background development work by these contractors and other important contributors in this field.

Airesearch. The Airesearch history of Rankine cycle air conditioning began in 1970 with delivery of small high speed turbo compressor Rankine systems to the U.S. Army (R-5). These were air cooled R-11 cycles designed for waste heat source temperatures of 750°F. This design was later modified for 200°F solar source temperature. Subsequently, Airesearch completed an NSF funded solar powered residential air conditioner design study in 1975 (R-2), which is the basis for the current "404" project to develop 3, 25, and 75 ton models. Table 1.1-2 lists salient features of these designs. The low temperature water cooled design is compatible with flat plate collectors. Major effort has been directed toward high turbine and compressor efficiency, because improved overall COP is recognized as vital to reducing solar system cost. Systems study indicated that regenerators and recuperators were uneconomical at this temperature level. Airesearch has recognized the need to develop process gas lubrication to eliminate the problems of thermal stability, heat transfer, etc., associated with oil lubrication. R-11 was chosen because it has a high Rankine cycle efficiency as well as having above-atmospheric pressures in all but the evaporator.

Carrier Corporation. Carrier built and field tested (5) "Unipak" 40 ton gas-fired Rankine powered water chillers in the mid-1960's. This design (data are given in Table 1.1-2) used water as the working fluid in the Rankine cycle and RC 318 as the working fluid in the vapor compression cycle. The turbine was a 46,000 rpm two-stage axial flow machine directly connected to a single stage centrifugal compressor. The bearing lubricant was water, and water leakage into the RC 318 cycle was tolerated. A separator was provided to return the water to the power cycle. The source temperature was limited to 240°F on the field test units by local codes governing maximum pressure on unattended fired boilers. The design was capable of operation at higher temperatures and resulting higher pressures. Considerable valuable data on system design and control, field service, and component performance were obtained from this program.

TABLE 1.1-1

DOE SOLAR RANKINE COOLING PROJECTS

<u>Contractor</u>	<u>Cooling Unit Tonnages</u>
Airesearch	3, 25, 75
Carrier	25
Energy Technology	3
General Electric	3, 10
Honeywell	2, 25
Los Alamos	77
University of Pennsylvania	11-25
United Technology	3, 18

TABLE 1.1-2
COMPARISON OF SYSTEM CHARACTERISTICS REPORTED IN THE LITERATURE

	<u>Airesearch</u>			<u>Carrier</u>		<u>Thermo Electron</u>	<u>G.E.</u>		<u>Honey- well</u>	<u>LASL</u>	<u>U.P.</u>	<u>U.T.</u>		<u>III</u>
Cooling Capacity (tona)	<u>3</u>	<u>25</u>	<u>75</u>	<u>2.5</u>	<u>40</u>	<u>3</u>	<u>3</u>	<u>10</u>	<u>3</u>	<u>77</u>	<u>25</u>	<u>3</u>	<u>18</u>	<u>20</u>
Heating Capacity (Btu/hr)	80M	800M	2000M	0	1250								500M	228M
OCOP	0.63			0.25	0.47	0.376	0.65	0.8	0.5	0.9	1.2	0.4	0.6	
Solar Supply Water °F	200	200	200	200	Gas		210	300	215	200	300	213	300	221
Boiler Outlet °F	183			180	245	230	200			185	1000	200		
Condenser Type	Evap			Water	Water	Air	Water	Air	Water	Water		Water	Air	Water
Condensing Temp °F	90.1			100	105	120	100		95	75		105		105
Evaporator Type	DX			DX	Chill	DX	DX	DX	DX	Chill		Chill		
Evaporating Temp °F	45.6			45	37	45	45		45			45		
Expander Type	Tur	Tur	Tur	Rotary	Tur	Recip	Vane	2-Stage Vane	Tur	Tur	Tur	Tur	Tur	Screw
Expander Eff'y	0.72	0.85	0.85	0.5	0.7	0.72	0.75	0.35	0.72		0.7	0.63		
Working Fluid	R-11			R-114	Water	R-114	R-11	FC-88	R-113	R-11	Water	R-11	R-113	R-11
Lubrication	Gas	--	--	Oil	Water	Oil	Oil		Oil	Oil				Oil
RPM (1000's)	61	41	25	1.0	45	1.8	600			11	34	25		
Compressor Type	Cent	Cent	Cent	Recip	Cent	Recip	Recip		Recip	Cent		Cent	Cent	Screw
Compressor Eff'y	0.71	0.82	0.82	0.65	0.71	0.72	0.76		0.85			0.63		
Working Fluid	R-11			R-12	RC 318	R-22	R-22	R-22	R-12	R-11		R-11	R-113	R-22
Lubrication	Gas	--	--	Oil	Water	Oil	Oil		Oil	Oil				Oil
RPM (1000's)	82	41	25	1.6	45	1.8			0.6	11		25		
Feed Pump Eff'y	0.5			0.3	0.1	0.8	0.7		0.17					
Auxiliary Energy	Motor			Motor	Fossil	Motor	Motor		Motor	Fossil	Fossil		Fossil	Motor

In 1976, Carrier built and installed a 2.5 ton Rankine driven air conditioner in its solar demonstration house at Naples, Florida. Source temperature in this system was limited to 212°F by the use of water in the atmospheric vented solar system. The expander was a rotary, positive displacement, gerotor engine obtained from Sun Power (Kinetics in Reference C-3). The working fluid, R-114, was primarily chosen to match the displacement of the expander to the desired speed and power output. The expander drives an open reciprocating compressor through a 1.6:1 speed increasing belt drive. The vapor compression cycle uses R-12, again chosen to match available equipment to the desired speed and capacity. Both cycles are water cooled. The evaporator is a remote, direct expansion coil in the air stream. Figure 1.1-2 is a schematic of the system, showing the electric motor auxiliary drive for the compressor. Performance and cycle data are given in Table 1.1-2. Several factors contributed to the relatively low overall COP. The poor efficiency product of the compressor and expander (0.325) resulted from the use, without modification, of off-the-shelf components. The OCOP reflects the net Rankine cycle work, because the feed pump is belt driven by the expander. The belt drive system is inefficient, and the windmilling motor absorbs significant power. The operating speed varied directly with supply water temperature. Thus, the cooling capacity varied directly with water temperature. A low water temperature limit was used to cut in the motor and turn off the expander vapor. The feed pump speed, which in turn varied with the expander speed, was not adequate under low speed conditions. Because shaft speed was fixed at the design point under motor operation, motor assist was used to speed up the system and the pump whenever the refrigerant level in the boiler was too low. This strategy was also employed during each start-up, to assure a boiler level.

This system has performed well through one complete season (1977). The major problem encountered was leakage from shaft seals. The seal on the compressor has been replaced.

The experiences obtained on this project have been valuable in setting design goals for the proposed next generation of Rankine powered air conditioners. These include:

- High efficiency compressor and turbine.
- Hermetic design
- Elimination of oil lubrication
- Air cooling
- Direct drive for minimum friction loss.
- High efficiency pump with independent drive
- Higher source temperatures for improved Rankine cycle efficiency.

General Electric. A multi-vane expander for a low temperature Rankine powered vapor compression air conditioner has been under development at the General Electric Space Division for about 7 years. A three-ton water cooled R-11 unit has been on test for three years (E-1, C-6).

FIGURE 1.1 - 2
SCHEMATIC OF NAPLES RANKINE UNIT

Expander efficiencies as high as 0.75 have been measured, but with further development, G.E. anticipates that an efficiency of 0.85 can be achieved. A conventional open reciprocating compressor is belt driven from the expander, and a motor can be "clutched" in to drive the compressor on auxiliary power. Data for this design are reported in Table 1.1-2. G.E. has a "404" contract to develop 3 and 10 ton versions of this original design. The new air cooled cycles will be designed for 300°F supply water to reach higher OCOP levels in order to achieve cost effectivity. The expander will be two stage and drive a hermetic motor reciprocating compressor unit at 1200-2200 rpm via a magnetic coupling. Available data for the 10 ton model are given in Table 1.1-2.

Los Alamos Scientific Laboratory. A 77-ton Rankine powered chiller was designed and built for the LASL National Security and Resources Center by Barber Nichols (B-4). A York 100 ton centrifugal chiller was modified by the addition of a turbine and Rankine cycle heat exchangers to create a packaged machine. The turbine drives the compressor directly at 12,000 rpm. Data are given in Table 1.1-2. Problems reported (Barber, C-6) include oil migration to the chiller. Barber recommends a turbo-gearbox drive for a conventional open centrifugal chiller to create two separate loops which will also facilitate electric auxiliary input and ease of retrofit. Data taken at LASL during 1977 (Balcomb, C-6) show that the operation of the Rankine cycle has compared favorably with a parallel absorption system. The Rankine system only takes one minute to reach steady state operation compared to 30 minutes for the absorption unit.

Honeywell. A three-ton water cooled Rankine air conditioner was built for the Honeywell Mobile Solar Laboratories in 1974 by Barber-Nichols (B-17, C-6). Data on this unit are given in Table 1.1-2. A high speed R-113 turbine drives a 600 rpm reciprocating compressor through a gear box. This system was designed to operate from 215°F water from flat plate collectors. A motor can be energized to drive the compressor through a clutch for auxiliary power. It can also serve as a Rankine powered generator. The system is water cooled.

Barber observed substantial losses in the gearbox and belt drive system, which could be reduced in future designs. He expects the turbine efficiency to reach 0.80. The next model will be evaporatively cooled.

Honeywell is also developing a 25 ton water cooled machine, with improved turbine, compressor, drive, and feed pump. There are no published data on this design.

Battelle. Battelle conducted a feasibility study under NSF, for an advanced concept Rankine heat pump using a rotary vane compressor and expander (F-2, F-1). The unique feature of this machine was the use of pivoting tips to reduce friction. Experimental studies on pivoting tips with process gas lubrication demonstrated extremely low friction characteristics. The advantages of this design would be: very high expander and compressor efficiencies at low rotational speeds (they predict efficiencies as high as 0.85 for each); ease of matching the turbine and compressor; compatible with motor speeds; elimination of oil type lubricants. It is claimed that capacities up to 100 tons are practical.

Thermo-Electron Corporation. Thermo-Electron Corporation has been engaged in the development of Rankine power and air conditioning systems using reciprocating engines since the mid-1960's (S-3). Recently, tests (D-2) on a 5 hp reciprocating single cylinder engine (designed for CP-34) with R-114 and R-22, indicated that expander efficiencies of 0.80 were practical. Performance peaked at 1800-2000 rpm. A conceptual design for a solar powered unit has been developed (S-5). Data are reported in Table 1.1-2. This design has been optimized for flat plate collector temperature levels of 180-230°F. The major contribution of their work has been in the development of reciprocating expanders utilizing present day reciprocating engine and compressor technology.

Ishikawajima-Harima Heavy Industries Co., Ltd., Japan. IHI has developed and is marketing a line of Rankine cycle waste heat power units and solar air conditioning units. They use R-11 and are designed for moderate source temperature levels 212°F-221°F. Data are shown on Table 1.1-2 for their solar Rankine powered heat pump Model SR/HP-20. This machine is sold with a built-in cooling tower. It utilizes revolving volume type (screw) expander and compressor. These open machines are directly coupled through an auxiliary electric motor. The Rankine cycle working fluid is R-11, and the vapor compression cycle uses R-22.

University of Pennsylvania. The University of Pennsylvania has undertaken a DOE contract to develop a superheated steam Rankine cycle which will mechanically drive a conventional vapor compression air conditioner. Detailed data on the latest conceptual design are not available; earlier versions are described in the literature (L-2, C-6), and data are shown in Table 1.1-2. Curran's evaluation (C-4) of this concept showed it to be superior in payback to other versions of the Rankine cycle now under development. Figure 1.1-1 shows the expected performance level in relation to various other absorption and Rankine cycles. The concept is to boil at relatively low temperature using solar energy and then superheat the vapor entering the turbine, up to 1000°F, by adding auxiliary energy. Up to 75% of the cycle input would be supplied by solar at a low temperature level. Water has a high specific energy, so superheating between prespecified pressure limits can increase the potential enthalpy drop. Many of the high molecular weight fluids currently under investigation do not increase the potential enthalpy drop with superheat. Water also has the advantages at these temperatures of stability, non-toxicity, non-flammability, high thermal conductivity, and low cost. Another feature of this cycle is that feed pump work is very small. Pump work for a 30 ton unit is estimated at 0.1 hp.

United Technology. United Technology background extends from the decade of the 1960's when gas fired Rankine cycle air conditioners were under development (B-8). For a Rankine source temperature between 300 and 400°F, R-113 was selected over R-11, R-114, and R-21 for stability reasons. Study results from a refrigerant supplier showed that at 370°F maximum vapor temperature, decomposition was less than 1% in an oil-free system. Most recent test experience (B-10, B-11, B-12) has been on an R-11 two-stage centrifugal machine (Table 1.1-2) which reached an OCOP of 0.40 at 180°F boiler temperature and 90°F condensing temperature. The compressor efficiency was 0.60-0.67. The anticipated level of compressor/turbine efficiency product is 0.64 for small systems and 0.72 for larger systems.

Data on current development of an 18 ton high temperature system are not available. However, Biancardi has indicated (C-6) that the temperature level will be 300°F as supplied by a medium concentration collector array. Auxiliary energy will be provided by fossil fuel. OCOP is predicted in the 0.6 to 0.8 range. A heat pump mode will be included.

1.1.1.3 Choice of Working Fluids

Miller (M-3) has studied (11) Rankine cycle working fluids for solar to electrical energy conversion. While the temperature level of his study (350-400°F) was above the proposed application (250-300°F), a number of important factors are related. One of the candidates evaluated as a benchmark was R-113. Miller states that the maximum use temperature of R-113 is 300°F. Miller listed a number of important trade-offs to be evaluated when selecting a working fluid:

1. Cycle pressures
2. Circulation rate
3. Expander enthalpy drop
4. Nozzle spouting mach number
5. Regenerator size
6. Fluid cost
7. Thermal stability
8. Availability
9. Flammability
10. Toxicity
11. Freezing point

Allen and Stiel (A-1, A-2, A-3) have evaluated 35 working fluids for use in Rankine cycle solar heat pumps at temperature levels in the range of the proposed design.

Rankine cycle computations were made by computer for a single set of conditions (Table 1.1-3 from Reference A-2). Power cycle efficiency, vapor compression cycle COP and overall COP are reported, as well as a number of cycle state point pressures and fluid properties. Fluid properties were calculated from well known equations listed in the reference. Formulas used are:

$$\eta = \frac{W_{\text{exp}} - W_p}{Q_B}$$

$$\eta = \text{Rankine cycle efficiency}$$

$$W_{\text{exp}} = \text{expansion work}$$

$$W_p = \text{feed pump work}$$

$$Q_B = \text{boiler input}$$

TABLE 1.1-3

CYCLE EFFICIENCIES - COOLING MODE

$T_U = 180^\circ\text{F}$, $\text{Regen} = 0.5$, $T_{PC} = 125^\circ\text{F}$, $E_{EXP} = 0.8$, $E_P = 0.7$

$T_E = 40^\circ\text{F}$, $T_{SH} = 65^\circ\text{F}$, $T_{CC} = 125^\circ\text{F}$, $T_{SC} = 120^\circ\text{F}$, $E_{COMP} = 0.8$

Substance	T_B °F	C_P 300°K k-cal/mole°K	η	COP	OCOP	P_E psia	P_{CC} psia	P_B psia	V_{COMP} cfm/ton	M_{COMP} lb/min/ton	M_P lb/min/
$T_B = -50^\circ - 0^\circ\text{F}$											
R-22	-41.7	13.70	.044	3.72	.163	82.1	284.3	459.4	2.2	3.0	17.7
R-500	-28.3	17.44	.050	3.76	.189	60.8	215.2	379.3	2.9	3.4	16.0
R-12/31	-21.3	15.33	.051	3.86	.195	52.0	119.2	320.1	3.2	3.3	15.3
R-12	-19.0	17.36	.052	3.81	.197	49.1	175.6	311.1	3.5	4.0	18.3
R-134a	-15.9	20.47	.051	3.73	.191	49.0	187.4	348.5	3.5	3.4	15.7
R-152a	-12.7	17.57	.052	3.84	.201	45.8	177.8	324.0	3.5	2.0	9.1
R-134	- 3.7	22.54	.053	3.73	.199	39.4	166.7	330.8	4.0	3.1	13.4
$T_B = 0^\circ - 50^\circ\text{F}$											
R-31/114	9.6	15.25	.052	3.96	.207	28.3	119.2	214.9	5.0	2.5	10.7
iso-butane	10.7	23.24	.060	3.84	.229	26.6	102.8	201.9	6.1	1.8	6.3
R-124a	13.4	25.06	.059	3.80	.224	25.8	106.8	216.3	6.1	4.0	14.7
R-142b	13.4	20.26	.060	3.92	.235	24.8	103.7	210.7	6.0	2.8	10.0
1-Butene	20.4	21.14	.061	3.92	.238	22.5	91.9	184.5	6.7	3.9	5.6
R-12B1	24.6	17.89	.059	3.98	.234	20.1	82.3	159.4	7.2	4.5	16.7
R-114/133a	26.7	25.02	.061	3.87	.234	19.3	81.4	165.5	7.8	4.2	14.5
R-114	38.6	27.42	.061	3.85	.233	15.0	67.4	140.5	9.7	4.6	15.7
R-12/133a	41.2	19.55	.062	4.00	.246	14.2	70.3	152.3	8.9	2.6	9.2
R-133a	42.7	21.56	.062	3.97	.244	13.7	67.7	142.2	9.4	2.8	9.8
$T_B = 50^\circ - 100^\circ\text{F}$											
R-11	75.1	18.81	.061	4.08	0.25	6.85	35.6	76.7	17.4	3.0	10.2
R-12B2	75.9	21.84	.060	4.08	0.244	6.8	34.6	72.5	17.7	4.6	16.4
R-113/152	76.9	20.89	.063	4.06	0.256	6.1	30.9	87.3	17.95	2.1	6.9
R-142	94.9	19.83	.061	4.10	0.251	4.0	26.1	61.7	25.8	1.9	6.4
$T_B = 100^\circ\text{F}$											
R-132b	116.0	23.58	.064	4.09	.260	2.4	17.4	44.6	41.3	2.4	7.8
R-113	117.4	30.44	.063	4.02	.254	2.6	16.8	40.8	42.2	3.7	11.6
CCl ₄	169.5	19.97	.058	4.17	.242	0.8	6.4	15.6	112.4	2.5	8.9
Toluene	230.8	24.96	.065	4.19	.272	0.15	1.8	5.95	507.1	1.2	3.8
Chloro-benzene	268.8	23.20	.059	4.20	.248	0.1	0.8	2.6	1193.3	1.4	4.8

$$\text{COP} = Q_E / W_{\text{comp}}$$

Q_E = evaporator input (cooling capacity)

W_{comp} = compressor work

$$\text{OCOP} = Q_E / Q_B = \eta * \text{COP}.$$

This data show that the power and vapor compression efficiencies increase with boiling point. Low pressure refrigerants, such as R-11 and R-113, rank high with respect to OCOP.

A comparison of dual fluid system OCOP is given in Table 1.1-4 (from Reference A-2). This combines the higher pressure fluids more commonly used in vapor compression cycles with the best low pressure Rankine cycle fluids. Overall COP for most combinations falls short of R-113 and R-11 single fluid systems.

An evaluation of selected fluids over a range of operating parameters was also made. Figure 1.1-3 (from Reference A-2) shows the relative performance of seven fluids with source temperatures ranging from 180 to 300°F. R-11 and R-113 have identical performance (higher pump work, which is deducted from power cycle output, offsets the higher OCOP of R-11). On this basis, R-11 or R-113 is more than 27% better than R-114 at 300°F. An evaluation of regenerator effectiveness was made for R-113. As shown on Figure 1.1-4 (from Reference A-2), the regenerator is much more effective for high temperature cycles, such as the proposed system, than for low temperature cycles.

Atwood (A-4) has studied a number of refrigerants for use in centrifugal water chillers. He points out that R-11 is the predominant refrigerant in service today. While higher pressure refrigerants could reduce maintenance problems associated with subatmospheric systems, they may have the disadvantage of lower COP and the liability of costly "pressure code" shells. Table 1.1-5 (from Reference A-4) compares performance factors of a number of refrigerants in a typical centrifugal vapor compression cycle. This shows R-11 to be the best of the well known refrigerants (under the conditions noted) in terms of gas horsepower/ton. Two newer refrigerants, R-21 and R-31/114, are reported to be nearly equal in horsepower/ton, but require higher rotational speeds. The comparison of R-113 with R-11 is clearly shown. R-113 requires a larger, lower speed impeller for a given capacity. The high rotational mach number of R-113 penalizes the stage efficiency of the impeller, resulting in a higher horsepower/ton.

One key item discussed by Atwood is the "submergence" effect resulting from the relatively large effect of a static liquid head on the refrigerant evaporating process. This effect is more pronounced on the low pressure refrigerants. Figure 1.1-5 (from Reference A-4) shows the ΔT penalty of a 2 ft head on several refrigerants. R-113 is most severely affected, and would require a special evaporator design to minimize the penalty.

TABLE 1.1-4

OVERALL COEFFICIENTS OF PERFORMANCE
FOR TWO FLUID SYSTEMS - COOLING MODE CONDITIONS

<u>POWER FLUID</u>	<u>Refrigerant</u>			
	<u>R-12</u>	<u>R-22</u>	<u>R-12/31</u>	<u>R-31/114</u>
R-11	.232	.227	.235	.242
R-113	.240	.234	.243	.249
R-133a	.236	.231	.239	.246
R-132b	.244	.238	.247	.253
Toluene	.248	.242	.251	.257
R-114/133a	.232	.227	.235	.242

VARIATION OF OCOP WITH T_U

FIGURE 1.1-3

VARIATION OF OCOP WITH REGEN FOR R-113

FIGURE 1.1-4

TABLE 1.1-5

PERFORMANCE OF CENTRIFUGAL REFRIGERANTS
BASED ON A CYCLE AT 35° EVAPORATION/105° CONDENSATION
NO SUPERHEAT/5° LIQUID SUBCOOLING

Refrigerant	R-113 ⁺	R-11	R-21 ⁺	R-133a ⁺	R-114 ⁺	R-31/114 ⁺	R-12	R-12/31 ⁺	R-500	R-115 ⁺	R-22
Condenser Pressure (psia)	11.58	25.60	43.58	49.05	49.70	88.11	141.3	144.9	167.1	193.4	225.3
Evaporator Pressure (psia)	2.33	6.26	11.08	12.27	13.55	25.47	47.26	48.14	55.3	67.57	76.2
Pressure Ratio	4.96	4.09	3.94	4.00	3.67	3.46	2.99	3.01	3.01	2.89	2.96
Net Refrigeration Effect (BTU./lb.)	56.28	67.49	87.80	72.12	45.92	82.91	49.83	56.94	59.99	30.43	68.45
Isentropic Compression Work (BTU./lb.)	8.93	10.55	13.82	11.62	7.72	13.18	8.33	9.37	10.10	5.68	11.61
Equivalent Head (ft.-lbs./lb.)	6,950	8,210	10,750	9,040	6,010	10,250	6,480	7,280	7,880	4,420	9,950
Ideal Coefficient of Performance	6.31	6.40	6.35	6.21	5.94	6.29	5.98	6.08	5.93	5.36	5.88
Mass Flow Rate (lb./min.-ton)	3.55	2.96	2.28	2.77	4.37	2.41	4.01	3.51	3.24	6.56	2.93
Specific Volume of Suction Vapor (cu.ft./lb.)	12.31	6.037	4.514	3.511	2.292	2.110	0.8424	0.915	0.863	0.446	0.717
Suction Flow Rate; (CFM/ton)	43.7	17.92	10.28	9.74	10.00	5.09	3.38	3.21	2.79	2.93	2.095
Tip Speed (U ₂) (from assumed head coefficient); (ft./sec.)	615	668	765	702	573	747	594	630	655	483	702
Sonic Velocity at Inlet (a ₁); (ft./sec.)	374	444	516	466	386	531	446	474	491	377	541
Rotational Mach Number (Mo)	1.65	1.505	1.48	1.506	1.49	1.41	1.33	1.33	1.33	1.28	1.30
Relative Efficiency Due to Mach Number	0.900	0.929	0.932	0.929	0.933	0.947	0.959	0.959	0.959	0.967	0.964
Stage Efficiency (Assuming 80% Base)	0.720	0.742	0.745	0.742	0.746	0.757	0.767	0.767	0.767	0.775	0.770
Resultant Gas H. P. Required/Ton	1.038	0.993	0.997	1.020	1.062	0.990	1.03	1.013	1.035	1.13	1.040
RPM for 20" Impeller	7,050	7,650	8,750	8,040	6,550	8,550	6,800	7,220	7,500	5,530	8,050
Capacity Factor (Q/ND ³)	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18
Tons for 20" Impeller	134	355	710	690	547	1,400	1,670	1,870	2,240	1,570	3,200
1 Tonnage Compared to R-11	38%	100%	200%	194%	154%	395%	473%	527%	630%	443%	900%
1 Tonnage Compared to R-12	8.0%	21.2%	42.5%	41.3%	32.8%	83.9%	100%	112%	134%	94%	192%
Discharge Temperature	105°	121°	144°	112°	105°	130°	114°	120°	118°	105°	137°
Liquid Flow Rate (cu.in./min.-ton)	64.0	56.7	47.3	59.7	85.5	54.0	88.0	78.5	83.1	150.0	71.0

*Refrigerants for which new data is presented.

+Cycles based on enough suction superheat to avoid "wet" compression, full credit taken for cooling effect from suction vapor

FIGURE 1.1-5

EFFECT OF REFRIGERANT LIQUID DEPTH ON
THE EVAPORATING TEMPERATURE

REFRIGERANT	113	11	21	133a	114	31/114	12
TEMPERATURE AT SURFACE (°F)	35°	30°	30°	30°	30°	30°	30°
PRESSURE AT SURFACE (PSIA)	2.33	6.26	11.08	12.27	13.55	25.47	47.25
PRESSURE EFFECT OF 2 FEET OF LIQUID (PSIA)	1.40	1.33	1.23	1.20	1.32	1.18	1.21
TOTAL PRESSURE AT BOTTOM OF SHELL (PSIA)	3.73	7.59	12.31	13.47	14.87	26.65	48.47
EVAPORATING TEMPERATURE @ BOTTOM PRESSURE (°F)	53.3°	43.5°	39.7°	39.0°	39.3°	37.2°	36.4°
Δ T PENALTY AT BOTTOM (°F)	18.3°	8.5°	4.7°	4.0°	4.3°	2.7°	1.4°

1.1.1.4 Working Fluid Stability

Sealed tube stability tests on refrigerants were reported by Parmelee (P-1). Tests were run on 12 commonly used refrigerants in combination with or without various metals and oils. Test temperatures ranged from 250 to 700°F for periods of a few days up to two years. This report is the most comprehensive in the literature, and it is still the main authority in the field. In the absence of oil, Parmelee ranks refrigerants in the order of increasing stability as follows: R-11 < R-113 < R-13B1 < R-114 = R-12 < R-22 < R-115 < RC-318. When napthenic oil is present, the order is the same except R-114 < R-12. Stability of R-11 and R-113 in the presence of oil and metals is given in Table 1.1-6 (from P-1).

TABLE 1.1-6

STABILITY OF R-11 AND R-113 WITH OIL AND METALS
(0.21 moles refrigerant and equal volume oil and metals)

	Temp., °F	Time Days	Al	Cu	Decomposition *		Al + Cu + Fe
					Fe	S.S.	
R-11	250	52	0.13	0.36	16.0	2.8	83
R-11	250	1160	100	--	--	--	--
R-11	300	52	0.33	0.77	Burst	Burst	Burst
R-113	250	660	0.36	0.76	4.4	2.1	3.7
R-113	250	1160	0.75	--	--	--	--
R-113	300	52	0.20	0.42	23.0	12.0	11.0

* Decomposition measured by chloride ion analysis.

By comparison, in oil free tests for two years, the stability is much improved as shown in Table 1.1-7.

R-11 stability is substantially less than the other refrigerants for all metals. R-11 is not recommended for the proposed application (B-18). R-113 stability is low with mixed metals Al + Cu + Fe. This is believed to be due to the copper-iron combination. This trend is also evident on refrigerants R-114 and R-12, suggesting that the boiler, preheater, and other high temperature refrigerant-containing parts should be made from a single metal.

Other high temperature stability tests were reported by Downing of DuPont (D-4). Sealed tube test results were compiled and tabulated for R-11, R-12, R-113, and R-114 from 250 to 700°F. Decomposition rates per year for steel have been plotted as a function of temperature for these refrigerants, Figure 1.1-6. The strong temperature dependence is apparent. Decomposition of R-11 is an order of magnitude higher than R-113, R-12, and R-114 at temperatures below 300°F.

TABLE 1.1-7

TWO YEAR STABILITY OF REFRIGERANTS WITH METALS, OIL ABSENT

	<u>Temp.</u> <u>°F</u>	<u>Decomposition* %</u>				
		<u>Al</u>	<u>Cu</u>	<u>Fe</u>	<u>S.S.</u>	<u>Al + Cu + Fe</u>
RC-318	400	< 0.001	< 0.001	< 0.001	< 0.001	< 0.001
R-115		0.005	0.010	0.025	0.040	Burst
R-22		0.037	0.16	0.29	0.17	1.5
R-114		0.030	0.54	1.00	0.27	6.2
R-12		**	0.074	0.61	Burst	Burst
RC-318	300	< 0.001	< 0.001	< 0.001	< 0.001	< 0.001
R-115			0.002	< 0.001	0.001	0.001
R-22			0.001	< 0.001	0.006	0.010
R-114			0.18	0.013	0.022	1.1
R-12			0.13	0.013	0.043	0.16
R-113		0.008	0.13	0.042	0.028	23
R-11		0.022	60.0	0.72	5.3	58

* Decomposition measured by chloride or fluoride analysis.

** Unaccountably high.

FIGURE I.1 - 6

DECOMPOSITION OF REFRIGERANTS WITH STEEL
IN SEALED TUBE TESTS. NO OIL PRESENT
(FROM REF. D-4)

Eismann (E-3) describes reactions between chlorofluorohydrocarbons and metals. Stability of these compounds is related directly to the fluorine content. "Copper plating" is a well known reaction which occurs in sealed refrigerant systems with oil present at high temperatures. It is not believed to occur without oil (B-18). Hydrolysis takes place in the presence of moisture, though very slowly. Reaction of R-113 with zinc is well known and is used in manufacturing chlorofluoroethylene, a monomer for certain resins. Reactions with metals can be accelerated by raising the temperature and pressure. It is known that chlorinated compounds containing no fluorine are reactive at much lower temperatures than fluorinated compounds. In general, metals which form thermally stable chlorides and fluorides will be capable of reacting with chlorofluorohydrocarbons at elevated temperatures to form metal chlorides. The common metals of construction, copper, iron, and aluminum, fall into this category.

Eismann describes, with supporting experimental data, a destructive reaction between aluminum and R-12 in which the aluminum impellers of high speed compressors have been destroyed. This is now attributed to a failure mode in which the rotating impeller rubs on the housing creating very high temperatures. Only refrigerants with higher heats of reaction and higher vapor pressures are prone to this reaction. The order of rank of the reaction of several refrigerants with aluminum, with the least vigorous first, is as follows:

Decomposition products are described for certain refrigerants (B-14, B-18). The first step in the decomposition is a reversible formation of Cl_2 and a perhalogenated ethane. This may be followed by a so-called "disproportionation reaction" to other refrigerants; for instance, $R-12 \rightarrow R-13 + R-11$, or $R-11 \rightarrow R-12 + CCl_4$. Decomposition products expected in an oil free system would include chlorides of metals, polymerized "teflon like" solids, small amounts of other "refrigerants" in the liquid and gas phase, and possibly carbonaceous solids if temperatures are high enough.

1.1.1.5 Commercially Available Equipment

Heat Operated

The only heat operated air conditioning equipment in the 25 ton range which is commercially available in the U.S. is the Arkla Model WFB-300 lithium bromide absorption machine. It is a water cooled chiller designed for 195°F solar water. Catalog performance data, shown in Figure 1.1-7 at the rated condenser water temperature condition of 85°F, demonstrates the drop-off in cooling capacity with decreasing water temperatures. The coefficient of performance reported excludes the electrical auxiliaries. The auxiliaries required for operation comparable to an air cooled unit are a solution pump at 0.15 kW, a cooling tower pump at approximately 0.6 kW, and a cooling tower fan at 2.0 kW. Pertinent product specifications are given in Table 1.1-8. Tabular performance data over a range of outdoor wet bulb temperatures, entering hot water temperatures, and entering chilled water temperatures are given in Table 1.1-9.

FIGURE 1.1 - 7

ARKLA WFB 300 PERFORMANCE DATA

TABLE 1.1-8

SPECIFICATION DATA FOR ARKLA WFB-300 WATER COOLED CHILLER

Performance Data

Capacity	306,000 Btu/hr
COP (at rated conditions)	0.685

Input Data

Hot Water Temperature	195°F
Hot Water Input	447,000 Btu/hr
Hot Water Flow	90 gpm
Generator Pressure Drop	20.7 ft of water
Voltage	115/60/1
Watts	150

Chiller Data

Outlet Temperature	45
Chiller Water Flow	60 gpm
Chiller Pressure Drop	9.8 ft of water
Water Volume	12 gallons
Fouling Factor	0.0005

Condenser Data

Heat Rejection Design	753,000 Btu/hr
Max	853,000 Btu/hr
Condenser Inlet Temperature	85°F
Condenser Water Flow	90 gpm
Condenser Pressure Drop	22.9 ft of water

Physical Data

Operating Weight	3420 lbs
Overall Dimensions L X W X H	118 x 38 x 72-1/2 in.

TABLE 1.1-9
TABULAR PERFORMANCE DATA FOR ARKLA WFB 300 ABSORPTION CHILLER

Ent. Chilled Water		50					55					60				
OUTDOOR WB	H ₂ O ENT COND	TON	OCOP	WM	WP	WFP	TON	OCOP	WM	WP	WFP	TON	OCOP	WM	WP	WFP
II a		TSOL = 195														
35	80	25	.6	0	.15	2.6	28.0	.625	0	.15	2.6	30.0	.72	0	.15	2.6
⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈
70	80	25.0	.6	0	.15	⋈	28.0	.625	0	.15	⋈	30.0	.72	0	.15	⋈
75	80	25.0	.6	0	.15	⋈	25.4	.625	0	.15	⋈	27.2	.74	0	.15	⋈
80	90	17.0	.6	0	.15	2.6	21.2	.625	0	.15	2.6	24.0	.75	0	.15	2.6
	90	0														
II b		TSOL = 180														
35	80	20.5	.70	0	.15	2.6	22.0	.73	0	.15	2.6	22.9	.72	0	.15	2.6
⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈
70	80	20.5	.70	⋈	⋈	⋈	22.0	.73	⋈	⋈	⋈	22.0	.72	⋈	⋈	⋈
75	85	17.8	.696	⋈	⋈	⋈	19.6	.725	⋈	⋈	⋈	20.9	.712	⋈	⋈	⋈
80	90	13.8	.712	0	.15	2.6	16.0	.74	0	.15	2.6	17.6	.73	0	.15	2.6
	90	0														
IIc		TSOL = 165														
35	80	12.6	.67	0	.15	2.6	13.3	.642	0	.15	2.6	13.7	.58	0	.15	2.6
⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈	⋈
70	80	12.6	.67	⋈	⋈	⋈	13.3	.642	⋈	⋈	⋈	13.7	.58	⋈	⋈	⋈
75	85	10.8	.655	⋈	⋈	⋈	12.0	.635	⋈	⋈	⋈	12.8	.58	⋈	⋈	⋈
80	90	7.6	.65	0	.15	2.6	10.1	.638	0	.15	2.6	11.3	.58	0	.15	2.6

* Cooling Tower Fan = 2 hp = 2 KW
Cooling Tower Pump = .75 hp = .6 KW

WB = Wet Bulb Temperature °F
WP = Solution Pump KW
WFP = Cooling Tower Fan and Pump KW

The list price of this unit to a contractor or dealer is reported to be \$16,675. Assuming the contractor operates on a margin of 30%, the price to a customer would be approximately \$23,821.

Conventional

The conventional type of equipment which most closely parallels the proposed design is a packaged, air cooled, water chiller such as the Carrier Model 30GA. This model is available from 20 to 100 tons. Physical specifications for a 25 and 50 ton unit are given in Appendix A1.1, and tabular performance data at various outdoor air and entering chilled water temperatures are given in Table 1.1-10 for the 25 ton unit. The physical size data on a 50 ton unit is of significance because the total heat rejection of the 25 ton dual loop Rankine cycle is approximately the same as a 50 ton vapor compression unit. The physical size of these machines is dictated primarily by the condenser coil and air handling system.

The conventional type of equipment which most closely parallels the Arkla water cooled water chiller is a packaged water cooled water chiller such as the Carrier Model 30HH, which is available from 7-1/2 to 100 tons. Tabular performance data at various outdoor wet bulb temperatures and entering water temperatures are given on Table 1.1-11.

1.1.1.6 Solar Collector Performance

The importance of the interrelationship between the collector and the solar powered cooling machine has been stressed in several studies (G-4, T-4, W-6, A-5). The high cost of collector surface, relative to the cooling machine, implies that trade-offs favoring machine efficiency will be cost effective, and that optimization of collector/machine against operating temperature level is very important. Therefore, some background in collectors has been referenced.

There are two categories of collectors: flat plate, and concentrating. Evacuated tubes can be used to improve both types. Collectors can be tracking or nontracking, but only concentrating collectors benefit economically from tracking.

The selection of collector type depends upon the temperature level desired. Conventional flat plate collectors are limited to around 250°F (T-4) by efficiency considerations. Non-concentrating evacuated tube collectors may be usable to 400°F. For temperatures beyond, it is necessary to employ concentrating collectors.

State-of-the-art flat plate collectors employ selective surfaces, such as black chrome, and antireflective coatings on high transparency glazing to increase performance.

Evacuated non-concentrating collectors also employ selective surfaces and antireflective coatings as well as shaped reflecting back surfaces to maximize performance.

TABLE 1.1-10

TABULAR PERFORMANCE DATA ON MODEL 30-GA-030
AIR-COOLED ELECTRIC VAPOR COMPRESSION WATER CHILLER

ECWT	50			55			60		
TAIR	TONS	WM	WF	TONS	WM	WF	TONS	WM	WF
-20	26.2	27.5	0*	28.6	28.8	0	31	30.1	0
5	}	}		}	}		}	}	
25	26.2	27.5	1.63	28.6	28.8	1.63	31	30.1	1.63
35	}	}	1.63	}	}	1.63	}	}	1.63
45	}	}	1.63	}	}	1.63	}	}	1.63
55	}	}	2.98	}	}	2.98	}	}	2.98
65	}	}	2.98	}	}	2.98	}	}	2.98
75	}	}	2.98	}	}	2.98	31	}	2.98
85	26.2	27.5	4.33	28.6	28.8	4.33	31	30.1	4.33
95	24.4	28.9		26.7	30.5		29	32.1	
105	22.7	30.5	}	24.9	32.2	}	27.1	33.9	}
115	21	32.0	}	23.1	33.9	}	25.2	35.7	}
115	0								

* WITH MOTOR MASTER CONTROL ON 3RD FAN

ECWT = Entering Chilled Water Temp. F

WM = Compressor KW

WF = Condenser Fan KW

TABLE 1.1-11

TABULAR PERFORMANCE DATA ON MODEL 30EH 030
WATER-COOLED ELECTRIC VAPOR COMPRESSION

WATER CHILLER

30 HH 030		<u>WATER CHILLER</u>									
E CWT		50				55				60	
OUTDOOR	H ₂ O										
WET BLUB	ENT										
COND.	COND.	TONS	WM	*	TONS	WM	WW	TONS	WM	WW	
	75	26.8	24.1	2.6	29.4	25.0	2.6	32.1	25.8	2.6	
68	75	26.8	24.1	2.6	29.4	25.0	2.6	32.1	25.8		
73	80	26.2	25.0		28.7	26.0		31.2	26.9		
78	85	25.6	26.0		28.1	27.0		30.5	28		
83	90	24.8	27.1	2.6	27.1	28.1	2.6	29.4	29.2	2.6	

* COOLING TOWER FAN & PUMP

GPM $\approx$ 85, hp = .75 ; .6 KW

CTFAM hp = 20 ; 2.0 KW

~ SIZED FOR ARKLA ABSORPTION, ASSUME
7° APPROACH ON COND. WATER INSTEAD
OF 10° APPROACH.

ECWT = Entering Chilled Water Temp. °F

WM = Compressor KW

WFP = Cooling Tower Fan and Pump KW

Concentrating collectors may employ selective surfaces, evacuated tubes or other schemes such as cavities to minimize losses. Compound parabolic collectors, such as the Winston (W-7) design with low concentration ratios (i.e., 3-5) provide an angular field of view wide enough to eliminate tracking other than on a seasonal basis. Fresnel lense concentrating collectors which are manufactured with concentration ratios of 6-12 must use diurnal tracking. Larger concentration ratios also require tracking.

Data on performance of state-of-the-art collectors available in the market place are given in Table 1.1-12 and Figure 1.1-8. There is a difficulty in presenting concentrating and non-concentrating collectors on the same figure, resulting from the treatment of diffuse radiation. The non-concentrating collector accepts total radiation, while the concentrating type only accepts a fraction of the diffuse radiation, depending upon the type and the concentration ratio. Thus, the curves presented which are based on total radiation assume clear day radiation with very little diffuse. If diffuse radiation was, for example, 20%, then the concentrating collectors would intercept the efficiency curve at a point approaching a 20% lower value.

The relationship between collector performance and total system performance powering a Rankine cycle is shown clearly on Figure 1.1-9 (from Reference A-5). This figure plots the combined annual system performance, when the system consists of a Rankine cycle (defined as 75% of Carnot efficiency) and a collector operating at a fixed temperature. For each combination of collector type and Rankine cycle, there appears to be an optimum temperature of operation at which the system performance is maximum. Although the curves are relatively flat, they serve to demonstrate that operating temperature should not be an arbitrary choice, but depends upon the characteristics of the components selected. In addition, they imply that sophisticated, high temperature collectors might be cost effective because the resulting high system performance reduces the required collector area. For instance, comparing Case 6 and Case 1 on Figure 1.1-9, the ratio of system performance is about 3.3 to 1, at temperature levels of 500°F and 180°F respectively. If the cost of Case 6 was less than 3 times Case 1, it would be the most cost effective system.

The cost performance relationship for the collectors listed in table 1.1-12 is shown on Figure 1.1-10 by plotting cost per peak cooling rate (\$/Btu/hr) as a function of operating temperature level. The Rankine cooling COP was based on Curran's Rankine cycle curve at $T_C = 120^\circ\text{F}$ and $\text{COP} = 3.5$ (C-3). Each curve shows a minimum cost per cooling rate at an optimum temperature. It is interesting to note that the flat plate collector (B) costs \$0.225 at 260°F, and the evacuated collector (A) costs \$0.25 at 400 °F, while the first cost ratio between A and B is 1.5:1. This analysis does not include installation costs.

FIGURE 1.1 - 8

COLLECTOR EFFICIENCY CURVES

TABLE 1.1-12

STATE OF THE ART COLLECTORS AVAILABLE IN THE MARKET PLACE

<u>Type</u>	<u>Model</u>	<u>Manufacturer</u>	<u>Description</u>	<u>Collector Cost, \$/sq ft</u>
Flat Plate	LSC-1	Lennox	Double glazed with anti-reflective surface Black chrome selective surface	13.50
Evacuated Tube	Sunpak	Owens- Illinois	Double wall vacuum tube with CuO selective surface on inner wall. Shaped reflective backing surface.	21.25*
	TC-100	General Electric	Double wall vacuum tube with black chrome selective surface on inner metal fin. V-shaped reflective back surface (G4)	20.00
Concentrating		Northrup	Acrylic Fresnel lens, black chrome selective surface on tube.** Tracking unit. About 6X	12.50
	Slats	Sheldahl	Tracking reflecting segments, Teflon coated aluminum reflecting surface. Selective coated single glazed absorber (S2) about 40X.	29.00

* Does not include assembly labor or mounting panel.

** Upper temperature limit 600°F.

FIGURE 1.1 - 9

INFLUENCE OF SOLAR PANEL TEMPERATURE
ON ANNUAL OVERALL SYSTEM EFFICIENCY

FIGURE 1.1 - 10

FIRST COST PER PEAK ENERGY RATE
VS COLLECTION TEMPERATURE

1.1.2 PRELIMINARY PRODUCT SPECIFICATIONS

A preliminary Product Specification was developed as a guideline for the conceptual design. This specification addresses major features in the same manner used for other commercial unitary packaged equipment. Where specific details of the proposed design could not be projected, specifications from a current line of commercial air cooled water chillers were substituted as a yardstick. The proposed Product Specification follows as Table 1.1-13, and the major headings are discussed in detail below:

A. Description

A one-piece factory assembled outdoor package design follows the trend in the commercial water chiller market. It has been demonstrated that installed costs are much lower for packaged units than for applied equipment. Valuable floor space, or specially constructed penthouses are not required. Built-in protection features and factory quality control during assembly reduce service costs.

The present chiller market is shifting from water cooled to air cooled. Since 1974, the market has shifted from 2:1 in favor of water cooled to 1:1. The trend will continue as air cooled designs become more efficient. Although water cooled units use less electricity at rated conditions, the difference on a seasonal basis may not be large. There is an installation cost penalty for cooling towers, and the water problems associated with them result in higher maintenance costs, and may limit usage in off-seasons. The deployment of large numbers of water cooled units for solar cooling could create water supply problems reminiscent of the 1950's, which led many cities to limit the use of cooling towers.

Water chillers are normally used where long piping runs or multiple coil units are involved. Water is easier to pipe than refrigerant and presents fewer problems in multi-coil applications. However, outdoor water chillers have been sold primarily in the south where the unit and piping do not undergo extreme freezing conditions. Chillers are used primarily for commercial buildings, schools and offices. Individual units are predominant in the multi-family market today because of individual metering of utilities. However, for solar cooling, chillers fit the multi-family market well. A 25 ton unit can handle 10-20 apartments. Use of a diversity factor allows installation of a smaller tonnage unit.

B. Voltages

Commercial packaged air cooled water chillers in the 20 to 100 ton range are available with name plate voltages of 200, 230, 460 and 575. All require 3 phase current in either 50 Hz or 60 Hz. The design will be required to be adaptable to all voltages eventually, and this requirement will be a consideration in designing the high speed motor power supply.

C. Codes

Commercial packaged equipment must be U.L. approved in order to meet local codes and ordinances once installed (CSA in Canada). Presently, the Standard for Centrifugal Compressor Liquid Coolers is UL-465A (Ref. U-2), which is based on UL-465, Standard for Central Cooling Air Conditioners (Ref. U-1). The standard for hermetic motors is UL 984, Hermetic Refrigerant Motor-Compressors (Ref. U-3). The design of the package, piping and pressure vessels, and its electrical content will meet these standards. UL-465 cites the ANSI B9.1 Safety Code for Mechanical Refrigeration (Ref. A-9) which governs the piping and pressure vessel design, relief valve selection and pressure limiting means, as well as the use of the refrigerant. This code, in turn, cites Section VIII, Division I, of the ASME Boiler and Pressure Vessel Code (Ref. A-14), for pressure vessel design. Installation aspects of the design will meet the Natural Electric Code Requirements.

D. Ratings

Ratings will be determined according to ARI 550-77, Standard for Centrifugal or Rotary Water Chilling Packages (Ref. A-12). Off-design conditions will include the ranges of conditions specified in this standard under "Application Rating Conditions."

The cooling capacity of the conceptual design will be 25 tons. This is within the 20-100 ton size range of commercially available, unitary packaged air cooled water chillers. It is an ideal size for development, and scaling-up into an economical line of products. Based on past experience, smaller sizes would become increasingly difficult to achieve economical centrifugal compressor designs.

The minimum design goal for overall COP (OCOP) is 0.68. ASHRAE Standard 90-75R, Energy Conservation in New Building Design (Ref. A-13), states that indirect fired heat operated cooling equipment shall show a cooling COP not less than 0.68 after January 1, 1980. Previous calculations indicate that this type of Rankine vapor compression cycle can be designed to 0.68 OCOP.

E. Dimensions

As a preliminary specification, not all dimensions can be fixed. However, to enable the equipment to be shipped as a single piece, the width dimension is limited to 90 inches.

F. Features

Commercially available, air cooled packaged water chillers include the listed features.

Minimum condenser pressure is established by the capacity of the expansion device to handle a flow equal to the compressor displacement under low pressure difference. This minimum condenser pressure can be maintained by cycling fan motors off in several stages, allowing operation of the unit down to very low outdoor ambients.

A replaceable core filter-drier will be used to scavenge moisture in the system. The subatmospheric pressure at the evaporator may result in air leaks on some percentage of the units manufactured. Introduction of air will bring moisture into the system which will enter into a reaction with the refrigerant at the high temperature levels. The filter-drier is a practical solution to minimize the moisture problem on this sized equipment.

Freeze protection is necessary when using chilled water without antifreeze additives. Low suction temperatures, which can cause freezing and blockage of flow through the cooler can be limited with a low water temperature thermostat. To guard against freezing at low outdoor temperatures, thermostatically controlled heaters are located around the cooler shell under a heavy layer of insulation.

Flow switches on chilled water and solar water will be standard safety equipment to prevent operation without proper water flow. Water pumps are not included in commercial chillers because of the extreme application variations encountered.

G. Operating Ranges

These ranges are arbitrary in the preliminary specification. The lowest chilled water temperature will be limited by the low water temperature thermostat. There will be no upper limit as far as operation of the chiller goes, but proper control of humidity requires the leaving water to be limited to approximately 50°F by proper system design.

The maximum condenser air temperature under which commercial air cooled equipment properly operates is 115°F. This upper limit will be established by a high pressure limit switch on the condenser. With condenser pressure control through staged fans, the lower limit will probably be less than normal cooling operation requires, and cannot be determined at this point.

The exact boiler water range will be determined at a later date. The values of 250°F-300°F are based on the requirements of the RFP.

H. Glycol Operation

For outdoor installations in the northern climates, where below freezing ambients are common, or for special applications requiring lower chiller supply temperatures, glycol can be used. Ratings will be reduced because of the poor heat transfer properties of glycol solutions.

TABLE 1.1-13

**ENERGY SYSTEMS
CARRIER CORPORATION
SYRACUSE, NEW YORK**

PRODUCT SPECIFICATION

CONTRACT EG-77-C-03-1590
PROJECT E-3006, TASK 1.1.4

**HIGH TEMPERATURE SOLAR POWERED WATER CHILLER
PROPOSED PRODUCT SPECIFICATIONS - 1/9/78**

A. Description

One piece, outdoor air cooled chiller package. Unit to be completely factory assembled and charged, ready for electrical and piping connections in the field. Unit to be equipped with lifting brackets. Upward air discharge.

B. Voltages

230 - 3 - 60 (Prototype Design)
460 - 3 - 60 (Design Capability)

C. Codes

Design will meet the following codes:

UL-465 Standard for Central Cooling Air Conditioning

UL-465A Proposed Standard for Centrifugal Compressor Air Conditioning Liquid Chillers

ANSI B9.1 Safety Code for Mechanical Refrigeration

ASME Section VIII, Div. 1 - Unfired Pressure Vessel Code

National Electric Code

D. Rating

Capacity 88 kW (25 tons)
Input 110 kW (375,000 Btu/hr)
OCOP >0.68
Fan Power 2.75 kW (total)
Pump Power 1.0 kW
Auxiliary
Motor Power 30 kW*
S.R.N. 24*

Rating Conditions:

6.7 C (44 F) Leaving Chilled Water Temp. **
3.78 x 10⁻³ m³/s (60 gpm) Chilled Water Flow**
35 C (95 F) Entering Condenser Air Temp. **
18.4 m³/s (39,000 cfm) Condenser Air Flow
149 C (300 F) Entering Boiler Water Temp.
4.73 x 10⁻³ m³/s (75 gpm) Boiler Water Flow

E. Dimensions

Length 4.17 m (164 inches)*
Width 2.29 m (90 inches)
Height 2.03 m (80 inches)

* Present Commercial Line Values

** Per ARI 550-77 Standard for Centrifugal or Rotary Water Chiller Packages.

TABLE 1.1.13 (cont'd.)

ENERGY SYSTEMS CARRIER CORPORATION SYRACUSE, NEW YORK		PRODUCT SPECIFICATION	
		CONTRACT EG-77-C-03-1590 PROJECT E-3006, TASK 1.1.4	
		<u>HIGH TEMPERATURE SOLAR POWERED WATER CHILLER PROPOSED PRODUCT SPECIFICATIONS - 1/9/78</u>	
F. Features	Multiple Direct Drive Fans Fan Cycling on Condenser Pressure Control Replaceable Core Filter Drier 115 Volt Control Circuit Freeze Protection Thermostat and Cooler Heater Flow Switch Capacity Control *** Condenser Coil Grille (Chilled Water Pump Not Included)		
G. Operating Ranges	Leaving Chilled Water Temp. 1.7-10 C (35-50 F) Condenser Air -6.7-46.1 C (20-115 F)* Boiler Water Temperature 121.1-149 C (250-300 F)		
H. Glycol Operation	Capacity Correction @ 40% = 0.96 Chilled Water Flow Correction @ 40% = 1.10		
		*** Undefined Method * With Head Pressure Control	

1.2 CYCLE STUDY

The first task undertaken to conceptualize the design was a study of dual loop Rankine powered vapor compression cycles. A simple computer program was written to calculate cycle performance, heat and mass flows, and state points for a number of working fluids and configurations. The study was limited to dual cycles with single working fluids of the halogenated fluorocarbon type. The objectives of this study were:

1. Evaluation of cycle parameters to characterize the optimum performance Rankine cycle/vapor compression cycle configuration.
2. Selection of the optimum working fluid based on evaluation of thermodynamic performance, heat transfer properties, thermal stability, and turbo/compressor performance.
3. To provide preliminary working data for subsequent design of the turbo/compressor, pump and heat exchangers.

1.2.1 COMPUTER PROGRAM - RANK05

As a first step in the design process, a simple computer model of the dual loop Rankine driven vapor compressor cycle, name RANK05, was written. Figure 1.2-1 is an example of a pressure-enthalpy diagram for R-114 showing the state point designations used in the program. This program uses a proprietary subroutine package to compute refrigerant properties, RDCI699C. Refrigerants which are currently available in this program are R-11, R-12, R-22, R-113, R-114, R-500, R-502, R-13B1, and R-13. RANK05 has three regenerator heat exchanger and one liquid suction heat exchanger options open to the user. These heat transfer processes are indicated on Figure 1.2-1. Also, in order to handle wet expansion or a supercritical refrigerant, the option is open to specify both the boiler pressure and a leaving temperature above saturation. Computation is done on a "per ton" basis. Inputs include the following:

1. NREF - refrigerant
2. TB - saturated boiling temperature
3. TC - Saturated condensing temperature
4. TE - saturated evaporating temperature
5. η_e - turbine efficiency
6. η_c - compressor efficiency
7. η_p - pump efficiency
8. Tsup - superheat leaving evaporator
9. Tsub - subcooling leaving condenser
10. N_{HX} - regenerator effectiveness
11. Z - type of regenerator heat exchange, user option
 - 1, between liquid and vapor leaving compressor only
 - +1, between liquid and vapor leaving expander only
 - 0, between liquid and mixed vapor
12. N_{LSH} - liquid suction heat exchanger effectiveness
13. NST - specific speed of turbine
14. DST - specific diameter of turbine
15. NSC - specific speed of compressor
16. DSC - specific diameter of compressor
17. TONS - capacity for determining absolute wheel size and speed.

NOTE: A second generation version of this program, RANK06, is listed in Appendix A1.2, along with an abstract. This program allows inputs of supply water and air stream temperatures, and computes UA and LMED values of the major heat exchangers, but is otherwise the same as RANK05.

The manner of calculating mass flow, work, and heat quantities around the cycle is done as follows:

FIGURE 1.2-1

PRESSURE-ENTHALPY DIAGRAM
R113-DUAL LOOP RANKINE COMPRESSOR CYCLE

1. P, T, H, V, and S properties are determined for points 2, 5, 7, 8, 10, 14 based on input data.
2. T10 is found using the effectiveness formula for the liquid suction heat exchanger (LSHX).

$$T_{10} = T_{101} + \eta_{LSH} * (T_7 - T_{101}).$$

and P, H, V and S are determined.

3. Point 11 is the isentropic compression discharge of the compressor and is determined knowing P5 and constant entropy S10. The actual discharge condition is found as:

$$H_{12} = H_{10} + (H_{11} - H_{10}) / \eta_c.$$

4. Similarly, point 4 is the isentropic expansion exhaust and is determined knowing P5 and constant entropy S2. The actual discharge condition is found as:

$$H_6 = H_2 - (H_2 - H_4) * \eta_e$$

P, H, V and T are then found for points 4 and 6.

5. For one ton capacity, the vapor compression mass flow is:

$$WRF = 12,000 / (H_{10} - H_7)$$

where: $(H_7 - H_{71}) = (H_{10} - H_{101})$ is the LSHX heat balance.

6. The work out of the turbine must equal the work of the compressor:

$$WRF * (H_{12} - H_{10}) = WRC * (H_2 - H_6)$$

therefore, the Rankine cycle mass flow can be found:

$$WRC = WRF * (H_{12} - H_{10}) / (H_2 - H_6).$$

7. The regenerator solution depends upon the user option type chosen:

$$\text{For type +1; } T_{13} = T_6 - (T_6 - T_7) * \eta_{HX}$$

where point 13 is the leaving vapor condition

$$\text{and } Q_{HX} = (H_6 - H_{13}) * WRC \quad ; \text{ Btu/(Hr-ton)}$$

$$\text{For Type -1; } T_{13} = T_{12} - (T_{12} - T_7) * \eta_{HX}$$

where point 13 is the leaving vapor condition.

$$\text{and } Q_{HX} = (H_{12} - H_{13}) * WRF \quad ; \text{ Btu/(hr-ton)}$$

For type 0, point 13 is the entering mixed vapor condition and T13 can be found from H13 where:

$$H_{13} = (H_6 * WRC + H_{12} * WRF) / (WRC + WRF)$$

$$T_1 = T_7 + (T_{13} - T_7) * \eta_{HX}$$

$$Q_{HX} = (H_1 - H_7) * WRC.$$

8. Other heat quantities are:

$$\text{For preheater } Q_{PRHT} = (H_{14} - H_1) * WRC \quad ; \text{ Btu/(hr-ton)}$$

$$\text{For boiler } Q_{BOILR} = (H_2 - H_{14}) * WRC \quad ; \text{ Btu/(hr-ton)}$$

$$\text{Total input } Q_{BUIL} = Q_{BOILR} + Q_{PRHT} \quad ; \text{ Btu/(hr-ton)}$$

$$\text{For condenser } Q_{COND} = Q_{BOILR} + Q_{EVAP} \quad ; \text{ Btu/(hr-ton)}$$

where: $Q_{EVAP} = 12,000 \text{ Btu/hr.}$

9. Efficiencies are:

For Rankine cycle $N_{RC} = (H_2 - H_6) * WRC / Q_{BOILER}$

For Vapor compression cycle $COP = 12,000 / [(H_{12} - H_{10}) * WRF]$

For overall $OCOP = N_{RC} * COP$

10. Pump work is:

$QPUMP = WRC * 144 * V_7 * (P_B - P_C) / N_p * 33000$

$QPUMP = WRC / 60 * V_7 * (P_B - P_C) * 144 / (N_p * 33000); \text{hp/ton.}$

1.2.2 COMPARISON OF REFRIGERANTS IN BASIC CYCLES

Design conditions and expected ranges of operating conditions were established for comparison of refrigerants and cycles. These are given in Table 1.2-1. Also given are the first estimate of corresponding boiling, condensing and evaporating temperatures within the cycle.

A set of cycle calculations were made to compare the performance at design conditions between four refrigerants: R-11, R-12, R-113, and R-114. The results are shown in Table 1.2-2. Turbine, compressor, and pump efficiencies used were based on the original design goals. A sample computer printout, Run 23 for R-114, is shown as Table 1.2-3. Cycle state points correspond to Figure 1.2-1.

On the basis of this preliminary screening, several points of comparison were made.

- The working fluid with the highest OCOP is R-11.
- The working fluid with the lowest pump work is R-113.
- The heat exchanger costs would be highest on R-12 -- due to very high pressures and larger heat quantities.

TABLE 1.2-1

PRELIMINARY DESIGN AND OPERATING CONDITIONS

	<u>Design Conditions</u>	<u>Operating Range</u>
Outdoor Ambient Temperature	95°F	75-115°F
Condensing Temperature	115°F	
Leaving Chilled Water Temperature	45°F	37-50°F
Evaporator Temperature	40°F	
Entering Solar Hot Water Temperature	300°F	220-300°F
Boiler Temperature	273°F	

TABLE 1.2-2

COMPARISON OF DUAL LOOP RANKINE CYCLES WITH DIFFERENT WORKING FLUIDS

<u>Run</u>	<u>2A</u>	<u>3A</u>	<u>23</u>	<u>16</u>
Working Fluid	R-11	R-113	R-114	R-12
Boiler Temperature	273	273	273	273
Boiler Pressure	231.0	131.7	383.39	850
Condenser Temperature	115	115	115	115
Condenser Pressure	30.35	14.0	58.11	161.45
Evaporator Temperature	40	40	40	40
Evaporator Pressure	7.02	2.60	15.08	51.66
Turbine Efficiency	0.8	0.8	0.8	0.8
Compressor Efficiency	0.8	0.8	0.8	0.8
Pump Efficiency	0.5	0.5	0.5	0.5
Rankine Cycle Efficiency	0.153	0.147	0.137	0.136
Vapor Compression Cycle COP	4.837	4.737	4.449	4.491
Overall COP	0.739	0.697	0.609	0.611
Boiler Input, Btu/hr-ton	16,237	17,220	19,708	19,649
Condenser Rejection, Btu/hr-ton	29,227	30,757	33,191	33,820
Pump Work, hp/ton	0.062	0.039	0.165	0.533

LIMIT SECONDS(03)

?

114,273,115,.8,.5,.8,40,5,1,.8

SPECIFIED CYCLE PARAMETERS FOR DUAL LOOP, R 114

1. SATURATED BOILER TEMPERATURE, F = 273.0
2. SATURATED CONDENSER TEMPERATURE, F = 115.0
3. SATURATED SUCTION TEMPERATURE, F = 40.0
4. CONDENSER SUBCOOLING, F = 5.0
5. EVAPORATOR SUPERHEAT, F = 1.0
6. EXPANDER EFFICIENCY = 0.800
7. FEED PUMP EFFICIENCY = 0.500
8. DESUPERHEAT EXCHANGER EFFICIENCY = 0.800
9. COMPRESSOR EFFICIENCY = 0.800

TABLE 1.2-3

CALCULATED RANKINE CYCLE PERFORMANCE

1. BOILER PRESSURE, PSIA = 383.4
2. CONDENSER PRESSURE, PSIA = 58.1
3. EXPANDER EXHAUST TEMPERATURE, F = 159.3
4. RANKINE CYCLE MASS FLOW, LB/HR-TON = 304.153
5. RANK CONDENSER REJECTION, B/HR-TON = 18493.723
6. RANKINE BOILER INPUT, B/HR-TON = 19708.750
7. REGENERATOR HEAT EXCHANGE, B/HR-TON = 1482.124
8. PERCENT REGENERATOR HEAT = 6.994
9. BOILER SENSIBLE HEAT, B/HR-TON = 12053.488
10. PERCENT SENSIBLE HEAT = 61.158
11. SPECIFIC VOLUME, EXPANDER INLET, CFP = 0.063
12. SPECIFIC VOLUME, EXPANDER OUTLET, CFP = 0.612
13. EXPANSION RATIO = 6.597
14. EXPANDER DISPLACEMENT, CF/MIN-TON = 3.103
15. EXPANDER HEAD, FT-LBS/LB = 0.014
16. THEO. CYCLE EFFICIENCY, W/O HX = 0.127
17. ACTUAL CYCLE EFFICIENCY, W/HX = 0.137
18. FEED PUMP WORK INPUT, HP/TON = 0.165
19. FEED PUMP DISPLACEMENT, CF/MIN-TON = 0.058

CALCULATED REFRIGERATION CYCLE PERFORMANCE

1. EVAPORATOR PRESSURE, PSIA = 15.1
2. DISCHARGE TEMPERATURE, F = 106.5
3. REFRIGERANT MASS FLOW, LB/HR-TON = 287.994
4. COMPRESSOR WORK INPUT, HP/TON = 1.060
5. REFRIG CONDENSER REJECTION, B/H-TON = 14697.145
6. SPECIFIC VOLUME, SUCTION, CF/LB = 2.000
7. SPECIFIC VOLUME, DISCHARGE, CF/LB = 0.541
8. COMPRESSION RATIO = 3.854
9. COMPRESSOR DISPLACEMENT, CF/MIN-TON = 9.602
10. COMPRESSION HEAD, FT-LBS/LB = 0.010
11. SONIC VELOCITY COMPRESSOR INLET, F/S = 382.089
12. REFRIGERATION COP = 4.449

CALCULATED OVERALL CYCLE PERFORMANCE, /TON

1. OVERALL COP, W/O PUMP = 0.609
2. TOTAL CONDENSER REJECTION, B/HR-TON = 33190.867

STATE POINT	T	P	H	U	S
0	110.000	383.392	34.307		
1	129.058	383.392	39.179		
14	273.000	383.392	79.809		
2	273.000	383.392	103.978	0.063	0.171
4	115.000	58.112	92.894		0.171
5	115.000	58.112	86.888		
6	159.289	58.112	95.111	0.612	
7	110.000	58.112	34.307	0.011	
9	40.000	15.078	34.307		
10	41.000	15.078	75.974	2.000	0.155
11	96.252	58.112	83.466		0.155
12	106.536	58.112	85.339	0.541	
13	133.822	58.112	90.358		

READY

1.2.3 COMPARISON OF REFRIGERANTS ON COMPRESSOR GEOMETRY

Data from the cycle computations previously described were used to generate some preliminary details of the compressor. The first analysis was based on a non-dimensional specific speed parameter which has been shown to correlate with compressor stage performance. Specific diameter versus specific speed compressor curves, Figure 1.2-2, show the optimum specific speed for this type of machine to be about 100 and the optimum specific diameter to be about 1.28. Table 1.2-4 compares radial bladed compressors at the 25 ton capacity level, for four refrigerants.

On the basis of this preliminary screening, several points of comparison were made:

- The refrigerant requiring the largest diameter lowest speed is R-113. Generally, larger impellers can be made more efficient.
- Highest mach numbers occur with R-113. Generally, high mach numbers will reduce stage efficiency.
- R-11 requires the lowest horsepower.
- R-113 requires the largest piping connections.
- R-12 results in the smallest impeller and highest speed. There may be cost penalties associated with this design. Efficiency goals would be difficult to reach with such a small impeller.
- Tip speeds for all refrigerants are low, and therefore, stress levels will not be a critical consideration in impeller design.

Compressor peak efficiency may well vary with size. Figure 1.2-3 indicates the range of efficiencies for small wheels projected by MTI. This compares well with data from Airesearch (R-3). The next iteration of cycle computations will utilize values from this curve as compressor efficiency estimates. The level of the curve was generated using a component level analysis program (i.e., R-11 @ 45° impeller backsweep angle).

FIGURE 1.2-2

Specific Speed (N_s), Specific Diameter (D_s)
Relationship for Centrifugal Compressor

TABLE 1.2-4

COMPARISON OF RADIAL BLADED
CENTRIFUGAL COMPRESSORS FOR VARIOUS REFRIGERANTS
FOR 25 TONS CAPACITY

Run Number	2A	3A	27	16	
Refrigerant	R-11	R-113	R-114	R-12	
Inlet Temperature	41	41	41	41	°F
Inlet Pressure	7.03	2.66	15.08	51.66	psia
Exit Temperature	148.0	117.8	106.5	136.2	°F
Exit Pressure	30.3	14.0	58.1	161.4	psia
Pressure Ratio	4.32	5.28	3.85	3.13	
Mass Flow	4536	5602	7200	6244	lbs/hr
Volume Flow	411.2	1000	240	80.8	cfm
Specific Speed*	100	100	100	100	
Adiabatic Head	10634	8795	7288	8324	ft
Horsepower	24.4	24.9	26.5	26.3	hp
Rotational Speed	40000	22250	39440	75120	rpm
Impeller Tip Speed	690	627	571	610	ft/sec
Impeller Tip Diameter	3.95	6.46	3.32	1.86	in
Specific Diameter **	1.28	1.28	1.28	1.28	
Inlet Pipe Diameter (100 ft/sec velocity)	3.54	5.53	2.71	1.57	in
Inducer Tip Mach Number	1.03	1.10	0.99	0.90	

* Specific Speed = $N\sqrt{Q}/H^{3/4}$
 where: N = rotational speed = rpm
 Q = volume flow = ft³/sec
 H = adiabatic head = ft

** Specific Diameter = $DH^{1/4}/\sqrt{Q}$
 where: D = impeller tip diameter = ft
 H and Q as defined above.

FIGURE 1.2 - 3

ADIABATIC EFFICIENCY VS COMPRESSOR IMPELLER
DIAMETER FOR A 25 TON MACHINE

KEY:

— REPORT N76-30655 1975, AIRESEARCH; REF. R-3

↑ FROM MECHANICAL TECHNOLOGY INC.

⊠ VALUES USED IN CYCLE COMPUTATIONS

NOTE:

IMPELLER DIAMETER BASED ON $N_s = 100$

$D_s = 1.28$

TONS = 25

1.2.4 COMPARISON OF CYCLE MODIFICATIONS

The four refrigerants were carried through a number of cycle computations to compare the effect of heat exchanger configuration, operating temperature levels, heat exchanger effectiveness, and turbo/compressor performance. A preliminary evaluation of conventional air cooled condenser heat transfer suggested that a more realistic condensing temperature corresponding to 95°F entering air would be 120°F. Compressor efficiency levels were adjusted from the 0.80 used previously to the values shown on Figure 1.2-3 which were thought to be more realistic values for comparing refrigerants. For R-12, it was discovered that the refrigerant properties equations did not properly model the saturated vapor dome in the super critical region, and values reported previously were in error. The computer code was revised to correct this problem for subsequent runs.

The first comparisons were made to determine the best regenerator configuration for each refrigerant. Figure 1.2-4 defines the three types of regenerator configurations which were explored. For dry expansion fluids such as R-114 and R-113, where the isentropic expansion line moves away from the saturated vapor dome and results in substantial superheat at the expander exhaust, a heat exchange can be made between the exhaust vapor and the relatively cool condensate to improve the cycle performance. For wet expansion fluids, such as R-11 and R-12, the isentropic compression line moves away from the saturated vapor line and results in substantial superheat at the compressor discharge, a heat exchange can be made between the discharge vapor and the condensate. The slope of the actual compression and expansion processes is accentuated by the efficiencies of these components. At the efficiency and temperature levels chosen for these computations, R-11, a wet expansion fluid, did not expand into the vapor dome, and consequently, superheat at the expander inlet was not necessary. For R-12, vapor was superheated to 273°F from a subcritical boiling temperature of 220°F, with a resulting expansion line well outside of the vapor dome. In the cases of wet compression fluids, R-113 and R-114, to avoid possibility of liquid droplets forming in the compressor, superheat was required at the compressor suction. One way to superheat the vapor entering the compressor is to heat the vapor through heat exchange with the condenser condensate. Such a device, a liquid-suction heat exchanger (LSHX), will be discussed later.

Results of computations on various regenerator cycles are reported in Table 1.2-5. Significant improvements in Rankine cycle efficiency are shown with heat exchange between liquid and either compressed vapor or turbine exhaust vapor or both, depending upon the properties of the working fluid. The greater improvement of R-12 occurred because of the large superheat entering the expander.

If a liquid suction heat exchanger is used in conjunction with the regenerator, additional and significant cycle improvements occur. Normally, for conventional compression cycles, a heat exchange between the warm liquid leaving the condenser and the cold vapor leaving the evaporator results in a relatively small improvement in the vapor compression cycle COP which is not sufficient to justify the cost. The increased latent heat capacity of the refrigerant is partially offset by the increased compression work due to the higher superheat entering the compressor. However, in conjunction with the regenerator, this heat is transferred to the Rankine

FIGURE 1.2 - 4
 REGENERATOR HEAT EXCHANGER CONFIGURATION

TYPE +1

TYPE -1

TYPE 0

TABLE 1.2-5

COMPARISON OF REGENERATOR CYCLES $\pm$

Run	100	401	201	300
Refrigerant	R-11	R-113	R-114	R-12
Refrigerant Type*	Wet	Dry	Dry	Wet
Compressor Efficiency	0.8	0.825	0.75	0.725
Turbine Efficiency	0.8	0.8	0.8	0.8
Regenerator Effectiveness	0.8	0.8	0.8	0.8
Regenerator Type	-1	+1	+1	0
Rankine Cycle Efficiency w/o HX	0.140	0.131	0.124	0.106
Rankine Cycle Efficiency w/HX	0.148	0.147	0.138	0.124
Vapor Compression COP	4.434	4.442	3.745	3.682
Overall COP	0.656	0.651	0.518	0.457
Base COP, w/o Regenerator	0.621	0.582	0.464	0.39
% Improvement OCOP	5.7	11.8	11.6	17.1

* Refers to the Isentropic Expansion Process

$\pm$ Base Conditions: TB = 273°F (except R-12; TB = 220°F, superheat 53°F)

TC = 120°F

TE = 40°F

TSUP = 1°F

TSUB = 5°F

cycle reducing the input of new energy and improving the overall COP. In other words, the compressor acts like a heat pump, increasing the temperature level of the superheated discharge vapor and making the LSHX energy available for transfer to the Rankine cycle with the regenerator. The liquid suction heat exchange cycle is shown schematically on Figure 1.2-5. For the pressure-enthalpy diagram, the reader is referred back to Figure 1.2-1. The results of the combined regenerator and LSHX cycle performance and optimum regenerator configuration are shown in Table 1.2-6. The total cycle improvement from the combined heat exchange ranges from 15 to 30%. The addition of the LSHX contributes from 10 to 16% to the overall COP, even though it only contributes from -0.2 to 8.3% to the vapor compression COP. This synergistic effect is an important contribution to making this cycle competitive.

Some improvement in the R-12 cycle performance was obtained by going from a 273°F superheated cycle to a 273°F supercritical cycle, where the condition entering the turbine was chosen for minimum superheat at the turbine exhaust. These cycles are compared on Figure 1.2-6, and results given in Table 1.2-7. The supercritical R-12 cycle is 16% more efficient than the superheat cycle and approaches R-114 OCOP. However, the increase in pump work and the higher cycle pressure would be serious drawbacks to implementing such a design.

Performance of the four refrigerants was evaluated over the range of boiling temperatures corresponding roughly to the solar water temperature range of 220-300°F. Figure 1.2-7 shows the overall COP of each refrigerant cycle as a function of boiling temperature (in the case of R-12, it is the temperature entering the turbine). The relative ranking of these refrigerants remains nearly constant over this temperature range.

Throughout the foregoing comparisons, the performance ranking of the refrigerants has remained the same, with R-11 giving the highest OCOP and R-12 the lowest. Although the OCOP of R-113 is slightly lower than R-11, the pump work is also lower, suggesting that the true difference in performance in a system would be less. In the proposed system, the pump will be electrically driven, making it necessary to find a means to combine pump work, expressed as an operating cost, with OCOP, expressed as system first cost, in order to compare the refrigerants. This relationship, called the Figure of Merit, F, is defined as the reciprocal of simple payback, P.

$$F = 1/P$$

Payback is defined as the ratio of system cost, C, to annual savings, S.

$$P = C/S$$

$$F = S/C$$

A solar cooling system's cost is strongly related to the OCOP of the cooling unit. This is because the area of collector will be inversely proportional to the OCOP, and the collector is the major cost of the system. The assumption is made that the cooling unit cost is the smaller part of the total system and any cost variation with OCOP is negligible.

$$C \approx 1/\text{OCOP}.$$

FIGURE 1.2-5
SIMPLIFIED FLOW SCHEMATIC
DUAL LOOP, RANKINE WATER CHILLER WITH LIQUID-SUCTION HEAT EXCHANGER

TABLE 1.2-6

COMPARISON OF COMBINED REGENERATOR AND LIQUID SUCTION HEAT EXCHANGER CYCLES [±]

Run		405	203	303
Refrigerant	R-11	R-113	R-114	R-12
Compressor Efficiency	0.8	0.825	0.75	0.725
Turbine Efficiency	0.8	0.8	0.8	0.8
Regenerator Effectiveness	0.8	0.8	0.8	0.8
LSHX Effectiveness	0.7	0.7	0.7	0.7
Regenerator Type	-1	0	0	0
Rankine Cycle Efficiency	0.161	0.153	0.146	0.134
Vapor Compression COP	4.425	4.621	4.057	3.757
Overall COP	0.714	0.707	0.593	0.504
Base COP, w/HX	0.621	0.582	0.464	0.39
% Improvement OCOP	15.0	21.5	27.8	29.2
% Improvement in VC-COP	-0.2	4.0	8.3	2.0

[±] Base Conditions: TB = 273°F (except R-12; TB = 220°F, Superheat = 53°F)

TC = 120°F

TE = 40°F

TSUP = 1°F

TSUB = 5°F

FIGURE 1.2-6

PRESSURE ENTHALPY DIAGRAM
FOR REFRIGERANT 12

KEY:

- RUN 305, SUPERCRITICAL CYCLE
- RUN 303, SUPERHEATED CYCLE

TABLE 1.2-7

COMPARISON OF R-12 SUPERHEAT AND SUPERCRITICAL CYCLES

Run	303	306
TB, Boiling Temperature	220	>Critical
T2, Temperature Entering Turbine	273	273
TC	120	120
TE	40	40
Compressor Efficiency	0.725	0.725
Turbine Efficiency	0.8	0.8
Regenerator Type	0	-1
Regenerator Effectiveness	0.8	0.8
LSHX Effectiveness	0.7	0.7
Rankine Cycle Efficiency	0.134	0.155
Vapor Compression COP	3.757	3.757
Overall COP	0.504	0.584
Boiling Pressure, psi	524.4	700.0
Condensing Pressure, psi	172.3	172.3
Evaporating Pressure, psi	51.6	51.6
Pump work, hp/ton	0.28	0.41

FIGURE 1.2-7
COMPARISON OF OCOP FOR FOUR REFRIGERANTS
AS A FUNCTION OF TEMPERATURE

Savings is the difference between the operating cost of a conventional electric chiller and the operating cost of the solar system. These costs are assumed to be proportional to the horsepower involved. In the case of the solar cooling machine, we will only consider the feed pump horsepower.

$$S = HP_C - HP_S$$

where: HP_C = horsepower of a conventional chiller per ton;
 ≈ 1.28 for an R-12 vapor compression chiller

HP_S = solar feed pump horsepower.

$$S = 1.28 - HP_S$$

Therefore,

$$F = OCOP * (1.28 - HP_S).$$

A comparison of the Figure of Merit of the four refrigerants is given in Table 1.2-8. Inclusion of other solar system power requirements, such as water pump horsepower and additional fan horsepower, which might be reasonably consistent between the refrigerants, will only tend to accentuate the effect of feed pump horsepower. Therefore, it is felt that this evaluation is conservative, and clearly demonstrates the equivalency of R-113 and R-11, and their advantage over the other refrigerants.

TABLE 1.2-8

COMPARISON ON FIGURE OF MERIT BASIS

Run	112	405	203	306
Refrigerant	R-11	R-113	R-114	R-12
HP_S	0.07	0.041	0.188	0.41
OCOP	0.714	0.707	0.593	0.584
Figure of Merit	0.86	0.876	0.646	0.508

The R-113 cycle was chosen to examine the effect of other operating variables on performance. These include the effect of condensing temperature, evaporating temperature and boiling temperature on overall COP as well as Rankine cycle efficiency and vapor compression cycle COP. These results are shown on Figure 1.2-8 and several points can be made.

- 1°F increase in evaporation temperature improves the OCOP about 1%.
- An increase in boiling temperature of 2°F increases OCOP by about 1%.
- The condensing temperature influences both cycles in the same direction and the effect on the overall COP is the product of the two. Decreasing the condensing temperature one degree increases the OCOP about 1.5%.

FIGURE 1.2-8
EFFECT OF OPERATING CONDITIONS ON R113
DUAL LOOP CYCLE PERFORMANCE

□ - MAX. LIFT; TC = 140, TE = 30
 x - STD. CONDITIONS
 ⊕ - MIN. LIFT; TC = 95, TE = 50

Maximum and minimum lift performance points are shown on Figure 1.2-8. These are defined as the maximum and minimum pressure ratio conditions which the compressor will encounter in normal conditions and tend to show the range of performance expected.

Analysis of the method of cycle computation reveals that the effect of turbine and compressor efficiency on overall COP is a direct ratio function (i.e., a 1% increase in efficiency of either, increases the overall COP 1%).

It must be stressed here that these computations are for design points. In each case, the physical size of the heat exchangers and turbo/compressor would be different. Over this range of conditions, results for a fixed design machine might be different.

Again referring to the condenser performance in Figure 1.2-8, it can be observed that the condensing temperature influences the vapor compression cycle performance to a larger degree than it influences the Rankine cycle performance. This is because a given change in condensing temperature is a larger percentage of the sink-source temperature difference of the vapor compression cycle than it is of the Rankine cycle for the high boiling temperature chosen. This comparison can better be observed when plotted against the relative cycle performance as in Figure 1.2-9.

This effect suggests that a series arrangement of the air flow to the vapor compression condenser and the Rankine condenser could be used to advantage. By lowering the vapor compression cycle condensing temperature with cool inlet air at the expense of the Rankine condensing temperature, a trade-off in performance can be made.

A heat transfer analysis was made to determine the condensing temperatures which result in the log mean temperature difference (LMED) of the series condensers being equal to each other and equal to the LMED of a common condenser. Figure 1.2-10 superimposes diagrams of the two condenser arrangements for R-113. Air flow is assumed to be 1544 cfm/ton. The LMED is 10.9°F for the common condenser. If the coils are arranged in series with the vapor compression condenser on the entering air side, the vapor compression condensing temperature can be lowered from 120°F to 111°F at an LMED of 10.9°F. The Rankine condenser must then be raised from 120°F to 126°F to maintain an LMED of 10.9°F. If the heat rejection remains about the same, then the total heat transfer surface does not change. However, there is a net improvement in overall cycle performance. Referring back to Figure 1.2-9, at 111°F condensing, the vapor compression cycle improves about 17% while at 126 condensing, the Rankine cycle loses 3%, for a net improvement of 13%.

Figure 1.2-11 is a cycle diagram showing the piping arrangement with separate condensers. Since the Rankine condenser would be at a higher pressure level than the vapor compression condenser, there is a need for sealing between the systems. This would require a shaft seal in the turbo/compressor, and a separation of the vapor paths in the regenerator, as well as separate purge controls. It is also obvious that once separate cycles are accepted, there is an opportunity to consider the use of different working fluids in each cycle. However, such an evaluation was not undertaken in this cycle study.

FIGURE 1.2 - 9

RELATIVE CYCLE PERFORMANCE
AS A FUNCTION OF CONDENSING TEMPERATURE

FIGURE 1.2-10

SERIES VS COMMON CONDENSER
ANALYSIS OF HEAT EXCHANGE POTENTIAL

	LMED
①	11.0
②	10.9
COMMON	10.9

FIGURE 1.2-11
SIMPLIFIED FLOW SCHEMATIC
DUAL LOOP, RANKINE WATER CHILLER WITH SERIES CONDENSERS

Performance of two other refrigerants was compared on a theoretical basis. RC 318 is known to be one of the most stable refrigerants with a boiling point of 24°F, slightly below R-114. R12B1 is a new, high efficiency refrigerant of about the same boiling point. Table 1.2-9 compares these two refrigerants with R-114 and R-113 on an ideal basis, i.e., no turbine or compressor efficiencies. Cycle efficiencies do not promise to be as high as R-114 or R-113. In addition, pump work is greater. Even though RC 318 is very stable, the low performance does not warrant further evaluation. There is not sufficient stability, toxicity, or environmental data on R12B1 to consider it further, particularly since it has a slightly lower performance and higher pump work than R-114.

TABLE 1.2-9

COMPARISON OF OTHER REFRIGERANTS ON AN IDEAL BASIS

Refrigerant	R-114	R-113	RI2B1	RC318
TB °F	273	273	273	273
Boiler Pressure, psia	390	130	435	500
TC °F	120	120	120	120
Condenser Pressure, psia	65	15.5	79	91
TE °F	40	40	40	40
Evaporator Pressure, psia	15	2.6	21	22
Rankine Cycle Efficiency w/o HX*	0.147	0.155	0.165	0.137
Rankine Cycle Efficiency w/HX	0.162	0.173	--	0.158
Vapor Compression COP	5.4	5.55	5.23	3.35
Overall COP w/o LSHX	0.794	0.85	0.86	0.458
Overall COP w/LSHX	0.89	0.96	--	0.632
Pump Factor	1.0	0.267	1.14	2.18

* HX = Regenerator

1.2.5 REFRIGERANT SELECTION MATRIX

There are a great many factors to be considered in the selection of a refrigerant as a working fluid in this cycle beyond those related to performance which have been discussed previously. Choice of a refrigerant has a strong bearing on those parameters affecting the marketability of a product, such as:

- Reliability
- First Cost
- Acceptance
- Development Cost
- Operating Cost

In order to give consideration to these somewhat unrelated parameters, a system of ranking and weighting was used to create a selection matrix. Each of the major parameters was assigned a maximum point weight as shown on Table 1.2-10, and was subsequently subdivided into items which could be evaluated with some precision, to make up the actual point weight. These subdivisions and their point weights are also indicated on Table 1.2-11.

The detailed evaluation of each item is in Appendix A1.2. Certain key factors should be noted. High temperature refrigerant thermal stability data in the literature is difficult to relate to a specific application. Many opinions are based on experience with equipment which might not relate to this design. However, from the data and experience available at this time, use of R-11 at temperatures above 225-250°F is generally not recommended. The refrigerant matrix really does not adequately express this finding.

Considerable work was done on comparison of heat transfer rates between refrigerants in order to evaluate their impact on heat exchanger design. However, because the solar collector portion of the system is expected to cost about three times the cooling unit cost, the differences in heat transfer had very little impact on first cost of the system. The overriding refrigerant characteristic was the resulting cycle OCOP which governs the needed collector area and thus strongly influences system first cost and operating costs.

The final matrix is presented in Table 1.2-11. The results show the strong similarity to the relative performance levels presented previously. R-113 would seem to be best selection. On the negative side, R-113 may be less reliable than R-12 and R-114. The findings in this regard are:

- Toxicity. R-113 is a class 4 refrigerant under the U.L. group classification (Group 4- gasses or vapors up to 2-1/2%; no apparent injury).
- Pressures. The R-113 evaporator will be under a vacuum, which could require either a built-in purge unit or extra maintenance.
- Stability. R-113 is not as stable as R-114 or R-12 at the design temperatures.

TABLE 1.2-10

PARAMETERS AND WEIGHTINGS FOR REFRIGERANT SELECTION MATRIX

<u>Reliability</u>	<u>400</u>	<u>First Costs</u>	<u>400</u>	<u>Acceptance</u>	<u>100</u>
Fire Explosion	100	Collector	300	Serviceability	10
Toxicity	150	Compressor	10	Noise and Vibration	30
Environment	20	Heat Transfer	73	Size and Weight	30
Operating Pressures	20	Turbine	10	Installation Costs	30
Refrigerant Stability	100	Pump	5		
Effect on Materials	10	Fans and Motors	2		
<hr/>					
<u>Operating Costs</u>	<u>400</u>	<u>Development Costs</u>	<u>100</u>		

TABLE 1.2-11

REFRIGERANT SELECTION MATRIX

	<u>Relative Weight</u>	<u>R-11</u>	<u>R-12</u>	<u>R-113</u>	<u>R-114</u>
Reliability	400	208*	304	241	272
First Costs	400	394	154	370	249
Acceptance	100	85	61	85	69
Development Costs	100	75	75	75	75
Operating Costs	400	308	21	322	207
TOTAL	1400	1070*	615	1093	872

* Refrigerant not stable at operating temperatures expected.

1.2.6 PRELIMINARY COST PERFORMANCE OPTIMIZATION

Preliminary heat exchanger sizing and balancing for an R-113 cycle was accomplished by making performance trade-offs against total system costs. The computer program RANKØ5 was modified to RANKØ6 (Appendix A1.2) to calculate heat exchanger "sizes." The "size" is reported as the "UA" value, from the expression:

$$Q = U * A * (LMED)$$

where: Q = heat transfer rate Btu/hr
U = overall heat transfer coefficient (Btu/(hr-ft²F)
A = surface area.
LMED = log mean effective temperature difference.

The computer used values of temperature for both the cycle and external fluid streams to compute LMED, Q, and UA on a per ton basis for the following heat exchangers:

- boiler
- preheater
- condenser
- cooler

LMED, Q, and UA values for the regenerator and LSHX were calculated for input values of effectiveness.

Input values of external fluid streams were:

- entering solar water; 300°F @ 3 gpm/ton
- leaving chilled water; 45°F @ 2.5 gpm/ton
- entering air; 95°F @ 1544 cfm/ton

RANKØ6 also accounts for heat exchanger pressure drops, by computing the state point temperatures resulting from input values of pressure drop for each heat exchanger.

A cost structure was arbitrarily assumed for this study, based on the values assumed for the Refrigerant Selection Matrix study. A key assumption was made that the total installed cost of the base point design solar system would be four times the installed cost of the chiller. Therefore, 3/4 of the system cost is collector-related and its cost will vary from the base design inversely with chiller OCOP. For the chiller, it was assumed that the installation was 40% of the installed chiller cost, and the variable cost of all heat exchangers was approximately 60% of the chiller cost.

Variable costs of the heat exchanger components were assumed to be directly related to the fraction of the total heat exchange surface area employed. The heat exchange surface area of each component was calculated from the "U" estimate made under the Refrigerant Selection Matrix Study and the UA value determined for a base design unit with RANKØ6. These are reported in Table 1.2-12 along with the resulting variable cost fraction used.

TABLE 1.2-12

VARIABLE COST FRACTIONS FOR HEAT EXCHANGERS USED IN COST OPTIMIZATION

	<u>UA</u>	<u>U</u>	<u>A</u>	<u>Variable*</u> <u>Cost Fraction</u>
Boiler	1581	465	3.4	0.05
Preheater	114	19	6.0	0.08
Condenser	2106	126	16.7	0.23
Cooler	1340	269	5.0	0.07
Regenerator	139	20	7.0	0.10
LSHX	67	20	3.4	0.05
Total Cost Units			41.5	

* Variable cost fraction based on a total cost of 71.5 cost units, which assumes 41.5 cost units is about 58% of chiller cost.

The cost-performance optimization proceeded as follows: Starting with the boiler, individual computations were made for several boiling temperatures at the fixed supply condition. For each point, OCOP, LMED and required UA were computed. This resulted in a new boiler surface area and new total chiller cost, as well as a new collector system cost as changed by the OCOP. Figure 1.2-12 shows plots of resulting chiller package cost, collector system cost and total system cost as a function of OCOP, for variations of the heat exchanger sizes. As a first step, the boiler size was increased from the base design point of 273°F. The chiller package cost went up as did the OCOP, causing the collector cost to lower. The sum of these becomes the total system cost curve for the boiler which exhibits a definite minimum. This minimum boiler condition was used as the base for optimization of the next component, the condenser. It too shows a minimum value, which was used for the next component evaluation, etc. This process was repeated for all components. As a result of changing all components, it was found that the OCOP could be improved 21.4% while the "cost" was reduced 9%. Table 1.2-13 presents a detailed comparison of the initial and final designs.

Although the assumptions used in this study are debatable, it was felt that the trends were real, and that the values of "UA" found for the optimum design would be a good starting point for subsequent conceptual designs of the heat exchanger components. In addition, a method was demonstrated which could be used at a later date when more refined cost performance data become available to generate a new optimum design.

FIGURE 1.2-12

TABLE 1.2-13

RESULTS OF COMPONENT/SYSTEM OPTIMIZING FOR R-113

	Base Run	Optimization Run	
		Initial with Pressure Drops	Final
Run	405	444	455
Inlet Solar Water, °F	300	300	300
Boiling Temp., °F (sat.)	273	273	290
Boiler UA BH ⁻¹ F ⁻¹	594.6	593.9	1220
Approach (Pinch Point) °F	18.4	19.6	4.4
Inlet Condenser Air, °F	95	95	95
Condensing Temp. #1 °F (sat.)	120	123.0	120
Condensing Temp. #2 °F (sat.)	120	113.5	110.5
Condenser UA BH ⁻¹ F ⁻¹	1982.4	2106	2270
Approach #1 °F	16.9	10.2	7.4
Approach #2 °F	7.7	10.2	9.2
Leaving Chilled Water °F	45	45	45
Evaporating Temp. °F (sat.)	40	40	43.5
Evaporator UA BH ⁻¹ F ⁻¹	1339.5	1339	2502
Approach, °F	5.0	5.0	1.5
Liquid-Suction Effectiveness	0.7	0.8	0.9
Liquid-Suction UA BH ⁻¹ F ⁻¹	52.5	66.7	102.3
Liquid-Suction Approach °F	22.2	13.5	6.1
Regenerator Effectiveness	0.8	0.8	0.8
Regenerator UA BH ⁻¹ F ⁻¹	134.7	138.9	119.0
Regenerator Approach °F	12.5	11.7	11.7
Rankine Cycle Efficiency	0.153	0.148	0.161
Refrigeration Cycle COP	4.621	4.762	5.32
Overall COP	0.707	0.690	0.838