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DR#0454.5

DOE/CS/30576-T1
(DE84015132)

MODELING OF A SECOND-GENERATION SOLAR-DRIVEN
RANKINE AIR CONDITIONER

Final Report

By
M. W. Denius
W. D. Batton

July 1984

Work Performed Under Contract No. AC03-81CS30576

Barber-Nichols Engineering Company
Arvada, Colorado

Technical Information Center
Office of Scientific and Technical Information
United States Department of Energy

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BARBER-NICHOLS

ENGINEERING

DOE/CS/30576-T1
(DE84015132)

Distribution Category UC-59A

MODELING OF A SECOND-GENERATION
SOLAR-DRIVEN RANKINE AIR CONDITIONER

CONTRACT # DE-AC03-81CS30576

FINAL REPORT
July, 1984

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NOMENCLATURE LIST

BIC	Boiler in Collector Configuration
Cp	Specific Heat
D,d	Diameter
FY83	Fiscal Year 1983
h	Heat Transfer Coefficient
k	Thermal Conductivity
LCWT	Leaving Chilled Water Temperature
$\dot{M}$	Mass Flow Rate
Ns	Specific Speed
PMA	Permanent Magnet Alternator
Po	Inlet Nozzle Pressure
Pn	Prandtl Number
Pr	Pressure Ratio
Q	Heat Load
r	Radius
Re	Reynolds Number
SCR	Silicon Controlled Rectifier
2G	Second Generation Configuration
Tsi	Temperature Solar Fluid into Engine
Twb	Temperature Wet Bulb
U	Overall Heat Transfer Coefficient
U/Co	Velocity Ratio
W	Mass Flow Rate
Δ	Delta - Differential Between Two Points
η	Eta - Efficiency
μ	Mu - Viscosity
θ	Theta - Phase Angle or Temperature Depending On Context

1.0 EXECUTIVE SUMMARY

Ten configurations of a second-generation (2G), solar-powered, Rankine-driven air conditioner were simulated and the data presented to Honeywell International and Science Applications, Inc. for use in companion studies. The following ten configurations were simulated:

Baseline	- evaporatively cooled condenser
High temperature	- evaporatively cooled condenser
High temperature	- air-cooled condenser
High temperature	- cooling tower-cooled condenser
High temperature	- space heating application
High speed alternator	
10 & 30 HP scaling of Rankine engine	
Baseline configuration	- 45° chilled water
BNL low temperature Rankine cycle	
BIC configuration	- 1875 square feet of collector
	- 1250 square feet of collector
	- 2500 square feet of collector

The results of the analysis show that the boiling-in-collector (BIC) configuration generates more power per collector area than the other configurations. The models used to simulate the configuration are presented in this report. The generated data is also presented.

Experimental work was done under this study to both improve a novel refrigerant and oil lubrication system for the centrifugal compressor and investigate the aerodynamic unloading characteristics of the centrifugal compressor. The information generated was used to define possible turbo-gearbox configurations for use in the second generation computer simulation.

Further research work is needed to complete the definition and optimization of these systems and subsystems. Materials research, component/subsystem design and proof-of-concept are essential in two specific areas: 2G and BIC Rankine air conditioning systems.

2.0 INTRODUCTION AND OBJECTIVES

The objective of this work was to critically investigate various potential configurations for the second generation of the 25-ton solar Rankine air conditioner developed under the NASA 404 program. The goals were to improve performance, decrease parasitic losses, improve reliability and increase manufacturability.

2.1 BACKGROUND

In 1976 NASA awarded three contracts (designated the NASA 404 program) to develop solar heating and cooling systems. One of these contracts was awarded to a team composed of Honeywell (prime contractor), Barber-Nichols Engineering and Lennox. Because of the short duration before hardware was to be field tested, the team elected to use existing technology utilizing as much off-the-shelf hardware as possible. This very successful approach led to a total of five residential units (3-ton) and six light commercial units (25-ton) installed in the field at operational test sites.

While this approach led to a significant amount of operation test data providing proof of concept, the hardware was not optimum. The majority of the heat exchangers, air conditioners and control components were off-the-shelf. Only a few components (turbogearbox and regenerator) were fabricated for the project and these were existing oversized designs.

Both the residential and light commercial-sized machines used uniform thermodynamic cycles with refrigerant R-113 used as the working fluid in the Rankine engine and R-12 used as the air conditioning refrigerant. These fluids were selected based more on the availability of components and familiarity rather than optimum performance. The engines used heat from flat plate collectors at 200°F.

Early in the 404 program it was observed that a significant

advantage could be obtained by adding a generation mode to the machines. This is primarily beneficial in the spring and fall when there is adequate solar available to operate the engine but no cooling requirements exist. The generated power can be pumped into the grid and then recovered at some later date. Most of the solar air conditioners were manufactured with this generation capability which significantly improved the annual operation efficiency.

Another early observation was that the collector field was the driving cost for the entire system. If the field is sized based on cooling demand rather than heating, as is true for the vast majority of the United States, a more efficient engine will reduce the size of the collector field. The major increase in engine performance is determined by increased high-side (solar collector) temperatures. Changing collector type from a flat plate to a concentrating type, which typically increases cost, could be an overall benefit in going to higher temperature engines.

As a result of these observations, two machines were built to test these theories. One was a high temperature (300°F) engine driving one of the commercial-sized 25-ton chillers. This machine was just made operational (fall of 1982) at Marshall Space Flight Center in Huntsville. The second machine was one of the low temperature (200°F) engines, without the air conditioner, set up for electrical generation only. This machine began operating July 1981.

An additional, significant discovery was that the current state-of-the-art air conditioner compressor in the light commercial size was not very energy efficient. The mechanical measured efficiency of the 25-ton, reciprocating compressors used for the project was 0.65. This low efficiency makes the power requirement to produce 25 tons of cooling large, which requires more collectors than if a more efficient compressor were available. Therefore, one of the tasks in the 404 project was to develop a high efficiency 25-

ton compressor. This task resulted in a centrifugal compressor which was tested at 0.80 efficiency using R-11. The selection of R-11 was based on the possibility of using the same fluid in the low temperature Rankine engine and also the suitability and performance in the air conditioner.

In the fall of 1981 the 404 program was ended and the effort was transferred to DOE/SAN.

2.2 SECOND-GENERATION STUDY

During the last several years of the NASA 404 study, NASA was interested in redesigning the light commercial-sized hardware to reduce cost, increase manufacturability and increase performance. When the program was transferred to DOE/SAN, the next logical step was to begin consideration of this second-generation machine. However there were a number of alternative configurations to be considered before actual design could begin, therefore the current study was undertaken to select the best possible configuration. A number of variations in the selected configuration were then considered. Barber-Nichols' task was to simulate these variations and produce performance maps. A companion solar system (collectors, pumps, controls, etc.) was devised by Honeywell TSC and then the annual performance and system costs were determined by Science Application, Inc., of McLean, Virginia. Barber-Nichols, Honeywell and SAI were co-contractors under contract to DOE/SAN with LBL as the technical monitor.

In support of the cooling system simulations, some effort was expended in laboratory testing and component design studies. The laboratory testing was required to test concepts and to confirm operational status of the gearbox lubrication system. The component design support was necessary to provide realistic component performance values for the simulations.

2.3 ADVANCED SYSTEMS

As detailed above, the original goal of the study was to select a configuration for a second-generation machine that could be fabricated in FY83 and would represent the best current state-of-the-art hardware. Early in the study interest in more futuristic machines was expressed. This provided the opportunity to consider configurations that will provide more potential, but at higher risk, and will require some research before becoming feasible. As a result, the scope of work was changed to include several futuristic systems and delete several less promising variations of the baseline system. The configurations added were the boiling-in-collector (BIC) system, a configuration using low cost, plastic flat plate collectors (BNL) and a 45°F return water temperature chiller simulation for comparison purposes.

The boiling-in-collector (BIC) engine concept has great promise. The preheater and evaporator in the engine are replaced by the collector where the engine working fluid changes phase. The system costs are reduced due to the elimination of the preheater and boiler and the elimination of the collector loop pump. The performance is increased because the working fluid can be heated to the temperature to which the solar fluid would normally be heated. Several unresolved issues exist including questions on the collector heat transfer rates and potential problems associated with maintaining a leak-free collector system when the working fluid passes through the collector. Additionally, the collector piping system and collector tilt angle must be carefully considered. The BIC system could operate with flat plate collectors but most likely would be applied to trough-type concentrators due to the high achievable efficiencies.

The low temperature, low cost plastic flat plate collectors offer the potential of significantly lower cost solar systems. While performance penalties are paid because of the inefficiency of the collector and the lower engine operating temperature, the potential cost reduction could make the unit cost of power lower than a conventional flat plate collector system.

The baseline system uses a higher than standard leaving chilled water temperature (LCWT) (55°F) because of the significant performance benefit. This higher LCWT does require a new approach in terms of HVAC design. Higher air flow rates, larger coil areas and less strict dehumidification are required. Because of energy costs, the HVAC industry is already moving in this direction. In order to make comparisons with more conventional equipment, without benefit of this advantage, a chiller with LCWT of 45°F was simulated to evaluate the effect of this parameter change.

3.0 TEST PROGRAM

The following test program was conducted on the NASA 404 compressor test system built by Barber-Nichols for improving air conditioning compressor performance. There were two goals in this program. The first was to develop a more satisfactory internal lubrication system and the second was to analyze various methods of unloading the compressor.

3.1 LUBE SYSTEM IMPROVEMENT

During the compressor development program it was decided to lubricate and cool the compressor gearbox using a mixture of refrigerant oil and the refrigerant working fluid (R-11). This type of lubrication/cooling had been successfully pioneered on the Rankine engines developed by Barber-Nichols for the NASA 404 program. The system devised for the compressor was an improved version where the bulk of the oil remained in the gearbox system while only refrigerant, containing a trace of oil, would return to the refrigerant system. The lubrication system was evaluated during the compressor performance testing and the following problems were noted. The gearbox tended to distill the refrigerant leaving pure oil. This same effect took place in the lubricant tank. While these difficulties were considered minor and readily curable, there was not sufficient time available to make the necessary modifications. Therefore, in order to prove the concept, it was necessary in this program to analyze the system and make the required changes to produce a functional lubrication system.

The compressor gearbox system was disassembled and the following lubrication pump items were changed:

- 1) The pitot pump inlet probe (Figure 3-1) was enlarged to four times the original area.
- 2) The probe inlet was rotated from directly tangent to

UNLESS OTHERWISE SPECIFIED	ITEM NO.	QTY	PART NO.	DESCRIPTION	SPECIFICATION
INTERPRET PER AMS 1143-1073	1	1		THREADED PROJECTION	
ANGULAR DIMENSIONS ± 3°	2	1		APPROVED	DATE
SURFACE FINISHES 100/100	3	1		DATE	DATE
TOLERANCES UNLESS OTHERWISE SPECIFIED	4	1		DATE	DATE
1 PLACE 2 PLACE 3 PLACE	5	1		DATE	DATE
HEAT TREATMENT	6	1		DATE	DATE
REMOVE BURRS	7	1		DATE	DATE
REWORK	8	1		DATE	DATE
REWORK	9	1		DATE	DATE
REWORK	10	1		DATE	DATE
REWORK	11	1		DATE	DATE
REWORK	12	1		DATE	DATE
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the fluid flow to 15° from normal to the fluid flow to minimize windage back-pressure. The slight angling of the probe into the flow maintains automatic liquid ring level control in the pitot pump rotor.

The gearbox was reassembled and mounted on the test stand. A number of system changes were made on the test stand itself to improve lube system performance (Figure 3-2). A pitot pump discharge bypass line was added to allow recirculation of the reservoir oil coolant through the gearbox. Before this addition, coolant was boiling off in the gearbox with no means of replenishment. Eventually the gearbox lubricant would become pure oil without coolant, allowing the gearbox temperature to rise above acceptable limits. The excessive length of the pitot pump discharge line was shortened to reduce line pressure losses. Due to the increase of inlet probe area, the lube pump inlet line was also increased in size to reduce required suction head from the reservoir.

Two scavenging drains were included in the original design of the gearbox, one on each edge of the gear. The purpose of these drains was to allow quick scavenging of static oil level in the gearbox during startup. The two lines were joined together external to the gearbox and returned to the reservoir. It was noted during testing that flow from one side of the gear to the other was passing through these lines. This was remedied by blocking the drain on the suction side of the gear and connecting the scavenge drain on the pressure side of the gear to the bottom of the reservoir, thus providing a liquid trap to stop vapor flow. In the original system these scavenge lines were also the gearbox vent connection between the gearbox and the reservoir. A new vent line, which included a restriction to limit vapor flow, was added between the gearbox and the reservoir.

It was discovered during long-term testing that coolant was being purged to the refrigeration side of the system and not being returned by the small coolant supply pump in the bottom of the

FIGURE 3-2

* LOCATE LUBRICANT TANK CLOSE TO GEAR BOX.
BOTTOM OF TANK SHOULD BE 4.62\"/>

DATE	APPROVED

TEMPERATURE MEASUREMENTS

NO	TYPE	RANGE °F	Attachment	Note
T210	Type J T/C	40-160	Bare jet in gearbox	Use swagelock comb. with P210
T220	Type J T/C	40-180	Jet sealed to bearing support	
T230	Type J T/C	40-160	Jet sealed to bearing support	
T240	Type J T/C	40-180	Solder to tube	Wrap around 2 turns

PRESSURE MEASUREMENTS

NUMBER	TYPE	RANGE	NOTE
P210	Bourdon	-30 in. to 60 psia	*R11-Zephron mixture, see H. John for marked print
P220	Bourdon	-30 in. to 60 psia	*R11-Zephron mixture
P230	Bourdon	-30 in. to 60 psia	*R11-Zephron mixture
P240	Bourdon	-30 in. to 60 psia	*R11-Zephron mixture

*Use Robinair fittings, use Weisker Cat. No. B-1 6\"/>

LISTING OF COMPONENTS TO BE PURCHASED FOR LUBRICATION CIRCUIT

- Two each Alco solenoid valve S120C 1/2\"/>
- One each Sporlan catch-all C-052-S
- One each see-all Sa-155 ODF solder
- Three each Fluid Products Co. level control switch LCS-M-100-B
- Two each solid state relay to be made up by Barber-Nichols
- Two each Henry liquid level gauge, Cat. No. 5024 (one of these for receiver)
- One each Curson model 20-02-PX0 micromerger filter combination with metal bowls (this number came from CAT 76 pp 61-62 which is out of date). Also order replacement element for each part of filter MRP-15-410 and MRP-15-538.
- One each McCombs Thermal Company, Inc. liquid receiver, horizontal, model no. 368.
- One each Grimes Cat. Walworth 125 B swing check-regrid type. Fig. 609 solder joint ends, 1/2\"/>
- One each Graco dual pump HP26-64

TEMP NO'S.
 (1) P210, P220, P230, P240
 (12) T210, T220, T230, T240

GENERAL NOTES, UNLESS OTHERWISE INDICATED

NOTES:
 1. ALL DIMENSIONS ARE IN INCHES
 2. UNLESS OTHERWISE SPECIFIED
 3. TOLERANCES
 4. XX = ±
 5. XXX = ±
 6. XXXX = ±
 7. ALL SURFACES
 8. MATERIAL

QTY		ITEM NO.		PART NO.		DESCRIPTION		SPECIFICATION	
DIMENSIONING						ALL - STD - 1 COMMERCIAL			
UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES						APPROVED		DATE	
TOLERANCES						CHIEF		b n	
XX = ±						PRODUCT		b n	
XXX = ±						DATE		b n	
XXXX = ±						HWS		b n	
1/2 =						TEST		b n	
ALL SURFACES						CHECKED		b n	
MATERIAL						DRAWN		b n	
NEXT ASSY						WT		b n	
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evaporator. This pump was replaced with a pump having a higher head capacity and better NPSH characteristics which solved the problem.

The system presently is operating satisfactorily, unattended and with sufficient reliability to be considered a production system. This same technology is being used in the baseline, second-generation gearbox.

3.2 AERODYNAMIC UNLOADING

One of the significant concerns in designing a hermetic turbocompressor with motor/generator is the parasitic loss due to windage when either the compressor wheel or turbine wheel is unused. This windage loss must be reduced to an insignificant value in order to make the design viable. In the various modes of operation described in Section 4.0, there are times when either the turbine or the compressor is not being used but is rotating. Since the turbine wheel will act as a compressor during the unloaded mode of operation, the windage could be measured on the compressor test facility.

The preferred method of unloading is to reduce the pumping action of the wheel as opposed to mechanically disconnecting the wheel. Two possible means of reducing the pumping action are: closing off the compressor inlet valve, leaving just enough flow to cool the compressor wheel; and sealing off the inlet, discharge and shaft, and evacuating the wheel chamber.

The compressor test facility was retrofitted to test the two means of unloading. A recirculating bypass line and a flow control valve were added to the test system to allow compressor operation at very low flow rates without surge. Tests were then conducted with reduced inlet flow by incrementally closing the compressor inlet valve. Data was taken both with and without bypass flow. No definite surge line was detected with this system. The discharge line was then blanked off and operated with no

through flow while evacuating the compressor chamber with a vacuum pump.

The results of these tests are shown in Figure 3-3 where the variation of power input with inlet pressure to the compressor wheel is shown. At an inlet pressure of 5.3 psia, the maximum pressure ratio capability of the compressor wheel combined with the minimum flow required to maintain a reasonable wheel temperature resulted in a minimum power input of about 6 hp. At inlet pressures larger than 5.3 psia, the power input increases until the inlet pressure equals the design evaporator pressure of 10.2 psia where the compressor wheel consumes the design point power output of 20 hp.

Because the compressor wheel is limited by design on maximum pressure ratio allowable, the inlet pressure will not drop below 5.3 psia if the discharge pressure is the standard condenser pressure. The only way the inlet pressure and, hence, the fluid density and power loss can be reduced is to seal off both compressor inlet and discharge and evacuate the chamber. The result of this test is the data shown (Figure 3-3) for inlet pressures below 5.3 psia. The power consumed by the wheel drops below 1 hp as the pressure drops.

The results of these unloading tests show that the idea of simply closing the compressor inlet valve does not produce low enough losses. On the other hand, closing off the inlet and outlet to the compressor while evacuating the chamber does reduce the parasitic losses to acceptable values. Thus a means of unloading the wheels has been demonstrated. The practicality of this method must be further scrutinized.

BARBER-NICHOLS

ENGINEERING

FIG. 3-3

INLET PRESSURE
VS.
WHEEL HORSEPOWER

4.0 COOLING MODULE SIMULATIONS

The second major task in this program was to simulate the cooling module operation for the baseline configuration and for a number of variations. These simulations resulted in performance maps that were combined with solar systems conceived by Honeywell TSC. The combined systems were then applied to a particular building model and operation simulated over a year for several geographic locations. This annual simulation work was done by Science Applications, Inc. (SAI).

The cooling module simulation models developed as three separate parts: the Rankine engine; the air conditioner; and the condenser. This evolved out of the three possible modes of operation: as a solar-assisted air conditioner; as a solar-driven power-generation unit; and as an electric-driven air conditioner. System operation in the three modes is described below.

During the times that adequate solar input is available to drive the Rankine engine and there is a call for cooling, both the Rankine engine and the air conditioner will be on. The turbine and compressor wheels are on the same shaft and are directly coupled to the induction motor/generator. Depending upon the amount of solar energy available and upon the amount of cooling required, the power delivered by the turbine may be less than, or greater than the power required by the compressor. Only by chance would they be equal. If there is insufficient power to drive the compressor, the shaft will begin to slow down so that the electric motor speed becomes less than synchronous. Induction motors deliver increasing amounts of power as their "slip" speed (synchronous less actual) becomes greater. Therefore, a balance will be reached where the combined power delivered by the turbine and electric motor will equal the compressor requirements. In this mode the condenser receives hot fluid from both the turbine and the compressor. This is the condition for which the condenser was designed.

If there is sufficient solar input to operate the turbine but no call for cooling (such as a typical spring or fall day, the Rankine engine will operate without the air conditioner. As described in the section on testing (3.0) and later in Section 4, the compressor wheel can be unloaded so it consumes only a minimal amount of power while rotating with the turbine. Electric motor speed increases because it is the only load on the turbine. As the motor increases its speed above synchronous, it begins to generate electricity. This is described in more detail in Section 4.1.10. A balance is reached when the power used to generate electricity is equaled by the turbine power delivered. In this mode the condenser needs only to reject heat received from the engine. Therefore, the condenser is oversized for this mode of operation.

The third mode of operation occurs when air conditioning is required and no solar is available. In this situation the compressor is operating and the turbine is unloaded. The electric motor delivers power to drive the compressor. The condenser rejects heat from the air conditioner cycle only, which is much less than the design load for the condenser.

The three separate models for the engine, air conditioner and condenser were combined to get the cooling module subsystem performance. The use of separate models made it easy to make changes in one model while keeping the others fixed and also reduced computation time on the computer.

The configuration for the cooling module subsystem that most nearly represented the improved version of the original NASA 404 cooling subsystem was chosen as a baseline. From this baseline a total of nine variations were explored. A change in program direction caused some of these variations to be dropped and several new options added. This generally broadened the scope of interest and extended the work to include more futuristic machines.

The three separate models for the engine, air conditioner,

and condenser were built around component performance based on past field experience and on component design analysis carried out specifically for this study. These component analyses were used only in instances where actual data did not exist or did not cover a sufficient range. The models thus generated were simulated on a digital computer and performance maps generated. These performance maps were used by Honeywell and SAI to obtain annual operating performance for comparison. The best performing system could then be chosen.

4.1 CONFIGURATIONS

The scope of work suggested nine configurations for exploration. As a result of the preliminary analysis and a change of scope, three of the nine configurations received no extended development. Table 4.1 summarizes the status of the nine original configurations. Four additional configurations were added in place of the configurations that were not modeled.

TABLE 4.1
MODELED CONFIGURATIONS

- 1) Baseline, evaporatively cooled condenser
- 2) High temperature, evaporatively cooled condenser
- 3) High temperature, air-cooled condenser
- 4) High temperature, cooling tower condenser
- 5) High temperature, space heating application (co-generation)
- 6) High speed alternator.

CONFIGURATIONS NOT MODELED

- 7) Variable area turbine nozzle
- 8) Turbo-driven feed pump
- 9) Power factor connection device

Following is a discussion and description of the modeled and the unmodeled configurations.

4.1.1 Baseline Configuration

The baseline configuration was envisioned as a system that could be built with existing technology as a logical extension of the existing units operating in the field. The system was to include improvements that had been suggested by the NASA 404 program and the NASA 404 component improvement program. The baseline configuration is diagrammed in Figure 4-1.

The following design criteria were specified for the baseline system:

- Refrigerant R-11 working fluid for Rankine engine and air conditioner.
- NASA 404 follow-on centrifugal compressor.
- Evaporative condenser.
- 55°F cooling water to the building load.

As indicated above, a design criterion for the baseline system was that the Rankine engine and the air conditioning module use the same refrigerant working fluid. The use of a common working fluid would allow the construction of a hermetic Rankine turbine and air conditioning compressor assembly that would be free from sealing problems. As a consequence of using a common working fluid, a condenser common to the Rankine engine and the air conditioning cycle was envisioned for the system. The compressor tested under the NASA 404 component improvement program used refrigerant R-11 for a working fluid. Therefore, R-11 was used for the working fluid of the Rankine engine and the air conditioner.

The module used an evaporative condenser with high efficiency motors and a fan speed reduced from the value specified for the condenser. The reduced fan speed considerably decreased the parasitic power requirement of the condenser fan. The trade-off with lower parasitic power is the use of a larger condenser heat exchanger area.

The system was designed for a solar fluid design inlet

FIG 4-1
BASELINE SYSTEM

temperature to the Rankine evaporator of 200°F. The evaporative condenser was designed for a wet bulb temperature of approximately 75°F. The evaporator of the air conditioning cycle was to provide leaving water at 55°F for cooling the building. The cycle was simulated and used as a basis (baseline) for comparison with the other configurations that were thus generated.

4.1.2 High Temperature - Evaporative Condenser

The recent advancements in line-focusing concentrating solar collectors suggest that a Rankine cycle could be designed for a higher temperature source than flat plate collectors. The Carnot efficiency of the Rankine engine can be increased if the difference in temperature between the hot source and the cold sink can be increased. The actual efficiency of the Rankine engine will track the Carnot efficiency. The high temperature configuration was similar to the baseline configuration with the primary difference being the solar fluid inlet temperature. The higher temperature system required a change in the turbine configuration for optimization of the design performance because the pressure ratios across the turbine are larger with higher temperatures. The components of this module, as in all the configurations, were designed to optimize the module's performance. The configuration was simulated and the data is presented in the results section (5.0) and the Appendix.

4.1.3 High Temperature - Air-Cooled Condenser

An analysis was made of the high temperature configuration with an air-cooled condenser. The current standard industry practice is to use air cooling and therefore this analysis was considered useful for comparison purposes. The temperature of the working fluid in the air-cooled condenser was higher than the fluid temperature in the evaporatively cooled condenser. The higher sink temperature lowered the efficiency of this engine compared with the high temperature evaporatively cooled configuration. The temperature of the condenser required that the air conditioning model be rerun and new data entered into the analysis. The system

was designed for a solar fluid design inlet temperature to the Rankine evaporator of 300°F . The condenser was air cooled with a design dry bulb temperature of 95°F . The configuration was modeled and the data presented in the Appendix.

4.1.4 High Temperature - Cooling Tower Condenser

The cooling tower heat rejection system was simulated in order to compare these second-generation machines with the NASA 404 machines which use a cooling tower to reject heat from the water-cooled condenser. The condenser temperatures obtained with a cooling tower are an intermediate value lying between the evaporatively cooled condenser and the air-cooled condenser. The efficiency of this Rankine engine, therefore, lies between the values for the evaporatively cooled condenser and the air-cooled condenser. The system was designed for a solar fluid inlet temperature to the Rankine evaporator of 300°F . The condenser was cooled by a cooling tower with a design ambient wet bulb temperature of 75°F . The configuration was modeled and the data presented in the Appendix.

4.1.5 High Temperature - Space Heating

If the source were at 300°F and the sink were at 100°F , then the Carnot efficiency would be 26%. If an engine were 50% of Carnot, a reasonable value, the engine efficiency would be 13%. A large percentage (over 80%) of the collected heat would then be rejected in the condenser assembly. If the rejected energy were of a high enough grade then space heating with this energy would be desirable (co-generation). A control philosophy was developed for using the rejected heat for space heating via a fan coil and alternate cooling tower configuration. The configuration was modeled and the data is presented in the Appendix.

4.1.6 High Speed Alternator

The previous configurations used a synchronous-speed motor/generator connected through a gearbox to the high speed turbine compressor shaft. The associated losses in the gearbox amounted

to 1.1 hp. This parasitic power could be reduced to 0.1 hp if a high speed motor/alternator were used in place of the motor/generator. Since the high speed motor/alternator operated at frequencies much higher than the standard 60 Hz, signal-conditioning equipment in the form of rectifiers and inverters were required. This type of hardware, using permanent magnet alternators (PMA) and solid-state signal-converting devices, is becoming very efficient. Data from the baseline configuration was collated and operated on to reflect the 1.0 hp change in parasitic power. The data is presented in the Appendix.

4.2 CONFIGURATIONS NOT MODELED

The following four configurations were not modeled as the preliminary analysis showed that the data generated would be of limited value.

4.2.1 Variable Area Turbine Nozzles

Variable area turbine nozzles are used primarily in radial inflow turbines to regulate the flow through the turbine by changing the nozzle area rather than by changing the upstream pressure (throttling). This variability of the nozzle area affects the turbine performance during off-design operation. The variable area nozzle was investigated but not simulated. The computations would be very lengthy and the potential gain over fixed nozzles is only marginal as described below.

The analysis of a variable area nozzle turbine is much like the analysis of a fixed nozzle turbine, except that every change in nozzle area effectively creates a new turbine performance map. The simulation, if carried out, would consider a variable nozzle turbine to be an infinite series of fixed nozzle turbines, each with a different nozzle throat area. Actually, a finite number of nozzle areas would be chosen from the entire range available so that smooth performance curves could be generated as a function of nozzle area. Each nozzle area chosen in this process would be used to generate a range of system performances, much like the

analysis used in this report for a fixed nozzle turbine. The engine modeling procedure would then be repeated for each new nozzle area until an entire operating map, including the effects of nozzle area variation, was generated. This would multiply the complexity of the analysis by as many different nozzle areas as required to obtain smooth performance curves. The calculations for a single nozzle area are very time consuming and this time would be multiplied by as many different nozzle areas as chosen. From the standpoint of practicality, studying the effects of variable area nozzles raises the solution time to solve the problem by an order of magnitude.

The benefits of the variable area nozzles are somewhat limited. The design point performance is unaffected as it is only the off-design performance that changes. The turbine uses the variability of the nozzles to control the fluid flow, eliminating the need for a throttling valve. The further off-design the system is operating, the more throttling is required. Therefore, it is only very far from the design point that significant amounts of throttling are avoided with variable area nozzles. At these low-power off-design points, the increase in efficiency due to variable nozzles produces a small improvement in power output. At the same time, the complexity of the operating system is increased because of the added turbine hardware. Since the improvement in turbine efficiency occurs only at low power points, the power-weighted net gain is small.

The use of variable area nozzles is currently limited to rectangular, converging-only designs because of the complex hardware required. They are most commonly used for radial inflow turbines, although they can be applied to axial turbines with much more complicated hardware. If the turbine requires converging-diverging nozzles for supersonic flow, variability is not possible. We can, therefore, conclude that variable area nozzles are of limited use and are not included in these simulations.

4.2.2 Turbopump

The feed pump for the Rankine engine has been envisioned as

being an electric-driven unit, with the electricity being generated by the turbine/generator pair or coming from outside sources. An alternative to this is the design of a separate turbopump unit in which a special turbine would be dedicated to driving the feed pump. This unit would be designed with its own shaft and bearings, making it totally independent of the power turbine and gearbox. The turbopump would also require its own controls.

The worst condition for the pump is the high fluid flow, high condenser pressure point, with a volume flow of 11.97 gpm and a required head of 128 feet. At this point, using the enthalpy available to the turbine and estimating the required turbine output power, a turbine mass flow was calculated. Using this and an assumed value of turbine specific speed, $N_s = 15$, a turbine shaft speed of 29,600 rpm was calculated. Since the turbine drives the pump directly, this speed can be used to calculate the pump specific speed, which is 127. The pump efficiency for this specific speed is about 0.80 and the size of the impeller would be 0.87 inches in diameter. A suction head was calculated based on a suction specific speed of 7500 to be 32.6 feet (about 21 psi), or about one-fourth of the feed pump head requirement. This is a very large required suction head, making the pump impractical.

There are no attractive alternatives, since reduction of the shaft speed below 29,600 rpm quickly reduces turbine specific speed and efficiency, while keeping the speed at 29,600 rpm means the pump design is impractical. If one considered the alternative of operating the shaft at a lower speed, say 10,000 rpm, the turbine specific speed would be 5, giving peak efficiencies of 0.55. The pump specific speed would be about 43, giving a peak efficiency of 0.70. The impeller diameter would be about two inches. The required suction head for this configuration would be 7.7 feet or 5 psi, a more reasonable number. Assuming bearing losses of 0.3 hp, the input power to the turbopump unit would be 2.0 hp. For the electrically driven pump, the input power to the turbine to operate the feed pump would be 2.5 hp (assuming the motor efficiency were 80% and the generator efficiency were 85%). This is

a small amount of power saved, considering the added complexity of the system which uses the turbopump. The turbopump would take time and money to design and develop, which would increase the cost of the Rankine system using it. A more productive alternative would be to drive the feed pump off the output shaft of the gearbox.

4.2.3 Power-Factor Controllers

This machine uses an induction motor which sometimes functions as an induction generator. A solid-state device used to control the power factor of induction motors was thought to have a potential benefit in increasing the useful power delivered in the generate mode. This was investigated and is discussed in detail below.

An induction generator is an induction motor driven above its synchronous speed by a suitable prime mover. This results in the production of electric energy. To see why this works, consider how a motor functions. Torque to turn the rotor is produced by the rotating magnetic field generated in the motor's stationary windings. This torque is produced because of the slight lag or slip in rotor speed compared to the synchronous speed of the rotating field. The relative motion reduces voltages in the squirrel cage bar conductors. These, in turn, circulate currents through the bars, generating their own magnetic fields, which interact with the stator field to produce torque. An increase in mechanical power demanded from the motor shaft slows the rotor down further, increasing the slip to boost the available torque.

If the same machine were driven by a turbine above synchronous speed, the direction of magnetic field interactions and induced rotor currents would be reversed. The mechanical power input to the shaft would be converted into electrical power output, which would flow into the connected electrical system. The faster the machine would be driven, the more power would be available (see Figure 4-2).

FIG 4-2

TYPICAL INDUCTION MOTOR-GENERATOR
CHARACTERISTICS

Since an induction generator, like an induction motor, consumes rather than supplies magnetizing kilovars, the connected power system must be capable of supplying these kilovars at a lagging power factor to excite the generator by supplying its rotating magnetic field. Unlike a synchronous generator, an induction generator cannot be used for isolated operation.

From Figure 4-2 it can be seen that as an induction motor or generator approaches synchronous speed, the power factor drops to zero as the motor or generator load drops to zero, but the current drops to about one-third of its full load rating. This is the current required to excite the motor or generator. Power factors as low as 0.1 or 0.2 can exist when such motors or generators are partially loaded or unloaded. When this is the case, relatively large currents flow and little work is being performed and I^2R losses will occur at all points in the distribution system, including the motor windings, even though no mechanical power is delivered.

An electronic power-factor controller has been developed by NASA which is capable of raising the power factor from 0.2 to 0.8 for a partially loaded or unloaded motor. The following is a general discussion of the power-factor controller and its application to induction motors.

The reactive volt-amps of an induction motor are high when the motor is unloaded or partially loaded. Since the current remains high in the unloaded motor, the phase angle between voltage and current shifts with load. Typically, the current may lag the voltage 80° in an unloaded motor and 30° when loaded. The power-factor control circuit continuously monitors the phase angle between the voltage and current, and produces a voltage proportional to the phase angle. This voltage is summed with a fixed reference voltage which is indicative of a desired phase angle. The difference of the two produces an error signal which biases a ramp voltage that is synchronized with the 60 Hz line voltage. The intersection of the ramp and the error voltages are detected

by a squaring amplifier whose output provides the time for turning on a triac (or SCR) in the motor line. Thus, the "ON" time of the triac varies with the load and varies the voltage to force the phase angle to remain at the commanded value. With the system in control, typical motor voltage and current waveforms are as shown in Timing Diagram A (Figure 4-3).

The phase angle, shown as " θ " in Timing Diagram A, is measured by detecting the time between the zero crossings of the voltage and the zero value of the trailing edge of the current. With the circuit in control of the current, voltage is applied to the motor for a portion of each positive and each negative half-cycle of line voltage by means of the solid-state switch (triac) as shown in Timing Diagram A. It would be desirable to vary the motor voltage sinusoidally, as a linear amplifier or changing taps on a transformer might do. This, however, would be extremely complex and expensive, and the slight improvement in performance over the simple triac switching method could not be justified.

When the triac switches on, rapid rise of the current is prohibited by the inductance of the windings. The current rises, reaches a peak, and then follows the voltage down as it approaches zero, but with a finite lag. Although the firing voltage to the triac goes to zero when the line voltage goes through zero, the triac inherently will remain on until the current goes to zero. This is indicated by the motor voltage wave form.

The phase lag between voltage and current is indicated by " θ " in Timing Diagram A. This is the parameter which is to be measured and controlled. The line voltage and its inverse are squared by squaring amplifiers as indicated by E and $\bar{E}$ in Timing Diagram A. Each current pulse is squared by similar amplifiers as indicated by I and I' . By "AND-ing" $\bar{E}$ with I, and I with I' and then "OR-ing" the two, a pulse train is produced which has a pulse width proportional to the phase angle between voltage and current. When acted on by a low-pass filter, the average value of this voltage will be proportional to the phase angle. This

TIMING DIAGRAM A

Reference:
 NASA brief
 #MFS-23280

Figure 4-3,

voltage is summed with a command voltage which is indicative of a desired phase angle. The difference of the two is the system error voltage. This error is compared with a ramp which is synchronized with the zero crossings of the line voltage. The intersection of the sloped portion of the ramp and the error voltage is detected by a comparator and forms the turn-on pulse for the triac.

As the load on the motor is decreased, the slight change in phase angle causes the error to drop and intersect the ramp at a lower point. This moves the firing pulse to the right, along the sine wave, causing the triac to turn on for a shorter duration, lowering the applied voltage. Conversely, an increase in load will cause the firing angle to move to the left and apply more voltage to the motor. Thus, a phase angle is commanded and the high gain of the feedback loop will vary the applied voltage to force the motor to operate at the desired phase angle regardless of load. Since the current is never higher than that required for a given load, motor losses are minimized.

For an induction motor the current lags the voltage, (lagging power factor), while for an induction generator the current leads the voltage (leading power factor). As the induction motor load decreases when the turbine starts, the power factor will decrease, causing the power-factor controller to start controlling by reducing the motor voltage.

The firing pulse in Timing Diagram A will move to the right as the power factor in the motor decreases. When the power factor drops to zero at synchronous speed, the firing pulses will stop. As the turbine starts to drive the induction motor as an induction generator, the power factor becomes leading. For the firing pulses to start again, the generator power factor must increase to the point that the firing pulses can be positive when the motor current is zero. If Timing Diagram A were redrawn for a leading power factor, this would occur when the pulse angle would be approximately 45° . Since the triac would not be firing during

this period, the generator will be unloaded and the turbine will overspeed. To keep this from happening, the power-factor controller should be bypassed when the lagging power factor drops to approximately 0.1.

An additional consideration, beyond the technical problems stated above, is that the leading power factor of the induction generator can be an advantage. Most motors or other electrical devices have lagging power factors which result in lower efficiency and can result in higher power costs if the utility considers the VARS in billing. The leading power factor developed by the induction generator tends to correct the lagging power factor due to other components. Therefore, the power-factor controller does not make technical sense for the induction generator and is not even desirable. It is a useful device for the motor function when the motor is nearly unloaded, such as when the solar input is highest and air conditioning is required. Generally, however, the 25-ton machines are not normally connected to utilities where charges are made for poor power factors.

4.3 CONFIGURATIONS ADDED

During the course of the study, several additional systems were proposed for consideration. The additional systems examined were a 10 hp Rankine engine, a 30 hp Rankine engine, a baseline system with a 45°F cooling water return to the building, a boiling-in-collector cycle (BIC) and a low temperature source cycle with a nominal solar fluid inlet temperature of 170°F.

4.3.1 10 HP and 30 HP Rankine Cycles

The Rankine cycle data generated for the 20 hp baseline system was adjusted to reflect an engine with a 10 hp output and an engine with a 30 hp output requirement. Condensers, pumps, heat exchangers and parasitics were resized and reanalyzed to reflect the 10 hp and the 30 hp design points. The data for the 10 hp and 30 hp Rankine engines was combined with the baseline air

conditioning data to produce the results presented in the Results section and the data presented in the Appendix.

4.3.2 45° Chiller Water

The air conditioning cycle was rerun to reflect a 45° evaporator leaving water temperature. The baseline air conditioning cycle model had used an evaporator leaving water temperature of 55°F to the building and the 45°F cycle was added for comparison with more conventional machines. As the 45°F chilled water temperature along with the 95°F condenser temperature determine the pressure ratio across the centrifugal compressor, the compressor model had to be adjusted to accept this new pressure ratio and the altered design point. The air conditioning data generated was combined with a baseline Rankine engine to produce the data reported in the Appendix. This cycle was generated on request for use in SAI's Active Program Research Requirements (APRR) study for comparison with conventional systems.

4.3.3 Boiling-In-Collector

This model eliminates the Rankine evaporator and preheater since the refrigerant changes phase in the collector. Also, the collector pump is eliminated. This system will potentially have higher performance at lower cost than the other systems in this study. Data from the high temperature evaporatively cooled system was combined with a concentrating collector model to determine the boiling-in-collector data. The collector characteristics used were supplied by SAI. This model was run for several ambient temperatures and collector areas to provide sufficient data for SAI to complete their simulations.

4.3.4 170°F BNL Cycle

A design temperature for the Rankine engine lower than the baseline value of 200°F was required for use with the low cost, plastic Brookhaven National Lab (BNL) collector system. New heat

exchangers, evaporative condensers and pumps were specified for the lower source temperature. The 170°F Rankine source temperature lowered the Rankine cycle efficiency to 6.3% compared to the baseline value of 9.3%. To obtain the required 20 hp output of the Rankine engine, the mass flow rate of the Rankine system was increased to a value of 9830 lbm/hr as compared to a design value of 6474 lbm/hr for the baseline system. The mass flow change, along with the reduced pressure ratio across the turbine which was the result of the reduced 170°F source temperature, required the turbine model be adjusted. The cycle was modeled and the data presented.

4.4 SIMULATION DEVELOPMENT

In order to obtain the best possible performance for a given design point, the components that comprise the Rankine engine and air conditioning module were designed to optimize their respective efficiencies. To predict the off-design performance of the Rankine engine and air conditioning module, the off-design characteristics of the components that made up the module had to be modeled. The individual components that were modeled included turbines, compressors, pumps, motors and heat exchangers. This section of the report will discuss these models and their organization into a process flow diagram.

4.4.1 Turbine Model

The turbine and nozzle assembly were modeled using design performance prediction programs. The results of the analysis were then curve fit and entered in equation form into the off-design modeling simulation. The wide range of conditions for the Rankine engine required that two different types of turbines be specified for optimum performance. A radial-type turbine with converging nozzles was specified for the low temperature cycles and an axial-type turbine was specified for the high temperature cycles.

4.4.1.1 Radial Model

The turbine and nozzle characteristics were based on the parameters of inlet pressure to the nozzles (P_0), pressure ratio across the nozzle and turbine assembly (P_r), and the turbine velocity ratio (U/C_0). Variations in these parameters produce coupled variations in turbine efficiency and mass flow factor. Figure 4-4 illustrates the turbine efficiency, as a function of velocity ratio and pressure ratio, for a specific wheel design.

The turbine efficiency can be seen to fall from its peak design value with either an increase or a decrease in pressure ratio from the design value. The efficiency will also fall from its peak design value with either an increase or a decrease of velocity ratio from the design value. The peaking characteristic of the turbine efficiency curve illustrates the importance of choosing a design point for the Rankine engine components that corresponds to the nominal operating point.

The flow rate to the machine was modeled by computing the flow factor. The flow factor is defined as the ratio of the mass flow rate to the choked mass flow rate resulting from the same nozzle inlet conditions. As the pressure ratio across the turbine changes, the pressure ratio across the nozzles changes and the nozzles can become unchoked. The flow factor, as a function of pressure ratio and velocity ratio, is shown in Figure 4-5.

4.4.1.2 Axial Model

The axial turbine characteristics were based on the pressure ratio (P_r) and the velocity ratio (U/C_0). The turbine efficiency for a specific axial wheel is shown in Figure 4-6. This graph shows the same peaking of efficiency tendency as the radial wheel graph.

The flow rate for an impulse axial turbine and convergent nozzle assembly can be completely determined by the inlet pressure

FIG 4-4

RADIAL INFLOW TURBINE EFFICIENCY VARIATION WITH
VELOCITY RATIO AND PRESSURE RATIO.

FIG.4-5

RADIAL INFLOW TURBINE FLOW FACTOR
 AS A FUNCTION OF PRESSURE AND VELOCITY
 RATIOS

FIG.4-6

TURBINE EFFICIENCY VS. U/C_0 FOR LINES OF CONSTANT PRESSURE RATIO. AXIAL FLOW TURBINE.

to the nozzle block. The mass flow is always choked over the operating region of the axial machine, therefore the flow factor for an axial machine is always 1.0.

4.4.2 Compressor Model

The centrifugal compressor performance was modeled using data from experimental testing of the compressor developed under the NASA 404 component improvement program. The pressure ratio across the compressor determines the mass flow relationship and the compressor efficiency. Figure 4-7 presents the graph of measured mass flow rate versus pressure ratio. The measured compressor efficiency versus pressure ratio is shown in Figure 4-8.

4.4.3 Pressure Drops

The model used for pressure drop calculations was a simple extension of Fanning's proportionality relation between mass flow rate and pressure drop.

$$\Delta P = \left(\frac{\dot{m}}{\dot{m}_{\text{design}}} \right)^2 \Delta P_{\text{des}}$$

The assumed, experimentally based, component pressure drops were scaled with mass flow rate to determine their values in the off-design mode.

For the low temperature system, the feed pump head was known. This allowed the pressure drop across the flow control valve to be a calculated value. The high temperature system pump head was not known and the control valve pressure drop was simply proportioned along with the other pressure drops in the system using the above proportionality. Experience with similar systems provided the values used for the design values of the designated pressure drops.

FIG 4-7

COMPRESSOR RATIO VS. CORRECTED MASS FLOW

FIG 4-8
EFFICIENCY VS. PRESSURE RATIO OF COMPRESSOR

4.4.4 Heat Exchanger Model

The heat exchangers that were modeled included the preheater and the evaporator of the Rankine engine, the air conditioning evaporator, and the common condenser between the Rankine engine and the air conditioner. In the following sections, the models used for the components will be discussed. Included in the preheater and evaporator section is a discussion of the boiling-in-collector configuration.

4.4.4.1 Preheater and Evaporator

Most of the configurations modeled used separate heat exchangers for the preheater and evaporator. A standard heat exchanger curve is shown in Figure 4-9. The hot solar collector fluid enters the evaporator and transfers heat to the working fluid refrigerant via a boiling heat transfer mechanism. The solar collector fluid then enters the preheater and transfers the remaining heat input required in the Rankine engine.

The following assumptions were used in the preheater/evaporator modeling:

- 1) Collector loop mass flow rate remained constant.
- 2) Collector loop specific heat remained constant.
- 3) The evaporative heat transfer coefficient remained constant.
- 4) The heat leak to the environment was considered negligible.
- 5) Superheat configurations were not used.

The collector loop mass flow rate is supplied by a constant-speed commercial pump. Since the pressure drop in the collector loop does not vary and the speed of the pump does not vary, the mass flow rate in the collector loop remains at a constant value. These conditions support assumption number two.

FIG. 4-9HEATING CURVE

- T_{SI} SOLAR COLLECTOR FLUID INLET TEMPERATURE
- T_{S0} SOLAR COLLECTOR FLUID EXIT TEMPERATURE
- T_1 RANKINE EVAPORATOR TEMPERATURE
- T_2 RANKINE PREHEATER INLET TEMPERATURE

The collector loop fluid envisioned for use with the heat exchangers was a mixture of ethylene glycol and water. According to values found in ASHRAE¹ for a 30% by weight fraction of ethylene glycol, the specific heat varies from 0.92 Btu/lb-°F at 125°F to 0.95 Btu/lb-°F at 225°F. For the high temperature applications, the value of specific heat varies from 0.95 Btu/lb-°F at 225°F to 0.98 Btu/lb-°F at 300°F. The variation in solar collector fluid for a given operating condition is 20°F. The percentage variation for the low temperature specific heat and high temperature specific heat equals:

$$\begin{aligned}\text{Low Temperature: } \% \text{ var.} &= (100\%) (20/100) (0.95 - 0.92)/(0.92) \\ &= 0.7\%\end{aligned}$$

$$\begin{aligned}\text{High Temperature: } \% \text{ var.} &= (100\%) (20/100) (0.98 - 0.95)/(0.95) \\ &= 0.6\%\end{aligned}$$

The small variation stated in this parameter supports assumption number two.

The evaporative heat transfer coefficient was kept at a constant value for both the Rankine engine evaporator and the air conditioning evaporator. To support this assumption, the following brief set of calculations is presented.

From McAdams' equation for heat flow in a tube:

$$h = \frac{k}{D} (.023) \text{Re}^{.8} \text{Pr}^{.3}$$

by definition, the overall heat transfer coefficient is:

$$U = 1 / \left(\frac{r_o}{r_i h_i} + \frac{r_o \ln(r_o/r_i)}{k} + \frac{1}{h_o} \right)$$

where h_o and h_i refer to the outside and inside film coefficient on the tube.

Using the following values,

$$r_o = .66''$$

$$r_i = .52''$$

30% ethylene glycol

$$\mu = .53 \text{ cps}$$

$$k_{\text{steel}} = 21$$

$$k = .30 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$c_p = .95 \text{ Btu/lb-}^\circ\text{F}$$

The Reynolds number is:

$$\begin{aligned} Re &= 6.31 \frac{W}{d \mu} = 6.31 \left(\frac{550000 \text{ BTU/H}}{(.95) 20^\circ\text{F}} \right) \left(\frac{1}{30} \right) / [(1.04 \text{ "Id})(.53 \text{ cps})] \\ &= 11046 \end{aligned}$$

where $W = Q / (c_p \Delta T (\# \text{ of tubes}))$

and the Prandtl number is:

$$\begin{aligned} Pr &= \frac{\mu c_p}{k} = (.53 \text{ cps})(2.419 \frac{\text{lb}}{\text{hr ft (cps)}})(.95 \frac{\text{Btu}}{\text{lb-F}}) / \left[(.30 \frac{\text{Btu}}{\text{hr ft}^2 \text{ F/ft}}) \right] \\ &= 4.06 \end{aligned}$$

Therefore, the inside film coefficient is:

$$\begin{aligned} h_i &= \frac{k}{D} (.023) Re^{.8} Pr^{.3} \\ &= \frac{(.30)}{(1.04/12)} (.023)(11046)^{.8} (4.06)^{.3} \\ &= 208 \frac{\text{Btu}}{\text{hr-ft-}^\circ\text{F}} \end{aligned}$$

and the overall heat transfer coefficient is:

$$U = 1 / \left[\frac{.66}{(.52 \times 2\pi)} + \frac{(.66/12) \ln(.66/.52)}{21} + \frac{1}{h_o} \right]$$

Using the above equation for overall heat transfer coefficient, the following table can be generated.

TABLE 4-1
Variation in Overall Heat Transfer Coefficient
With R-11 Film Coefficient

h_o Btu/hr-ft ² -°F	U Btu/hr-ft ² -°F
∞	149
1500	135
1000	129
500	115
200	85

h_i of the ethylene glycol/water side - 208 Btu/hr-ft²-°F

As can be seen from the above table of U values, the value of the overall heat transfer coefficient is fairly independent of changes in the R-11 film coefficient. The variation in the experimental data on R-11 boiling at 20° ΔT (value for baseline Rankine evaporator) is about 40%. Considering the non-availability of reliable data, the relative insensitivity of the overall heat transfer coefficient to the boiling coefficient leads logically to the assumption that the overall heat transfer coefficient should be made independent of the R-11 film boiling coefficient. The remaining terms in the heat transfer coefficient are functions of the ethylene glycol/water mixture and these values remained constant. The heat transfer coefficient is therefore constant and assumption number three is upheld.

The unit, when built, would be adequately insulated so that heat leak to and from the environment would be minimized. The exclusion of this heat leak from the heat exchanger calculations is considered valid and supports assumption number four.

The evaporators in the Rankine engine and air conditioner were pool-type evaporators. No superheat would be produced in a pool-type exchanger and if a small amount of heat were transferred in the superheat mode, it would not drastically affect the prediction. For these reasons, superheat was not included in the

evaporator models and assumption five is supported.

4.4.4.1.1 Preheater/Evaporator Model

Three models of the preheater/evaporator configuration were developed within this program. Two of the models were used for reported data and the third model generated was used as an independent check. Of the two reported models, one was used to generate the data for nine out of ten of the configurations. A different model and data presentation were developed for the boiling-in-collector configuration. The model generated for a check of the reported models produced results similar to the reported configuration. The model used to check the reported data will not be presented. The following is a presentation of the major preheater/evaporator model and the minor model which will be called the boiling-in-collector model.

4.4.4.1.1.1 Major Model

With certain given input parameters, heat exchanger parameters and solar fluid parameters, values of the heat generated by the solar field are compared with heat transferred to the Rankine engine. In an iterative process, a balance between the solar heat input to the Rankine engine and the heat transferred by the heat exchangers and input to the cycle is achieved.

The heat exchanger model determines if boiling occurs in the preheater or if preheating occurs in the evaporator and adjusts the model accordingly. A flow chart of the heat exchanger model appears in Figure 4-10.

The air conditioning evaporator model is constructed similarly to the Rankine preheater/evaporator model. The obvious departure from the model occurs in the preheating step. No preheating step exists in the air conditioning evaporator model.

FIGURE 4-10
HEAT EXCHANGER MODEL FLOW CHART

4.4.4.1.1.2 Boiling-in-Collector Model

The changes in the scope of work brought about by the emergence of the APRR made it necessary to simulate a boiling-in-collector system for this study. The intent was to make a preliminary cut at modeling the system by using data already generated for one of the 300°F cycles. Data was taken for the high temperature evaporatively cooled system and matched with a designated collector.

The inlet insolation to the collector was determined for a given set of conditions at three different ambient temperatures. The temperatures chosen were 40°F, 70°F and 100°F. The collector model chosen by SAI showed very little efficiency change over the change in ambient conditions and the results of the analysis shows data sets very close to one another.

Three collector areas were simulated, each with a set of three ambient temperatures. The three collector areas were 1275 ft², 1875 ft² and 2500 ft². For the 2500 ft² collector area, a method had to be devised to provide SAI with data for what SAI has called "QOVER". QOVER is the amount of heat that is collected that the Rankine engine cannot use. The situation occurs in this case when temperatures generated exceed the decomposition temperatures of the organic working fluid. A method to present the data was developed and the data was presented to SAI.

4.4.4.2 Condenser Model

Various condenser types and configurations were analyzed in this study and combined with the Rankine cycle and the air conditioning data to determine optimum configurations. Mathematical models were developed and used in the off-design performance simulations. The modeled configurations included an evaporative condenser, a shell-and-tube condenser used with a cooling tower and an air-cooled condenser. The following sections will document the condenser models used in the simulations.

4.4.4.2.1 Evaporative Condenser

The evaporative condenser is essentially a cooling tower and a refrigerant condenser in one piece of equipment. The refrigerant vapor to be condensed is circulated within a condensing coil. This condensing coil is continually wetted on the outer surface by recirculated water. Air is simultaneously blown over the wetted outer coil surface evaporating a small portion of the circulated water. The evaporation provides the driving force for removing heat from the refrigerant within the cooling coil. The refrigerant vapor is condensed and supplied to the Rankine engine.

The following assumptions were used in the evaporative condenser's mathematical model:

- 1) The cooling water temperature remains constant inside the evaporative condenser.
- 2) The working fluid is never heated in the course of its flow.
- 3) The cooling water perfectly wets the outside surface of the condensing coil.
- 4) The vapor velocities are low, permitting the liquid condensate to run off in a stratified condition in open-channel flow.
- 5) The saturation enthalpy of water is linear with the temperature range used in the analysis.

The following basic empirical equations were used as a basis for various transfer coefficients in the mathematical model.

1. Condensing heat transfer coefficient:

$$h_c = .555 \left[\frac{g \rho_R (\rho_R - \rho_{VR}) k_{eR}^3 \Delta h_{fg}}{\mu_{eR} D_i (t_c - t_T)} \right] \quad \text{Ref. 2}$$

where:

- h_c = condensing heat transfer coefficient
- g = acceleration of gravity
- ρ_{eR} = density of liquid refrigerant
- ρ_{VR} = density of vapor refrigerant
- k_{eR} = thermal conductivity of liquid refrigerant
- μ_{eR} = viscosity of liquid refrigerant
- Δh_{fg} = latent heat of refrigerant
- t_c = condensing temperature
- t_T = tube wall temperature
- D_i = inside diameter of the tube in the condensing coil

2. Cooling water heat transfer coefficient:

$$h_w = 118 \left(\frac{\Gamma}{D_o} \right)^{1/3} \quad \text{Ref. 5,6}$$

where:

- Γ = flow rate of cooling water per unit length
- D_o = outside diameter of the tube in the condensing coil

3. Overall capacity coefficient of mass transfer:

$$h_{og} a = 2.5 \times 10^4 (Re_g)^{0.9} (Re_w)^{0.15} (D_o)^{-2.6} \quad \text{REF. 2}$$

where:

- G' = flow rate of air based on minimum cross-section area
- μ_G = viscosity of air
- μ_w = viscosity of water
- $Re_g = \frac{D_o G'}{\mu_G}$
- $Re_w = \frac{4\Gamma}{\mu_G}$

Using algebra and basic transfer equations, the following mathematical model of the evaporative condenser was developed.

$$1. \text{Re}_g = \frac{\dot{W}_{\text{AIR}}}{(n+1) \cdot l \cdot \mu_{\text{AIR}}}$$

$$2. \text{Re}_w = \frac{\dot{W}_{\text{WATER}}}{n \cdot l \cdot \mu_w}$$

$$3. k_{og} a = 2.5 \cdot 10^{-4} \cdot \text{Re}_w^{0.15} \cdot \text{Re}_g^{0.9} \cdot D_o^{-2.6}$$

$$4. S = (2n+1) D_o \cdot l$$

$$5. a' = \sqrt{3} \cdot D_o (2n+1)$$

$$6. h_w = 118 \left(\frac{\dot{W}_{\text{WATER}}}{4n \cdot D_o \cdot l} \right)^{1/3}$$

$$7. E = e^{\frac{k_{og} \cdot a \cdot S \cdot Z}{\dot{W}_{\text{AIR}}}}$$

$$8. h_c = .4561 \left(\frac{g \cdot S_{ER} (S_{ER} - S_{VR}) \Delta h_{fs} \cdot k_{ER}^3 \cdot a' \cdot S \cdot Z}{D_o \cdot \mu_{ER} \cdot Q} \right)^{1/3}$$

$$9. u_o = \frac{1}{\frac{1}{h_w} + \frac{D_o}{D_i \cdot h_c} + R_s}$$

$$10. t_w = t_c - \frac{Q}{u_o \cdot a' \cdot S \cdot Z}$$

$$11. i_w = 1.38 t_w - 75.66$$

$$12. i_1 = i_w + \frac{Q \cdot E}{\dot{W}_{\text{AIR}} (1-E)}$$

$$13. t_{WB} = \frac{i_1 + 75.66}{1.38}$$

4.4.4.2.2 Shell-and-Tube Heat Exchanger and Cooling Tower

A conventional shell-and-tube condenser was used in one of the simulations. Refrigerant condensation occurred on the shell side of the condenser while cooling water from the cooling tower was pumped through the tube side of the condenser.

The following simplifications were used in the mathematical model.

- 1) Shell-and-tube heat exchanger effectiveness and overall heat transfer coefficients are constant over the range of the off-design conditions.
- 2) Cooling tower performance was found by curve fitting empirical correlations from manufacturer's performance data.

Assuming heat exchanger effectiveness of 85%, the following set of equations can be derived.

1. $t_{WOUT} = t_{WIN} + .85 (t_c - t_{WIN})$
2. $\dot{Q}_W = \frac{Q}{C_{PW} (t_{WOUT} - t_{WIN})}$ (REF. 8)
3. $t_{WB} = \frac{t_{WIN} - 28.45 (Q/Q_R)^{0.583}}{1 - .275 (Q/Q_R)^{0.543}}$

where:

- t_{WOUT} = temperature water out of tower
- t_{WIN} = temperature water into tower
- t_c = temperature condenser
- $\dot{Q}_W$ = mass flow of water
- t_{WB} = wet bulb temperature
- Q_R = refrigerant heat load

4.4.4.2.3 Air-Cooled Condenser

An air-cooled condenser was used for one of the simulations. The refrigerant vapor was condensed within the tubes of the condenser core as air flowing over the outer finned surface provided the cooling potential.

A plate fin heat exchanger core was selected for the air-cooled condenser on the basis of the generally higher heat transfer-to-pressure drop ratio for this type of heat exchanger core geometry. Condenser size and tube and fin space were selected on the basis of commercially available equipment.

The equations used in the mathematical model of the air-cooled condenser are documented in the Appendix. Standard techniques for developing the coefficients were used^{2,3,4}. Relevant data for heat transfer coefficient and pressure drop for extended surfaces was analyzed and reduced for this analysis^{2,3,4,7}.

4.4.5 Net Power Model

For the various modes of operation, the power generated or required had to be determined. In the power generation mode, the Rankine turbine shaft output power was reduced by the gearbox losses and the windage loss. The horsepower available was then converted to electrical power via the motor/generator. The calibration curve shown in Figure 4-11 was used. The third quadrant of the graph was used as the unit was producing power in this mode. After the gross kilowatts had been determined, the other parasitic powers in the configuration were computed and the net power calculated. The other parasitic powers included in this mode of operation were the Rankine feed pump power, control power, condenser fans and pumps, and the windage vacuum pump.

For the air conditioning-only mode, the power needed to drive the compressor, gearbox losses and windage were determined. The electrical power required to drive the compressor was deter-

mined from Figure 4-11 (Quad 1). To this gross power, the other required parasitic powers were added to determine the net power needed to provide cooling. The other parasitic powers included were control power, condenser fan and pump power, and the windage vacuum pump power.

In the combined Rankine and air conditioning mode of operation, the compressor shaft power is subtracted from the turbine shaft power available. The gearbox losses are added to the power required/generated and the proper quadrant of Figure 4-11 is consulted to determine the electrical power requirement. The parasitic powers are then added and the net power required or generated is determined. Parasitic power estimates for all the components are presented in the Appendix for the design points of the configured systems.

4.4.6 Process Flow Charts

This section of the report will present the process flow diagrams of the low temperature solar-powered Rankine engine (baseline) and the evaporative condenser. The modifications made to the baseline flow chart for the high temperature simulations will be described. The air conditioning cycle will also be briefly presented.

4.4.6.1 Rankine Low Temperature

A generalized process chart for the Rankine off-design system is presented in Figure 4-12. The information required to compute the cycle analysis for the chosen Rankine configuration is as follows:

- 1) The evaporator pressure
- 2) The condenser pressure
- 3) The turbine efficiency
- 4) The feed pump efficiency

FIG 4-11. MOTOR CALIBRATION CURVE

GENERALIZED RANKINE PROCESS CHART

FIGURE 4-12

- 5) The refrigerant mass flow rate
- 6) Component pressure drops
- 7) Working fluid properties

In the off-design mode of analysis, the first six of the above seven parameters are generated as a function of the off-design characteristics of the system components.

A more detailed process chart with a variable list for the Rankine low temperature off-design system is presented in Figure 4-13. The models that make up this simulation have been addressed in previous sections of the report.

4.4.6.2 Evaporative Condenser and Parasitics

The evaporative condenser process chart and variable lists are presented in Figure 4-14. Within this process chart, the power generated or required by the system operating mode is also calculated. The calculations of the power requirements were a logical extension of the condenser model. In the condenser model data was collated from the Rankine mode and the air conditioning mode for the condenser calculations. Included in the collation of data was the Rankine turbine power produced and the air conditioning power required. These two quantities are the data needed to calculate the power requirements of the various modes of operation. The models that comprise this simulation were discussed previously in this report.

4.4.6.3 High Temperature Rankine Process

The reader is referred to the Rankine process diagram, Figure 4-12 and Figure 4-13 in Section 4.3.6.1. There were two major changes made in the Rankine program when the high temperature configuration was simulated. The changes were in the off-design mass flow rate determination and in the off-design Rankine feed pump characteristic determinations. Since the high temperature system uses an axial turbine, the flow factor $\dot{m}/\dot{m}_{choked}$ is equal to 1 over the entire operating region of the configuration.

RANKINE LOW TEMPERATURE OFF DESIGN SYSTEM FLOW CHART

FIGURE 4-13

VARIABLE LIST FOR FIGURE 4-13

Description	Variable
Rankine evaporator temperature ($^{\circ}\text{R}$)	T_{10}
Rankine evaporator pressure (psia)	P_{10}
Rankine evaporator specific volume (ft^3/lbm)	V_{10}
Rankine evaporator vapor enthalpy (Btu/lbm)	H_{10}
Rankine evaporator vapor entropy (Btu/lbm- $^{\circ}\text{R}$)	S_{10}
Rankine condenser temperature	T_5
Rankine condenser pressure	P_5
Rankine condenser liquid specific volume	V_5
Rankine condenser liquid enthalpy	H_5
Rankine condenser liquid entropy	S_5
Power output (hp)	O
Rankine mass flow rate (lbm/hr)	M
Turbine efficiency	E_1
Pump efficiency	E_2
Gearbox loss (hp)	E_4
Rankine loop pressure drops: (psid)	
Turbine valve	P_{11}
Condenser liquid	P_{12}
Condenser desuperheating	P_{13}
Preheater	P_{16}
Check valve	P_{17}
Filter	P_{18}
Control valve	P_{19}
Boiler	P_{20}

Description	Variable
Rankine turbine inlet temperature	T_o
Rankine turbine inlet pressure	P_o
Rankine turbine inlet vapor specific volume	V_o
Rankine turbine inlet enthalpy	H_o
Rankine turbine inlet vapor entropy	S_o
Isentropic turbine exit temperature	T_1
Isentropic turbine exit pressure	P_1
Isentropic turbine exit vapor specific volume	V_1
Isentropic turbine exit vapor enthalpy	H_1
Isentropic turbine exit vapor entropy	S_1
Rankine turbine exit temperature	T_2
Rankine turbine exit pressure	P_2
Rankine turbine exit vapor specific volume	V_2
Rankine turbine exit vapor enthalpy	H_2
Rankine turbine exit vapor entropy	S_2
Condenser inlet temperature	T_4
Condenser inlet pressure	P_4
Condenser inlet vapor specific volume	V_4
Condenser inlet vapor enthalpy	H_4
Condenser inlet vapor entropy	S_4
Pump exit pressure - working	P_6
Pump exit temperature	T_6
Pump exit pressure	P_6
Pump exit liquid specific volume	V_6
Pump exit liquid enthalpy	H_6
Pump exit liquid entropy	S_6

Description	Variable
Preheater inlet pressure - working	P_7
Preheater inlet temperature	T_7
Preheater inlet pressure	P_7
Preheater inlet liquid specific volume	V_7
Preheater inlet liquid enthalpy	H_7
Preheater inlet liquid entropy	S_7
Boiler inlet vapor temperature	T_9
Boiler inlet vapor pressure	P_9
Boiler inlet vapor specific volume	V_9
Boiler inlet vapor enthalpy	H_9
Boiler inlet vapor entropy	S_9
$\dot{m} * C_p$ solar fluid (Btu/hr- $^{\circ}$ R)	R_{83}
Preheater surface area at design (ft ²)	R_{84}
Rankine evaporator surface area at design (ft ²)	R_{85}
T solar fluid in - T evaporator at design	R_{86}
Low end of condenser pressure loop (psia)	R_{87}
High end of condenser pressure loop (psia)	R_{88}
Condenser pressure loop increment	R_{89}
Low end of T solar fluid in ($^{\circ}$ F)	R_{90}
High end of T solar fluid in ($^{\circ}$ F)	R_{91}
T solar fluid in increment	R_{92}
Overall heat transfer coefficient for preheater (Btu/hr-ft ² - $^{\circ}$ F)	R_{94}
Overall heat transfer coefficient for the Rankine evaporator (Btu/hr-ft ² - $^{\circ}$ F)	R_{95}
$\dot{m}$ Rankine design	R_{65}

Description	Variable
P_{10} design	R_{66}
T_{10} design	R_{67}
Turbine valve design P	R_{69}
U/C_o design	N_{12}
Pressure ratio design	N_{16}
Turbine efficiency design	N_{17}
Condenser pressure loop variable	W
Solar collector fluid inlet temperature	B
Iterative evaporator pressure	R_{100}
Evaporator pressure	P_{10}
Evaporator temperature	T_{10}
Evaporator vapor specific volume	V_{10}
Evaporator vapor enthalpy	H_{10}
Evaporator vapor entropy	S_{10}
Initial guess of Rankine mass flow rate & Rankine ($\dot{m}$ mass flow rate)	R_{101}
ΔP_t turbine valve ΔP for new $\dot{m}$	P_{11}
Pressure for P_{noz} generation; P_{noz} (i-1)	R_{107}
$\dot{m}$ crit (i-1)	R_{109}
Pressure for P_{noz} (i-1)	R_{110}
Pressure increment on P_{noz}	R_{106}
Pressure for P_{noz} generator; P_{noz} (i)	R_{108}
Nozzle temperature	T
Nozzle vapor specific value	V
Nozzle vapor enthalpy	H
Nozzle vapor entropy	S
$\dot{m}$ crit (i)	R_{111}

Description	Variable
$\dot{m}_{crit} (i-2)$	R_{112}
C_o	N_{10}
U/C_o	N_{11}
$(U/C_o)/(U/C_o \text{ design})$	N_{13}
P_o/P_1 pressure ratio	N_{14}
$\dot{m}/\dot{m}_{crit}$ choked	N_5
Mass flow from $\dot{m}_{crit}$	N_6
Pump exit pressure	P_6
Pump efficiency	E_2
Pump exit enthalpy	H_6
Pump exit temperature	T_6
Heat input to Rankine (R_{11})	R_{115}
Solar fluid temperature out of Rankine ($^{\circ}F$)	R_{116}
Solar fluid temperature out of Rankine ($^{\circ}R$)	R_{117}
U preheater for current mass flow	R_{118}
Preheating heat $Q_{R11} \text{ pre}$	R_{119}
Area required in the preheater $A' \text{ pre}$	R_{120}
Area used in Hx boiler model $A' \text{ evap}$	R_{121}
Q_{evap} from Hx model	R_{122}
Total heat input from Hx model	R_{123}

Subroutine List:

Vap:	vap (P,T)	yields	(P,V,T,H,S)
	vap (P,H)	yields	(P,V,T,H,S)
	vap (P,S)	yields	(P,V,T,H,S)
P_{sat} :	$P_{sat} (temp(^{\circ}R))$	yields	saturation pres. (psia)
T_{sat} :	$T_{sat} (pres. (psia))$	yields	saturation temp. ($^{\circ}R$)

Liq: Liq (temperature ($^{\circ}\text{R}$)) yields (T,P,V,H,S) sat. liq.
 HT_{sat}: HT_{sat}(H_{sat} liq) yields T_{sat} ($^{\circ}\text{R}$)

EVAPORATIVE CONDENSER SYSTEM FLOW CHART

NOTE: Variable list on pages 4-62 and 4-63.

FIGURE 4-14

B

$$S: \\ (2(\# \text{ tubes}) + 1) \times \\ \text{tube O.D.} \times l = w23$$

$$a': \\ \frac{\pi \times \# \text{ of tubes}}{\sqrt{3} \times \text{tube O.D.} (2 \times \# \text{ tubes})} \\ = w24$$

$$hw: \\ 118 \left(\frac{\text{water flow}}{4(\# \text{ tubes}) (\text{tube O.D.}) l} \right)^{.333} \\ = w25$$

$$E: \\ \exp \left(\frac{KGA \times S \times \text{height}}{\text{air flow}} \right) \\ = w26$$

$$hc: \\ g(3600)^2 (P_{f \text{ cond}} (P_{f \text{ cond}} - P_{g \text{ cond}})) k^3 (a' \times S) \times \\ \text{height} = w28 \\ (\text{g} = \text{gravitational const.})$$

C

C

$$hc \text{ (cont.)}: \\ .456 \left(\frac{w28}{\text{tube O.D.} \mu_{\text{air}} m} \right)^{.333} \\ = w28$$

$$U_o: \\ \frac{1}{\left(\frac{1}{hw} + \frac{\text{tube O.D.}}{hc \times \text{tube I.D.}} + f_{\text{fouling}} \right)} \\ = w29$$

$$T_{\text{water}}: \\ T_{\text{cond}} = \frac{Q_{\text{cond}}}{(U_o a' S \text{ height})} \\ = w31$$

$$hw(T_w): \\ 1.38(T_{\text{water}}) - 75.66 \\ = w32$$

$$ha(T_{wb}): \\ hw(T_w) + \frac{(Q_{\text{cond}} \times E)}{(\text{air flow} (1-E))} \\ = w33$$

D

VARIABLE LIST: EVAPORATIVE CONDENSER MODEL
FOR FIGURE 4-14

Description	Variable
Condenser temperature ($^{\circ}\text{F}$)	W_1
R_{11} viscosity (cps.)	W_2
R_{11} thermal conductivity (Btu/ft-hr $^{\circ}\text{F}$)	W_3
Condenser tube, O.D.- (ft)	W_4
Condenser tube, I.D. (ft)	W_5
Number of tubes	W_6
Width of condenser assembly (ft)	W_7
Length of condenser assembly (ft)	W_8
Height of condenser assembly (ft)	W_9
Fouling factor, condenser tubes (ft 2 -hr- $^{\circ}\text{F}$ /Btu)	W_{10}
Air flow (lb/hr)	W_{11}
Water flow (lb/hr)	W_{12}
Air viscosity (c_p)	W_{13}
Water viscosity (c_p)	W_{14}
Air Reynolds number	W_{20}
Water Reynolds number	W_{21}
KOGA (heat transfer parameter - see referenced article)	W_{22}
S (heat transfer parameter - see referenced article)	W_{23}
a' (heat transfer parameter - see referenced article)	W_{24}
hw (water side film coefficient)(Btu/hr-ft $^{\circ}\text{F}$)	W_{25}
E (heat transfer parameter - see referenced article)	W_{26}
$\dot{m} R_{11}$ - RC only or RC/AC combination (lb/hr)	W_{27}
hc (R_{11} side film coefficient)(Btu/hr-ft $^{\circ}\text{F}$)	W_{28}
U_o (overall heat transfer coefficient)(Btu/hr-ft 3 - $^{\circ}\text{F}$)	W_{29}

Description	Variable
Q condenser (RC only or RC/AC comb.)(Btu/hr)	W ₃₀
Condenser water temperature (°F)	W ₃₁
$h_w (T_w)$	W ₃₂
$h_a (T_{wb})$	W ₃₃
T_{wb} (wet bulb) °F RC only or RC/AC comb.	W ₃₄
RC only or RC/AC combined mode flag	W ₃₅
$\dot{m}_{AC}$ (R_{11} mass flow rate - AC mode)(lb/hr)	W ₃₆
Q_{AC} (AC condenser heat load)(Btu/hr)	W ₃₇
R_{11} vapor density at $T_{condenser}$ (lb/ft ³)	W ₃₈
Condenser pressure flag	W ₄₀
Compressor power requirement (kW)	W ₄₁
Condenser pressure (psia)	P ₅
Condenser temperature (°R)	T ₅
R_{11} liquid density at $T_{cond.}$ (lb/ft ³)	D ₅
Turbine shaft power (hp)	R ₂
Q condenser RC only (Btu/hr)	R ₈
$\dot{m}_{R_{11}}$ RC only (lb/hr)	R ₁₀₁
Power to motor, hp, RC only or RC/AC combined	R ₃
Power out, RC or RC/AC combined	R ₄
Net power to grid, kW, RC or RC/AC combined	R ₅

The Rankine feed pump used was a variable speed-type pump which maximized the efficiency of this particular configuration. Since the pump was variable speed, the calculations of parasitic power required to drive this pump were also slightly altered in the analysis.

4.4.6.4 Air Conditioning Process

The air conditioning program generated was very similar to the Rankine program. If one takes Figure 4-12 and substitutes air conditioner for Rankine, the process is presented. The differences between the Rankine and air conditioning cycle have been previously discussed in the compressor modeling and heat exchanger modeling sections of this report.

5.0 RESULTS

The data for the ten configurations was tabulated and presented to SAI for their further development. This data along with the information required to produce the data is presented in the Appendix. The data was tabulated for use in computer simulation. Some of the data is also graphically presented for ease of comprehension.

The power generation mode for the baseline system is presented in Figure 5-1. For a solar collector fluid inlet temperature of 210°F , the baseline configuration would produce 14 kW over a large range of ambient wet bulb temperatures. The net power output shown has had the condenser fan and pump parasitics, gearbox losses, and control losses taken out. The power output of the Rankine engine generally increases with increasing solar temperature and decreasing wet bulb temperature. This trend is not surprising if one considers a standard Carnot engine. As the difference between the heat source and the heat sink increases in the Carnot engine, the power output available increases. The power output curve approaches an infinite slope condition at the high solar temperature and low wet bulb temperature portion of Figure 5-1 due to the particular characteristic of the turbine efficiency. This area of the plot represents a turbine pressure ratio much greater than at the design point for the turbine. The turbine efficiency for the high pressure ratio was shown in Figure 4-4.

The air conditioning cycle power required and tonnage for a given ambient wet bulb temperature is presented in Figure 5-2 for the baseline configuration air conditioner. The air conditioner operating without the Rankine engine produces 25 tons of cooling at an ambient wet bulb of 87°F . The warmest wet bulb temperatures in the world⁹ do not exceed a value of 86°F for the 1% design point. The power required to drive this air conditioner at 86°F wet bulb temperature is 16.5 kW. The tonnage curve in Figure 5-2 decreases with increasing wet bulb temperature. The wet bulb temperature is directly related to the condenser temperature. The increasing condenser temperature carries with it increasing

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FIG. 5-1

BASELINE SYSTEM
PERFORMANCE MAP
RANKINE CYCLE ONLY

FIG. 5-2

BASELINE SYSTEM
REFRIGERATION PERFORMANCE VARIATION
WITH WET BULB TEMPERATURE

saturated liquid enthalpy values. Since the air conditioning evaporator temperature is set, the value of the evaporator exit enthalpy remains a constant. The difference between the evaporator exit enthalpy and the increasing condenser enthalpy decreases with the rising condenser temperature. This difference (refrigeration effect) multiplied by the mass flow rate gives the cooling load supplied. The cooling load therefore decreases as the wet bulb temperature increases. This trend is evident in Figure 5-2. The power required to compress the air conditioning evaporator exit vapor to the condenser conditions increases as the difference in pressure between the air conditioner evaporator and the condenser increases. This trend is also shown in Figure 5-2.

The combined baseline Rankine cycle and air conditioner is presented in Figure 5-3. The net electrical power to the electric motor is shown along with the tonnage produced for various solar fluid temperatures and ambient wet bulb temperatures. Negative kilowatt quantities represent power required to drive the system to produce the listed tonnage. Positive kilowatt readings represent generated power supplied to the electrical grid in addition to the tonnage supplied by this configuration. The 0 kW line represents the conditions that will allow the Rankine engine to produce the power required by the air conditioner and supply the system parasitics requirements at no energy cost.

The power curves on Figure 5-3 follow the same trends as the power generation curves in Figure 5-1. The tonnage lines vary mostly as a function of ambient wet bulb temperature which is shown in Figure 5-2 with some variation due to heavier condenser loading due to the Rankine engine.

The high temperature configuration is presented graphically in Figures 5-4 through 5-6. The trends are similar to those for the baseline system (Figure 5-1 through 5-3) but the engine is more efficient due to the higher solar fluid temperatures. This engine requires a smaller collector field to produce the same power as the baseline system and would therefore be more cost effective.

FIG. 5-3

BASELINE SYSTEM
 PERFORMANCE MAP
 COMBINED RANKINE CYCLE & REFRIGERATION CYCLE

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FIG. 5-4

PERFORMANCE MAP
HI TEMP SYSTEM
RANKINE CYCLE ONLY

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FIG 5-5

HIGH TEMP SYSTEM

REFRIGERATION PERFORMANCE VARIATION
WITH WET BULB TEMPERATURE

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FIG. 5-6

PERFORMANCE MAP
HIGH TEMP SYSTEM
RANKINE CYCLE + REFRIGERATION CYCLE

Comparing the BIC configuration (Figure 5-7) with the baseline configuration (Figure 4-1) several differences are obvious. The solar collector loop pump no longer exists in the BIC configuration, thus reducing parasitic power. All of the preheating and evaporating of the refrigerant takes place in the collector assembly, eliminating the need for a Rankine engine preheater and evaporator. This not only eliminates two costly heat exchangers from the Rankine engine but is also a more efficient means of heat transfer. In the baseline configuration, the design point difference between the incoming solar collector fluid temperature and the saturated vapor refrigerant at the exit of the evaporator was 20°F. This temperature loss would not exist in the boiling-in-collector assembly.

Although not evident from the figures, the two systems differ in cycle efficiency. The high temperature BIC system (280°F evaporator) as compared with the baseline configuration (180°F evaporator) exhibits 25% and 13% respective Carnot efficiencies (95°F condenser). The actual engine efficiencies, although obviously less than Carnot, follow the same trends. The more efficient high temperature BIC system produces more energy per year for a given heat input when compared to the lower temperature baseline. It should be pointed out that high temperature (280°F evaporator) data was used for the BIC simulation. The design point for the BIC system should have been a 300°F evaporator to take full advantage of the eliminated preheater and evaporator. The cycle efficiency and the power output would have been higher than reported if the cycle could have been resimulated for the 300°F evaporator.

The BIC configuration would probably use tracking concentrator technology. The tracking concentrators deliver the higher temperatures specified and the net square footage of the collectors is decreased over the more conventional low temperature flat plate collectors that would be used for a baseline system. Other collector types are being used for both the high temperature and low temperature systems and future research efforts should pinpoint the optimum collector configurations.

FIG. 5-7
BOILING-IN-COLLECTORS

The boiling-in-collector data is presented in the Appendix. The data has been graphically presented in Figures 5-8, 5-9, and 5-10. The data graphed is for a collector area of 1875 ft² and an ambient temperature of 70°F. Solar insolation is used in this set of graphs rather than solar collector fluid inlet temperature (T_{si}) because this model includes the collector. The reason several ambient temperatures and collector areas (see Appendix) have been run is that the collector performance depends upon both the collector area and the ambient temperature. The trends of this data are similar to the baseline system data. That is, the engine performance increases with increased solar input and with decreasing wet bulb temperatures. The air conditioner performance is closely related to wet bulb temperatures and improves with decreasing wet bulb. The combined system performance then is best at high solar input with low wet bulb. Tonnage drops with increasing wet bulb. This system has better performance than the corresponding high temperature unit (Figures 5-4 through 5-6) because of the elimination of some heat exchange processes and some parasitic power losses. The result is the capability to deliver the cooling load with a smaller collector area (and therefore cost).

One of Barber-Nichols' tasks was to review the yearly simulation modeling (by SAI). Some of the more important results which were reviewed will be briefly discussed. The results are presented in more detail in the SAI final report¹⁰ and the Honeywell final report¹¹.

The results of the 10 hp and 30 hp scaling run are represented in Figure 5-11. The figure shows an optimum of about 20 hp. These results agree with the preliminary work that was done early in the NASA 404 program when 20 hp to 25 tons of cooling was chosen as a design point.

The high temperature (300°F) Rankine engine was simulated with an air-cooled condenser, a cooling tower water-cooled condenser, and an evaporatively cooled condenser. The results are shown in Figure 5-12. As was pointed out in Section 4, the

FIG.5-8

PERFORMANCE MAP

BIC SYSTEM

RANKINE CYCLE ONLY

FIG 5-9

BIC SYSTEM

REFRIGERATION PERFORMANCE VARIATION
WITH WET BULB TEMPERATURE

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FIG. 5-10

PERFORMANCE MAP

BIC SYSTEM

RANKINE CYCLE + REFRIGERATION CYCLE

ANNUAL ENERGY REQUIREMENT VARIATION WITH ENGINE SIZE

FIG. 5-II

ANNUAL ENERGY REQUIREMENT FOR HIGH TEMPERATURE CONFIGURATION WITH VARIOUS CONDENSER TYPES.

FIG. 5-12

evaporatively cooled condenser should win out from an energy standpoint. Indeed, the figure shows this. The application for which the configuration is to be used determines the choice of the type of condenser assembly that must be used but the evaporative configuration has the best performance.

Several of the previously discussed configurations are compared in Figure 5-13 using power and collector area as a basis. The higher temperature cycles generate more energy for a given collector area due mainly to the increased efficiency of the higher temperature cycles. The boiling-in-collector system is superior to the high temperature system because the nominal 10° to 20° temperature difference between the solar collector fluid and the Rankine engine turbine inlet temperature is eliminated in the BIC configuration. The 170°F BNL cycle has the least output for collector area; however other factors such as overall cost may make this type of configuration desirable.

The experimental work described in Section 3 showed that an internal refrigerant and oil lubrication system could be used in the compressor gearbox assembly. The work also produced an estimate of the windage loss for the compressor to be used in the computer simulations.

FIG 5-13
ENERGY & COLLECTOR AREA COMPARISONS

6.0 CONCLUSIONS

The following ten configurations were defined and modeled:

- Baseline - evaporatively cooled condenser
- High Temperature - evaporatively cooled condenser
- High Temperature - air-cooled condenser
- High Temperature - cooling tower-cooled condenser
- High Temperature - space heating application
- High Speed Alternator
- 10 and 30 HP Scaling of Rankine Engine
- Baseline Configuration - 45° leaving chilled water
- BNL Low Temperature Rankine Cycle
- BIC Configuration - 1875 ft² of collector
 - 1250 ft² of collector
 - 2500 ft² of collector

Some experimental work was done to support some of the modeled configuration assumptions. The data for these ten configurations was relayed to Science Applications Inc. for their use in a companion study.

Additional configurations were explored but not modeled as it was thought the data obtained from these configurations would not yield beneficial results. The configurations not modeled included:

- Variable Area Turbine Nozzle
- Turbo-feed pump
- Aerodynamic Unloading
- Power Factor Correcting Device

Originally a system configuration was to be chosen that could be built with current technology. During the course of the study this objective was modified to reflect the future requirements of the solar-powered Rankine engine air conditioner.

A configuration was identified that would decrease parasitic losses, lower the number of components, promote the highest energy savings per year and maximize energy output per unit area of collector. This technically superior system has been dubbed the boiling-in-collector system (BIC). The modeled boiling-in-collector configuration was shown in Figure 5-7. The output for this configuration was graphically presented in Figures 5-8 through 5-10. The data for this configuration is presented in the Appendix.

The program did not stipulate the inclusion of a high speed alternator in place of the motor/generator on the BIC configuration; however, it would save some parasitic power. Future research efforts should point the way for inclusion of this type of power converter in a future BIC system.

Under this program ten different configurations of solar-driven Rankine air conditioners were defined and modeled. The data was relayed to Science Applications Inc. and Honeywell International for use in companion studies which are listed in the Reference section. The relayed data as well as some of the mathematical models used to produce that data are presented within the context of this report. The results of the analysis show that a boiling-in-collector, high temperature, evaporatively cooled condenser system to be technically superior.

REFERENCES

1. ASHRAE Handbook, 1977 Fundamentals, Pg. 17.8.
2. Rohsenow and Hartnett, Handbook of Heat Transfer, McGraw Hill, 1973.
3. Kays and London, Compact Heat Exchangers, McGraw Hill, 1981.
4. Kakas, Bergles, Mayinger, Heat Exchangers, McGraw Hill, 1981.
5. Mizushima, Ito and Miyashita, "Experimental Study of an Evaporative Cooler", International Chemical Engineering, Vol 7, No. 4, Oct. 1967.
6. Mizushima, Ito and Miyashita, "Characteristics and Methods of Thermal Design of Evaporative Coolers", International Chemical Engineering, Vol 8, No. 3, July 1968.
7. Webb, R.L., "Air-Side Heat Transfer in Finned Tube Heat Exchangers", Heat Transfer Engineering, Vol 1, No. 3, January - March 1980.
8. Baltimore Aircoil, "VX Cooling Towers", Bulletin S280/1-0. Baltimore, MD, July 1978.
9. ASHRAE Handbook, 1977 Fundamentals, Pg. 23.18, Table 3.

The following two references are companion studies:

10. Choi, Mike, "Analysis for Commercial and Residential Solar Absorption and Rankine Cooling Systems", Contract DE-AC03815-F11573, Draft-final Report, Science Applications Inc., McLean, VA.
11. Aasen, R.K.; "Final Report: Second Generation Solar Rankine Air Conditioning System Study", Contract DE-AC03-81CS30574, Honeywell International.

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APPENDIX A

10 SIMULATIONS PRODUCED FOR SCIENCE APPLICATIONS, INC.

- BASELINE - EVAPORATIVELY COOLED CONDENSER
- HIGH TEMPERATURE - EVAPORATIVELY COOLED CONDENSER
- HIGH TEMPERATURE - AIR COOLED CONDENSER
- HIGH TEMPERATURE - COOLING TOWER COOLED CONDENSER
- HIGH TEMPERATURE - SPACE HEATING APPLICATION
- HIGH SPEED ALTERNATOR
- 10 & 30 HP SCALING OF RANKINE ENGINE
- BASELINE CONFIGURATION - 45° CHILLED WATER
- ENL LOW TEMPERATURE RANKINE CYCLE
- BIC CONFIGURATION
 - 1875 SQUARE FEET OF COLLECTOR
 - 1250 " " "
 - 2500 " " "

Due to the large amount of data originally in this Appendix, it was asked that it be reduced for printing. The input data for each set of simulations is included and the full data set for the BIC system for 1875 ft² collector area is retained. The full data set was conveyed to SAI for the completion of their government contract. Results from the analysis are presented in the text of this report and References 10 and 11.

INPUT PARAMETERS TO COMPUTER SIMULATIONS SYSTEM - BASELINE

INPUT PARAMETERS FOR DESIGN POINT CYCLE

RANKINE:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>180.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>95.0</u>
Turbine efficiency	<u>80.0%</u>
Feed pump efficiency	<u>41.0%</u>
Turbine shaft power output, hp	<u>20.00</u>
Pressure drops: psid	
Turbine valve	<u>2.0</u>
Condenser	<u>0.8</u>
Check valve	<u>2.0</u>
Filter	<u>2.0</u>
Control valve	<u>31.0</u>
Preheater	<u>2.0</u>
Evaporator	<u>0.0</u>

AIR CONDITIONER:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>50.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>95.0</u>
Compressor efficiency	<u>80.0%</u>
Cooling required, tons	<u>25.0</u>
Pressure drops: psid	
Compressor inlet	<u>0.38</u>
Compressor exit	<u>0.84</u>

INPUT PARAMETERS FOR THE CONDENSER SIMULATION:

Height, ft	<u>3.15</u>
Width, ft	<u>3.25</u>
Length, ft	<u>11.0</u>
Number of tubes	<u>18</u>
O.D., in	<u>1.05</u>
I.D., in	<u>0.83</u>
Air flow, cfm	<u>11,780</u>
Water flow, gpm	<u>151.3</u>
Condenser pressure -	
Upper limit, psia	<u>27.0</u>
Lower limit, psia	<u>9.0</u>
Increment	<u>2.0</u>

DESIGN POINT PARASITICS:

	RC only	AC only	RC & AC
Gearbox loss, hp	<u>1.15</u>	<u>1.15</u>	<u>1.15</u>
Turbine/compressor windage, hp	<u>0.1</u>	<u>0.1</u>	<u>- 0 -</u>
Controls, kW	<u>0.4</u>	<u>0.2</u>	<u>0.4</u>
Condenser fan, kW	<u>1.03</u>	<u>1.03</u>	<u>1.03</u>
Condenser pump, kW	<u>0.904</u>	<u>0.904</u>	<u>0.904</u>
Windage vacuum pump, kW	<u>0.1</u>	<u>0.1</u>	<u>- 0 -</u>
Feed pump (design point) kW	<u>1.2</u>	<u>- 0 -</u>	<u>1.2</u>
Motor/generator efficiency	<u>91%</u>	<u>91%</u>	<u>91%</u>

INPUT PARAMETERS FOR THE OFF-DESIGN SIMULATION:

RANKINE:

$\dot{m} c_p$ solar, Btu/hr-°F	27.443
Preheater surface area, ft ²	100.0
Evaporator surface area, ft ²	400.0
ΔT solar to evaporator, °F	20.0
Condenser pressure -	
Upper limit, psia	27.0
Lower limit, psia	9.0
Increment	2.0
Solar collector fluid temperature -	
Upper limit, °F	220.0
Lower limit, °F	130.0
Increment	10.0
Preheater U design, Btu/hr-ft ² -°F	41.7
Evaporator U design, Btu/hr-ft ² -°F	107.7
Turbine diameter, in	3.499
Turbine nozzle area, in ²	0.5520

AIR CONDITIONER:

$\dot{m} c_p$ chilled water, Btu/hr-°F	30.000
Evaporator surface area, ft ²	244.1
Evaporator U design, Btu/hr-ft ² -°F	135
Chilled water temperature, °F	55.0
Condenser pressure -	
Upper limit, psia	27.0
Lower limit, psia	9.0
Increment	2.0
ΔT water to evaporator, °F	5.0
Compressor diameter, in	5.05

INPUT PARAMETERS TO COMPUTER SIMULATIONS
SYSTEM HIGH TEMP. EVAP. COND.

INPUT PARAMETERS FOR DESIGN POINT CYCLE

RANKINE:

T _{evaporator} , °F	<u>280.0</u>
T _{condenser} , °F	<u>95.0</u>
Turbine efficiency	<u>76.7%</u>
Feed pump efficiency	<u>44.5%</u>
Turbine shaft power output, hp	<u>23.0</u>
Pressure drops: psid	
Turbine valve	<u>2.0</u>
Condenser	<u>0.8</u>
Check valve	<u>2.0</u>
Filter	<u>2.0</u>
Control valve	<u>20.3</u>
Preheater	<u>2.0</u>
Evaporator	<u>0.0</u>

AIR CONDITIONER:

T _{evaporator} , °F	<u>50.0</u>
T _{condenser} , °F	<u>95.0</u>
Compressor efficiency	<u>80.0%</u>
Cooling required, tons	<u>25.0</u>
Pressure drops: psid	
Compressor inlet	<u>0.38</u>
Compressor exit	<u>0.84</u>

INPUT PARAMETERS FOR THE OFF-DESIGN SIMULATION:

RANKINE:

$\dot{m}_p$ solar, Btu/hr-°F	17,719
Preheater surface area, ft ²	50.0
Evaporator surface area, ft ²	200.0
ΔT solar to evaporator, °F	20.0
Condenser pressure -	
Upper limit, psia	29.0
Lower limit, psia	9.0
Increment	2.0
Solar collector fluid temperature -	
Upper limit, °F	310.0
Lower limit, °F	230.0
Increment	10.0
Preheater U design, Btu/hr-ft ² -°F	54.3
Evaporator U design, Btu/hr-ft ² -°F	74.7
Turbine diameter, in	3.533
Turbine nozzle area, in ²	0.1053

AIR CONDITIONER:

$\dot{m}_p$ chilled water, Btu/hr-°F	30,000
Evaporator surface area, ft ²	244.1
Evaporator U design, Btu/hr-ft ² -°F	135
Chilled water temperature, °F	55.0
Condenser pressure -	
Upper limit, psia	29.0
Lower limit, psia	9.0
Increment	2.0
ΔT water to evaporator, °F	5.0
Compressor diameter, in	5.05

INPUT PARAMETERS FOR THE CONDENSER SIMULATION:

Height, ft	<u>3.14</u>
Width, ft	<u>3.52</u>
Length, ft	<u>8.0</u>
Number of tubes	<u>19</u>
O.D., in	<u>1.05</u>
I.D., in	<u>0.824</u>
Air flow, cfm	<u>9,300</u>
Water flow, gpm	<u>154.0</u>
Condenser pressure -	
Upper limit, psia	<u>29.0</u>
Lower limit, psia	<u>9.0</u>
Increment	<u>2.0</u>

DESIGN POINT PARASITICS:

	RC only	AC only	RC & AC
Gearbox loss, hp	<u>1.15</u>	<u>1.15</u>	<u>1.15</u>
Turbine/compressor windage, hp	<u>0.1</u>	<u>0.1</u>	<u>-0-</u>
Controls, kW	<u>0.4</u>	<u>0.2</u>	<u>0.4</u>
Condenser fan, kW	<u>0.795</u>	<u>0.795</u>	<u>0.795</u>
Condenser pump, kW	<u>0.904</u>	<u>0.904</u>	<u>0.904</u>
Windage vacuum pump, kW	<u>0.1</u>	<u>0.1</u>	<u>-0-</u>
Feed pump (design point) kW	<u>1.65</u>	<u>-0-</u>	<u>1.65</u>
Motor/generator efficiency	<u>91%</u>	<u>91%</u>	<u>91%</u>

INPUT PARAMETERS TO COMPUTER SIMULATIONS
SYSTEM HIGH TEMP. AIR COOLED

INPUT PARAMETERS FOR DESIGN POINT CYCLE

RANKINE:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>280.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>115.0</u>
Turbine efficiency	<u>76.7%</u>
Feed pump efficiency	<u>25.6%</u>
Turbine shaft power output, hp	<u>22.0</u>
Pressure drops: psid	
Turbine valve	<u>2.0</u>
Condenser	<u>0.8</u>
Check valve	<u>2.0</u>
Filter	<u>2.0</u>
Control valve	<u>20.0</u>
Preheater	<u>2.0</u>
Evaporator	<u>0.0</u>

AIR CONDITIONER:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>50.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>100.0</u>
Compressor efficiency	<u>80.0%</u>
Cooling required, tons	<u>25.0</u>
Pressure drops: psid	
Compressor inlet	<u>0.38</u>
Compressor exit	<u>0.84</u>

INPUT PARAMETERS FOR THE OFF-DESIGN SIMULATION:

RANKINE:

$\dot{m}c_p$ solar, Btu/hr-°F	19379
Preheater surface area, ft ²	50.0
Evaporator surface area, ft ²	200.0
ΔT solar to evaporator, °F	20.0
Condenser pressure -	
Upper limit, psia	31.0
Lower limit, psia	9.0
Increment	2.0
Solar collector fluid temperature -	
Upper limit, °F	310.0
Lower limit, °F	230.0
Increment	10.0
Preheater U design, Btu/hr-ft ² -°F	62.2
Evaporator U design, Btu/hr-ft ² -°F	89.2
Turbine diameter, in	3.345
Turbine nozzle area, in ²	0.1216

AIR CONDITIONER:

$\dot{m}c_p$ chilled water, Btu/hr-°F	30,000
Evaporator surface area, ft ²	244.2
Evaporator U design, Btu/hr-ft ² -°F	135.0
Chilled water temperature, °F	55.0
Condenser pressure -	
Upper limit, psia	31.0
Lower limit, psia	9.0
Increment	2.0
ΔT water to evaporator, °F	5.0
Compressor diameter, in	5.05

INPUT PARAMETERS FOR THE CONDENSER SIMULATION:

Height, ft	<u>0.43</u>
Width, ft	<u>5.0</u>
Length, ft	<u>10.0</u>
Number of tubes	<u>100</u>
O.D., in	<u>0.425</u>
I.D., in	<u>0.555</u>
Air flow, cfm	<u>28,165</u>
Water flow, gpm	<u>- 0 -</u>
Condenser pressure -	
Upper limit, psia	<u>31.0</u>
Lower limit, psia	<u>9.0</u>
Increment	<u>2.0</u>

DESIGN POINT PARASITICS:

	RC only	AC only	RC & AC
Gearbox loss, hp	<u>1.15</u>	<u>1.15</u>	<u>1.15</u>
Turbine/compressor windage, hp	<u>0.1</u>	<u>0.1</u>	<u>- 0 -</u>
Controls, kW	<u>0.4</u>	<u>0.2</u>	<u>0.4</u>
Condenser fan, kW	<u>1.77</u>	<u>1.77</u>	<u>3.5</u>
Condenser pump, kW	<u>- 0 -</u>	<u>- 0 -</u>	<u>- 0 -</u>
Windage vacuum pump, kW	<u>0.15</u>	<u>0.15</u>	<u>0.15</u>
Feed pump (design point) kW	<u>1.63</u>	<u>- 0 -</u>	<u>1.63</u>
Motor/generator efficiency	<u>91%</u>	<u>91%</u>	<u>91%</u>

INPUT PARAMETERS TO COMPUTER SIMULATIONS
SYSTEM HIGH TEMP. - COOLING TOWER

INPUT PARAMETERS FOR DESIGN POINT CYCLE

RANKINE:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>280.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>100.0</u>
Turbine efficiency	<u>76.7%</u>
Feed pump efficiency	<u>46.2%</u>
Turbine shaft power output, hp	<u>22.0</u>
Pressure drops: psid	
Turbine valve	<u>2.0</u>
Condenser	<u>0.8</u>
Check valve	<u>2.0</u>
Filter	<u>2.0</u>
Control valve	<u>20.0</u>
Preheater	<u>2.0</u>
Evaporator	<u>0.0</u>

AIR CONDITIONER:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>50.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>100.0</u>
Compressor efficiency	<u>80.0%</u>
Cooling required, tons	<u>25.0</u>
Pressure drops: psid	
Compressor inlet	<u>0.38</u>
Compressor exit	<u>0.84</u>

INPUT PARAMETERS FOR THE OFF-DESIGN SIMULATION:

RANKINE:

$\dot{m}_p$ solar, Btu/hr-°F	18,112
Preheater surface area, ft ²	50.0
Evaporator surface area, ft ²	200.0
ΔT solar to evaporator, °F	20.0
Condenser pressure -	
Upper limit, psia	63.0
Lower limit, psia	27.0
Increment	6.0
Solar collector fluid temperature -	
Upper limit, °F	320.0
Lower limit, °F	230.0
Increment	10.0
Preheater U design, Btu/hr-ft ² -°F	56.90
Evaporator U design, Btu/hr-ft ² -°F	78.88
Turbine diameter, in	3.345
Turbine nozzle area, in ²	0.1216

AIR CONDITIONER:

$\dot{m}_p$ chilled water, Btu/hr-°F	30,016
Evaporator surface area, ft ²	244.2
Evaporator U design, Btu/hr-ft ² -°F	135.0
Chilled water temperature, °F	55.0
Condenser pressure -	
Upper limit, psia	63.0
Lower limit, psia	27.0
Increment	6.0
ΔT water to evaporator, °F	5.0
Compressor diameter, in	5.05

INPUT PARAMETERS FOR THE CONDENSER SIMULATION:

Height, ft	3.14
Width, ft	3.25
Length, ft	8.0
Number of tubes	19
O.D., in	1.05
I.D., in	0.824
Air flow, cfm	9,300
Water flow, gpm	154.0
Condenser pressure -	
Upper limit, psia	63.0
Lower limit, psia	27.0
Increment	6.0

DESIGN POINT PARASITICS:

	RC only	AC only	RC & AC
Gearbox loss, hp	1.15	1.15	1.15
Turbine/compressor windage, hp	0.1	0.1	-0-
Controls, kW	0.4	0.2	0.4
Condenser fan, kW	1.03	1.03	1.03
Condenser pump, kW	1.36	1.36	1.36
Windage vacuum pump, kW	0.1	0.1	-0-
Feed pump (design point) kW	1.66	-0-	1.66
Motor/generator efficiency	91%	91%	91%

INPUT PARAMETERS TO COMPUTER SIMULATIONS

SYSTEM HIGH TEMP. COOLING TOWER - SPACE HEATINPUT PARAMETERS FOR DESIGN POINT CYCLE

RANKINE:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>280.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>100.0</u>
Turbine efficiency	<u>76.7 %</u>
Feed pump efficiency	<u>46.2 %</u>
Turbine shaft power output, hp	<u>22.0</u>
Pressure drops: psid	
Turbine valve	<u>2.0</u>
Condenser	<u>0.8</u>
Check valve	<u>2.0</u>
Filter	<u>2.0</u>
Control valve	<u>20.0</u>
Preheater	<u>2.0</u>
Evaporator	<u>0.0</u>

AIR CONDITIONER:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>50.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>100.0</u>
Compressor efficiency	<u>80.0 %</u>
Cooling required, tons	<u>25.0</u>
Pressure drops: psid	
Compressor inlet	<u>0.38</u>
Compressor exit	<u>0.84</u>

INPUT PARAMETERS FOR THE OFF-DESIGN SIMULATION:

RANKINE:

$\dot{m}_p$ solar, Btu/hr-°F	18,112
Preheater surface area, ft ²	50.0
Evaporator surface area, ft ²	200.0
ΔT solar to evaporator, °F	20.0
Condenser pressure -	
Upper limit, psia	63.0
Lower limit, psia	27.0
Increment	6.0
Solar collector fluid temperature -	
Upper limit, °F	320.0
Lower limit, °F	230.0
Increment	10.0
Preheater U design, Btu/hr-ft ² -°F	56.90
Evaporator U design, Btu/hr-ft ² -°F	78.88
Turbine diameter, in	3.345
Turbine nozzle area, in ²	0.1216

AIR CONDITIONER:

$\dot{m}_p$ chilled water, Btu/hr-°F	30,016
Evaporator surface area, ft ²	244.2
Evaporator U design, Btu/hr-ft ² -F	135.0
Chilled water temperature, F	55.0
Condenser pressure -	
Upper limit, psia	63.0
Lower limit, psia	27.0
Increment	6.0
ΔT water to evaporator, F	5.0
Compressor diameter, in	5.05

INPUT PARAMETERS FOR THE CONDENSER SIMULATION:

Height, ft	3.14
Width, ft	3.52
Length, ft	8.0
Number of tubes	19
O.D., in	1.05
I.D., in	0.824
Air flow, cfm	9,300
Water flow, gpm	154.0
Condenser pressure -	
Upper limit, psia	43.0
Lower limit, psia	27.0
Increment	6.0

DESIGN POINT PARASITICS:

	RC only	AC only	RC & AC
Gearbox loss, hp	1.15	1.15	1.15
Turbine/compressor windage, hp	0.1	0.1	- 0 -
Controls, kW	0.4	0.2	0.4
Condenser fan, kW	1.03	1.03	1.03
Condenser pump, kW	1.36	1.36	1.36
Windage vacuum pump, kW	0.1	0.1	- 0 -
Feed pump (design point) kW	1.66	- 0 -	1.66
Motor/generator efficiency	91%	91%	91%

INPUT PARAMETERS TO COMPUTER SIMULATIONS
SYSTEM HIGH SPEED ALTERNATOR

INPUT PARAMETERS FOR DESIGN POINT CYCLE

RANKINE:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>180.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>95.0</u>
Turbine efficiency	<u>80.0%</u>
Feed pump efficiency	<u>41.0%</u>
Turbine shaft power output, hp	<u>20.0</u>
Pressure drops: psid	
Turbine valve	<u>2.0</u>
Condenser	<u>0.8</u>
Check valve	<u>2.0</u>
Filter	<u>2.0</u>
Control valve	<u>31.0</u>
Preheater	<u>2.0</u>
Evaporator	<u>0.0</u>

AIR CONDITIONER:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>50.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>95.0</u>
Compressor efficiency	<u>80.0%</u>
Cooling required, tons	<u>25.0</u>
Pressure drops: psid	
Compressor inlet	<u>0.38</u>
Compressor exit	<u>0.84</u>

INPUT PARAMETERS FOR THE OFF-DESIGN SIMULATION:

RANKINE:

$\dot{m}_p$ solar, Btu/hr-°F	27,443
Preheater surface area, ft ²	100.0
Evaporator surface area, ft ²	400.0
ΔT solar to evaporator, °F	20.0
Condenser pressure -	
Upper limit, psia	27.0
Lower limit, psia	9.0
Increment	2.0
Solar collector fluid temperature -	
Upper limit, °F	220.0
Lower limit, °F	130.0
Increment	10.0
Preheater U design, Btu/hr-ft ² -°F	41.7
Evaporator U design, Btu/hr-ft ² -°F	107.7
Turbine diameter, in	3.499
Turbine nozzle area, in ²	0.5520

AIR CONDITIONER:

$\dot{m}_p$ chilled water, Btu/hr-°F	30,000
Evaporator surface area, ft ²	244.1
Evaporator U design, Btu/hr-ft ² -°F	135
Chilled water temperature, °F	55.0
Condenser pressure -	
Upper limit, psia	9.0
Lower limit, psia	27.0
Increment	2.0
ΔT water to evaporator, °F	5.0
Compressor diameter, in	5.05

INPUT PARAMETERS FOR THE CONDENSER SIMULATION:

Height, ft	<u>3.15</u>
Width, ft	<u>3.25</u>
Length, ft	<u>11.0</u>
Number of tubes	<u>18</u>
O.D., in	<u>1.05</u>
I.D., in	<u>0.83</u>
Air flow, cfm	<u>16780</u>
Water flow, gpm	<u>151.3</u>
Condenser pressure -	
Upper limit, psia	<u>27.0</u>
Lower limit, psia	<u>9.6</u>
Increment	<u>2.0</u>

DESIGN POINT PARASITICS:

	RC only	AC only	RC & AC
Gearbox loss, hp	<u>1.15</u>	<u>1.15</u>	<u>1.15</u>
Turbine/compressor windage, hp	<u>0.1</u>	<u>0.1</u>	<u>-0-</u>
Controls, kW	<u>0.4</u>	<u>0.2</u>	<u>0.4</u>
Condenser fan, kW	<u>1.03</u>	<u>1.03</u>	<u>1.03</u>
Condenser pump, kW	<u>0.904</u>	<u>0.904</u>	<u>0.904</u>
Windage/vacuum pump, kW	<u>0.1</u>	<u>0.1</u>	<u>-0-</u>
Feed pump (design point) kW	<u>1.2</u>	<u>-0-</u>	<u>1.2</u>
Motor/generator efficiency	<u>95%</u>	<u>95%</u>	<u>95%</u>

INPUT PARAMETERS TO COMPUTER SIMULATIONS
SYSTEM 10 HP. R/C

INPUT PARAMETERS FOR DESIGN POINT CYCLE

RANKINE:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>180.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>95.0</u>
Turbine efficiency	<u>80.0%</u>
Feed pump efficiency	<u>41.0%</u>
Turbine shaft power output, hp	<u>10.0</u>
Pressure drops: psid	
Turbine valve	<u>2.0</u>
Condenser	<u>0.8</u>
Check valve	<u>2.0</u>
Filter	<u>2.0</u>
Control valve	<u>31.0</u>
Preheater	<u>2.0</u>
Evaporator	<u>0.0</u>

AIR CONDITIONER:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>50.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>95.0</u>
Compressor efficiency	<u>80.0%</u>
Cooling required, tons	<u>25.0</u>
Pressure drops: psid	
Compressor inlet	<u>0.38</u>
Compressor exit	<u>0.84</u>

INPUT PARAMETERS FOR THE OFF-DESIGN SIMULATION:

RANKINE:

$\dot{m}_p$ solar, Btu/hr- $^{\circ}$ F	13,722
Preheater surface area, ft ²	50.0
Evaporator surface area, ft ²	200.0
ΔT solar to evaporator, $^{\circ}$ F	20.0
Condenser pressure -	
Upper limit, psia	27.0
Lower limit, psia	9.0
Increment	2.0
Solar collector fluid temperature -	
Upper limit, $^{\circ}$ F	220.0
Lower limit, $^{\circ}$ F	130.0
Increment	10.0
Preheater U design, Btu/hr-ft ² - $^{\circ}$ F	41.7
Evaporator U design, Btu/hr-ft ² - $^{\circ}$ F	107.7
Turbine diameter, in	1.7495
Turbine nozzle area, in ²	0.390

AIR CONDITIONER:

$\dot{m}_p$ chilled water, Btu/hr- $^{\circ}$ F	30,000
Evaporator surface area, ft ²	244.1
Evaporator U design, Btu/hr-ft ² - $^{\circ}$ F	135.0
Chilled water temperature, $^{\circ}$ F	55.0
Condenser pressure -	
Upper limit, psia	27.0
Lower limit, psia	9.0
Increment	2.0
ΔT water to evaporator, $^{\circ}$ F	5.0
Compressor diameter, in	5.05

INPUT PARAMETERS FOR THE CONDENSER SIMULATION:

Height, ft	2.81
Width, ft	3.50
Length, ft	5.67
Number of tubes	19
O.D., in	1.05
I.D., in	0.824
Air flow, cfm	8,500
Water flow, gpm	74.0
Condenser pressure -	
Upper limit, psia	27.0
Lower limit, psia	9.0
Increment	2.0

DESIGN POINT PARASITICS:

	RC only	AC only	RC & AC
Gearbox loss, hp	1.15	1.15	1.15
Turbine/compressor windage, hp	0.1	0.1	- 0 -
Controls, kW	0.4	0.2	0.4
Condenser fan, kW	1.6	1.6	1.6
Condenser pump, kW	0.5	0.5	0.5
Windage vacuum pump, kW	0.1	0.1	0.1
Feed pump (design point) kW	0.6	- 0 -	0.6
Motor/generator efficiency	91%	91%	91%

INPUT PARAMETERS TO COMPUTER SIMULATIONS
SYSTEM 30HP R/C

INPUT PARAMETERS FOR DESIGN POINT CYCLE

RANKINE:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>180.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>95.0</u>
Turbine efficiency	<u>80.0%</u>
Feed pump efficiency	<u>41.0%</u>
Turbine shaft power output, hp	<u>30.0</u>
Pressure drops: psid	
Turbine valve	<u>2.0</u>
Condenser	<u>0.8</u>
Check valve	<u>2.0</u>
Filter	<u>2.0</u>
Control valve	<u>31.0</u>
Preheater	<u>2.0</u>
Evaporator	<u>0.0</u>

AIR CONDITIONER:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>50.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>95.0</u>
Compressor efficiency	<u>80.0%</u>
Cooling required, tons	<u>25.0</u>
Pressure drops: psid	
Compressor inlet	<u>0.38</u>
Compressor exit	<u>0.84</u>

INPUT PARAMETERS FOR THE OFF-DESIGN SIMULATION:

RANKINE:

$\dot{m}_p$ solar, Btu/hr-°F	41,163
Preheater surface area, ft ²	150.0
Evaporator surface area, ft ²	600.0
ΔT solar to evaporator, °F	20.0
Condenser pressure -	
Upper limit, psia	27.0
Lower limit, psia	9.0
Increment	2.0
Solar collector fluid temperature -	
Upper limit, °F	220.0
Lower limit, °F	130.0
Increment	10.0
Preheater U design, Btu/hr-ft ² -°F	41.7
Evaporator U design, Btu/hr-ft ² -°F	107.7
Turbine diameter, in	5.249
Turbine nozzle area, in ²	0.676

AIR CONDITIONER:

$\dot{m}_p$ chilled water, Btu/hr-°F	30,000
Evaporator surface area, ft ²	244.1
Evaporator U design, Btu/hr-ft ² -°F	135.0
Chilled water temperature, °F	55.0
Condenser pressure -	
Upper limit, psia	27.0
Lower limit, psia	9.0
Increment	2.0
ΔT water to evaporator, °F	5.0
Compressor diameter, in	5.05

INPUT PARAMETERS FOR THE CONDENSER SIMULATION:

Height, ft	4.02
Width, ft	3.50
Length, ft	8.67
Number of tubes	19
O.D., in	1.05
I.D., in	0.824
- Air flow, cfm	14,000
Water flow, gpm	113.0
Condenser pressure -	
Upper limit, psia	27.0
Lower limit, psia	9.0
Increment	2.0

DESIGN POINT PARASITICS:

	RC only	AC only	RC & AC
Gearbox loss, hp	1.15	1.15	1.15
Turbine/compressor windage, hp	0.1	0.1	- 0 -
Controls, kW	0.4	0.2	0.4
Condenser fan, kW	3.4	3.4	3.4
Condenser pump, kW	0.7	0.7	0.7
Windage vacuum pump, kW	0.1	0.1	0.1
Feed pump (design point) kW	1.8	1.8	1.8
Motor/generator efficiency	91%	91%	91%

INPUT PARAMETERS TO COMPUTER SIMULATIONS
SYSTEM 45°F CHILLED WATER

INPUT PARAMETERS FOR DESIGN POINT CYCLE

RANKINE:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>180.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>95.0</u>
Turbine efficiency	<u>80.0%</u>
Feed pump efficiency	<u>41.0%</u>
Turbine shaft power output, hp	<u>20.0</u>
Pressure drops: psid	
Turbine valve	<u>2.0</u>
Condenser	<u>0.8</u>
Check valve	<u>2.0</u>
Filter	<u>2.0</u>
Control valve	<u>31.0</u>
Preheater	<u>2.0</u>
Evaporator	<u>0.0</u>

AIR CONDITIONER:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>40.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>95.0</u>
Compressor efficiency	<u>79.0%</u>
Cooling required, tons	<u>25.0</u>
Pressure drops: psid	
Compressor inlet	<u>0.38</u>
Compressor exit	<u>0.84</u>

INPUT PARAMETERS FOR THE OFF-DESIGN SIMULATION:

RANKINE:

$\dot{m}_p$ solar, Btu/hr-°F	27.443
Preheater surface area, ft ²	100.0
Evaporator surface area, ft ²	400.0
ΔT solar to evaporator, °F	20.0
Condenser pressure -	
Upper limit, psia	27.0
Lower limit, psia	9.0
Increment	2.0
Solar collector fluid temperature -	
Upper limit, °F	220.0
Lower limit, °F	130.0
Increment	10.0
Preheater U design, Btu/hr-ft ² -°F	41.7
Evaporator U design, Btu/hr-ft ² -°F	107.7
Turbine diameter, in	3.499
Turbine nozzle area, in ²	0.552

AIR CONDITIONER:

$\dot{m}_p$ chilled water, Btu/hr-°F	
Evaporator surface area, ft ²	
Evaporator U design, Btu/hr-ft ² -°F	
Chilled water temperature, °F	45.0
Condenser pressure -	
Upper limit, psia	27.0
Lower limit, psia	9.0
Increment	2.0
ΔT water to evaporator, °F	5.0
Compressor diameter, in	5.05

INPUT PARAMETERS FOR THE CONDENSER SIMULATION:

Height, ft	<u>3.15</u>
Width, ft	<u>3.25</u>
Length, ft	<u>11.0</u>
Number of tubes	<u>18</u>
O.D., in	<u>1.05</u>
I.D., in	<u>0.83</u>
Air flow, cfm	<u>11,780</u>
Water flow, gpm	<u>151.3</u>
Condenser pressure -	
Upper limit, psia	<u>27.0</u>
Lower limit, psia	<u>9.0</u>
Increment	<u>2.0</u>

DESIGN POINT PARASITICS:

	RC only	AC only	RC & AC
Gearbox loss, hp	<u>1.15</u>	<u>1.15</u>	<u>1.15</u>
Turbine/compressor windage, hp	<u>0.1</u>	<u>0.1</u>	<u>0.1</u>
Controls, kW	<u>0.4</u>	<u>0.2</u>	<u>0.4</u>
Condenser fan, kW	<u>1.03</u>	<u>1.03</u>	<u>1.03</u>
Condenser pump, kW	<u>0.904</u>	<u>0.904</u>	<u>0.904</u>
Windage vacuum pump, kW	<u>0.1</u>	<u>0.1</u>	<u>- 0 -</u>
Feed pump (design point) kW	<u>1.2</u>	<u>- 0 -</u>	<u>1.2</u>
Motor/generator efficiency	<u>91%</u>	<u>91%</u>	<u>91%</u>

INPUT PARAMETERS TO COMPUTER SIMULATIONS
SYSTEM 170°F BNL CYCLE

INPUT PARAMETERS FOR DESIGN POINT CYCLE

RANKINE:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>150.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>95.0</u>
Turbine efficiency	<u>83.0%</u>
Feed pump efficiency	<u>44.0%</u>
Turbine shaft power output, hp	<u>20.0</u>
Pressure drops: psid	
Turbine valve	<u>2.0</u>
Condenser	<u>0.8</u>
Check valve	<u>2.0</u>
Filter	<u>2.0</u>
Control valve	<u>33.0</u>
Preheater	<u>2.0</u>
Evaporator	<u>0.0</u>

AIR CONDITIONER:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>50.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>95.0</u>
Compressor efficiency	<u>80.0%</u>
Cooling required, tons	<u>25.0</u>
Pressure drops: psid	
Compressor inlet	<u>0.38</u>
Compressor exit	<u>0.84</u>

INPUT PARAMETERS FOR THE OFF-DESIGN SIMULATION:

RANKINE:

$\dot{m}_p$ solar, Btu/hr- $^{\circ}$ F	40,142
Preheater surface area, ft ²	81.2
Evaporator surface area, ft ²	793.7
ΔT solar to evaporator, $^{\circ}$ F	20.0
Condenser pressure -	
Upper limit, psia	29.0
Lower limit, psia	9.0
Increment	2.0
Solar collector fluid temperature -	
Upper limit, $^{\circ}$ F	190.0
Lower limit, $^{\circ}$ F	150.0
Increment	10.0
Preheater U design, Btu/hr-ft ² - $^{\circ}$ F	80.0
Evaporator U design, Btu/hr-ft ² - $^{\circ}$ F	100.0
Turbine diameter, in	2.787
Turbine nozzle area, in ²	1.267

AIR CONDITIONER:

$\dot{m}_p$ chilled water, Btu/hr- $^{\circ}$ F	30,000
Evaporator surface area, ft ²	244.1
Evaporator U design, Btu/hr-ft ² - $^{\circ}$ F	135.0
Chilled water temperature, $^{\circ}$ F	55.0
Condenser pressure -	
Upper limit, psia	29.0
Lower limit, psia	9.0
Increment	2.0
ΔT water to evaporator, $^{\circ}$ F	5.0
Compressor diameter, in	5.05

INPUT PARAMETERS FOR THE CONDENSER SIMULATION:

Height, ft	<u>3.15</u>
Width, ft	<u>3.25</u>
Length, ft	<u>14.4</u>
Number of tubes	<u>18</u>
O.D., in	<u>1.05</u>
I.D., in	<u>0.824</u>
Air flow, cfm	<u>15,330</u>
Water flow, gpm	<u>197.0</u>
Condenser pressure -	
Upper limit, psia	<u>29.0</u>
Lower limit, psia	<u>9.0</u>
Increment	<u>2.0</u>

DESIGN POINT PARASITICS:

	RC only	AC only	RC & AC
Gearbox loss, hp	<u>1.15</u>	<u>1.15</u>	<u>1.15</u>
Turbine/compressor windage, hp	<u>0.1</u>	<u>0.1</u>	<u>-0-</u>
Controls, kW	<u>0.4</u>	<u>0.2</u>	<u>0.4</u>
Condenser fan, kW	<u>1.18</u>	<u>1.18</u>	<u>1.18</u>
Condenser pump, kW	<u>1.34</u>	<u>1.34</u>	<u>1.34</u>
Windage vacuum pump, kW	<u>0.1</u>	<u>0.1</u>	<u>-0-</u>
Feed pump (design point) kW	<u>1.25</u>	<u>-0-</u>	<u>1.25</u>
Motor/generator efficiency	<u>91%</u>	<u>91%</u>	<u>91%</u>

INPUT PARAMETERS TO COMPUTER SIMULATIONS SYSTEM BOILING IN COLLECTOR

INPUT PARAMETERS FOR DESIGN POINT CYCLE

RANKINE:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>280.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>95.0</u>
Turbine efficiency	<u>76.7%</u>
Feed pump efficiency	<u>44.5%</u>
Turbine shaft power output, hp	<u>22.0</u>
Pressure drops: psid	
Turbine valve	<u>2.0</u>
Condenser	<u>0.8</u>
Check valve	<u>2.0</u>
Filter	<u>2.0</u>
Control valve	<u>20.3</u>
Preheater	<u>2.0</u>
Evaporator	<u>0.0</u>

AIR CONDITIONER:

$T_{\text{evaporator}}, ^\circ\text{F}$	<u>50.0</u>
$T_{\text{condenser}}, ^\circ\text{F}$	<u>95.0</u>
Compressor efficiency	<u>80.0%</u>
Cooling required, tons	<u>25.0</u>
Pressure drops: psid	
Compressor inlet	<u>0.38</u>
Compressor exit	<u>0.84</u>

INPUT PARAMETERS FOR THE OFF-DESIGN SIMULATION:

RANKINE:

$\dot{m}_p$ solar, Btu/hr- $^{\circ}$ F	17,179
Preheater surface area, ft ²	50.0
Evaporator surface area, ft ²	200.0
ΔT solar to evaporator, $^{\circ}$ F	20.0
Condenser pressure -	
Upper limit, psia	29.0
Lower limit, psia	9.0
Increment	2.0
Solar collector fluid temperature -	
Upper limit, $^{\circ}$ F	310.0
Lower limit, $^{\circ}$ F	230.0
Increment	10.0
Preheater U design, Btu/hr-ft ² - $^{\circ}$ F	54.3
Evaporator U design, Btu/hr-ft ² - $^{\circ}$ F	74.7
Turbine diameter, in	3.533
Turbine nozzle area, in ²	0.1053

AIR CONDITIONER:

$\dot{m}_p$ chilled water, Btu/hr- $^{\circ}$ F	30,000
Evaporator surface area, ft ²	244.1
Evaporator U design, Btu/hr-ft ² - $^{\circ}$ F	135.0
Chilled water temperature, $^{\circ}$ F	55.0
Condenser pressure -	
Upper limit, psia	29.0
Lower limit, psia	9.0
Increment	2.0
ΔT water to evaporator, $^{\circ}$ F	5.0
Compressor diameter, in	5.05

INPUT PARAMETERS FOR THE CONDENSER SIMULATION:

Height, ft	3.14
Width, ft	3.25
Length, ft	8.0
Number of tubes	19
O.D., in	1.05
I.D., in	0.824
Air flow, cfm	9,300
Water flow, gpm	154.0
Condenser pressure -	
Upper limit, psia	29.0
Lower limit, psia	9.0
Increment	2.0

DESIGN POINT PARASITICS:

	RC only	AC only	RC & AC
Gearbox loss, hp	1.15	1.15	1.15
Turbine/compressor windage, hp	0.1	0.1	-0-
Controls, kW	0.4	0.2	0.4
Condenser fan, kW	0.795	0.795	0.795
Condenser pump, kW	0.904	0.904	0.904
Windage vacuum pump, kW	0.1	0.1	-0-
Feed pump (design point) kW	1.65	-0-	1.65
Motor/generator efficiency	91%	91%	91%

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

HERM 695

BY

Cindy C

DATE

4/9/82

SUBJECT

- Tabulated Data -
Refrigeration Cycle Data

HIGH TEMP
EVAPORATIVE CONDENSER
T_{cond} design = 95°F

P cond Condenser Pressure psia	T _{wb} Wet Bulb Temperature (°F)	Kw Power Supplied To System	Tonage (tons)
9	40.23	- 6.82	29.66
11	49.62	- 9.35	29.02
13	57.84	- 11.22	28.46
15	65.08	- 12.73	27.96
17	71.63	- 14.11	27.50
19	77.77	- 15.21	26.82
21	83.62	- 15.84	25.44
23	89.60	- 15.67	23.08

parasitics:

gearbox loss

turbine pumping loss + associated compressor

motor inefficiencies

evaporative condenser fan & pump

control power

BARBER-NICHOLS
ENGINEERING CO.

PROJECT
HERM - 695

BY
G. SANDERS

DATE
5/5/82

SUBJECT
'BIC' Model
R.C. only data
T AMBIENT = 70 F

187551²

CONDENSED PRESSURE PSIA	INSULATION lb/hr-sq ft	WET Bulb Temp F	Net Power Output kW	Total Heat Input Btu/hr
9	150	46.5	3.95	189,500
	175	45.7	5.50	262,000
	200	44.9	7.00	256,000
	225	44.1	8.45	288,000
	250	43.3	9.89	320,000
	275	42.5	11.29	351,500
	300	41.7	12.65	383,500
	325	41.0	13.95	414,500
	350	—	—	—
11	150	55.9	3.70	190,500
	175	55.1	5.28	223,500
	200	54.2	6.74	255,500
	225	53.5	8.20	288,000
	250	52.7	9.60	319,000
	275	51.9	11.00	351,000
	300	51.1	12.36	382,500
	325	50.3	13.75	415,000
	350	—	—	—
13	150	63.9	3.30	191,000
	175	63.2	4.90	224,000
	200	62.4	6.40	255,000
	225	61.7	7.86	288,000
	250	60.9	9.30	319,000
	275	60.1	10.70	351,000
	300	59.3	12.05	382,500
	325	58.5	13.37	415,000
	350	—	—	—

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

HERM - 695

BY

G. SANDERS

DATE

5/5/82

SUBJECT

'BIC' Model
R.C. only data

T AMBIENT = 70 F

CONDENSED PRESSURE PSIA	INSULATION Btu/hr.-sq.ft	WET Bulb Temp F	Net Power Input kW	Total Heat Input Btu/hr
15	150	71.2	2.80	191,000
	175	70.4	4.50	224,000
	200	69.7	6.08	254,500
	225	68.9	7.55	285,500
	250	68.1	9.00	318,000
	275	67.3	10.40	349,500
	300	66.5	11.74	381,500
	325	65.7	13.10	413,000
	350	—	—	—
17	150	77.6	2.30	190,500
	175	76.8	4.09	222,500
	200	76.0	5.71	254,000
	225	75.2	7.28	287,000
	250	74.5	8.74	319,000
	275	73.7	10.14	354,500
	300	72.9	11.50	381,000
	325	—	—	—
	350	—	—	—
19	150	83.3	1.78	141,500
	175	82.7	3.60	223,000
	200	82.0	5.29	254,500
	225	81.1	6.89	286,500
	250	80.4	8.40	319,000
	275	79.5	9.86	351,000
	300	78.7	11.30	384,000
	325	—	—	—
	350	—	—	—

BARBER-NICHOLS
ENGINEERING CO.

PROJECT
HERM - 695

BY
G. SANDER

DATE
5/5/82

SUBJECT
'BIC' Model
R.C. only data
T AMBIENT = 70 F

CONDENSED PRESSURE PSIA	INSULATION Btu/hr.-sq.ft.	WET Bulb Temp F	NET POWER Output kW	Total Heat Input Btu/hr
21	150	88.9	1.39	190,000
	175	88.1	3.10	223,000
	200	87.4	4.74	255,000
	225	86.6	6.40	287,000
	250	85.8	8.01	329,000
	275	85.0	9.52	350,500
	300	84.1	10.98	381,500
	325	—	—	—
	350	—	—	—
23	150	94.0	0.92	189,000
	175	93.2	2.50	219,500
	200	92.5	4.12	253,000
	225	91.7	5.80	285,000
	250	90.9	7.48	319,000
	275	90.1	9.05	351,500
	300	89.3	10.60	383,000
	325	—	—	—
	350	—	—	—
	150			
	175			
	200			
	225			
	250			
	275			
	300			
	325			
	350			

BARBER-NICHOLS
ENGINEERING CO.

PROJECT
HERM - 695

BY
G SANDERS

DATE
5/5/82

SUBJECT
'BIC' Model 1875 8+2
R.C. only data
T AMBIENT = 100F

CONDENSED PRESSURE PSIA	INNOVATION btu/hr-sq-ft	WET Bulb Temp F	Net Power kW	Total Heat kW/hr
9	150	46.4	4.00	190,500
	175	45.6	5.56	223,500
	200	44.8	7.02	256,500
	225	44.1	8.48	288,500
	250	43.2	9.90	320,500
	275	42.5	11.30	352,000
	300	41.7	12.68	384,000
	325	40.9	13.99	415,000
	350	—	—	—
11	150	55.9	3.76	191,500
	175	55.0	5.29	224,000
	200	54.2	6.75	256,000
	225	53.5	8.18	287,500
	250	52.6	9.60	319,000
	275	51.9	11.01	351,500
	300	51.0	12.40	382,500
	325	50.2	13.77	416,500
	350	—	—	—
13	150	63.9	3.31	191,000
	175	63.1	4.93	224,500
	200	62.4	6.42	243,000
	225	61.6	7.90	277,000
	250	60.9	9.32	318,500
	275	60.0	10.71	359,000
	300	59.2	12.10	388,500
	325	58.4	13.46	414,500
	350	—	—	—

BARBER-NICHOLS
ENGINEERING CO.

PROJECT
HERM - 695

BY
G SANDERS

DATE
5/5/62

SUBJECT
'BIC' Model
R.C. only data
T AMBIENT = 100F

CONDENSER PRESSURE PSIA	INSULATION Btu/hr.-sqft	WET Bulb Temp F	Net Power Output KW	Total Heat Input Btu/hr
15	150	71.1	2.85	196,500
	175	70.4	4.50	224,000
	200	69.6	6.17	256,000
	225	68.9	7.62	287,500
	250	68.1	9.05	319,000
	275	67.3	10.45	350,500
	300	66.4	11.83	383,000
	325	—	—	—
	350	—	—	—
17	150	77.6	2.30	190,500
	175	76.8	4.10	222,500
	200	76.0	5.78	253,000
	225	75.2	7.30	287,500
	250	74.5	8.74	319,500
	275	73.7	10.17	350,500
	300	72.9	11.52	382,000
	325	—	—	—
	350	—	—	—
19	150	83.4	1.80	190,000
	175	82.7	3.57	222,000
	200	81.9	5.30	254,000
	225	81.1	6.89	286,000
	250	80.3	8.39	319,000
	275	79.5	9.84	351,000
	300	78.8	11.28	384,000
	325	—	—	—
	350	—	—	—

BARBER-NICHOLS
ENGINEERING CO.

PROJECT
HERM - 695

BY
G. SANDERS

DATE
5/5/82

SUBJECT
"BIC" Model
R.C. only data
T AMBIENT = 100F

CONDENSER PRESSURE PSIA	INSULATION Btu/hr.-sqft	WET Bulb Temp F	Net Power Output KW	Total Heat Input Btu/hr
21	150	88.9	1.34	189,500
	175	88.2	3.00	221,000
	200	87.4	4.78	259,500
	225	86.6	6.44	288,000
	250	85.8	8.01	320,000
	275	85.0	9.50	359,000
	300	84.2	10.96	381,500
	325	—	—	—
	350	—	—	—
23	150	94.0	0.91	188,500
	175	93.2	2.50	220,000
	200	92.4	4.21	253,500
	225	91.6	5.95	288,000
	250	90.9	7.58	324,000
	275	90.1	9.11	353,000
	300	89.3	10.62	384,000
	325	—	—	—
	350	—	—	—
	150			
	175			
	200			
	225			
	250			
	275			
	300			
	325			
	350			

**BARBER-NICHOLS
ENGINEERING CO.**

**PROJECT
HERM - 695**
**BY
C. Myers**
**DATE
5/5/82**
**SUBJECT
Tabulated Data for SAI
'BIC Model'
RC & AC Combined - 100°F Ambient**

Condenser Pressure (psia)	Solar Insolation (Btu/Hr-Ft ²)	Wet Bulb Temperature (°F)	Net Power Supplied To Grid (kw)	Total Heat Input (Btu/Hr)
9 Tonnage = 29.66 tons	150	—	—	—
	175	34.45	2.68	224500
	200	33.6	4.15	256500
	225	32.8	5.65	289500
	250	32.0	7.05	320000
	275	31.2	8.50	351000
	300	30.3	9.90	384500
	325	29.5	11.20	415000
	350	—	—	—
11 Tonnage = 29.02 tons	150	44.6	-1.40	191000
	175	43.8	0.15	224000
	200	42.9	1.65	256000
	225	42.1	3.15	288500
	250	41.3	4.60	320000
	275	40.5	6.05	352500
	300	39.7	7.45	384500
	325	38.9	8.80	415000
	350	—	—	—
13 Tonnage = 28.46 tons	150	52.8	-3.60	191500
	175	51.9	-1.85	223000
	200	51.1	-0.25	255000
	225	50.3	1.30	288000
	250	49.5	2.75	319000
	275	48.7	4.20	352000
	300	47.9	5.60	383500
	325	47.0	7.00	415500
	350	—	—	—
15 Tonnage = 27.96 tons	150	59.9	-5.45	191000
	175	59.1	-3.55	222500
	200	58.3	-1.85	254500
	225	57.5	-0.25	287500
	250	56.8	1.20	318500
	275	55.9	2.65	350500
	300	55.1	4.05	382000
	325	—	—	—
	350	—	—	—

BARBER-NICHOLS
ENGINEERING CO.

PROJECT
HERM - 695

BY
C. Myers

DATE
5/5/82

SUBJECT
Tabulated Data for SAI
'BIC Model"
RC & AC Combined - 100°F Ambient

Condenser Pressure (psia)	Solar Insolation (Btu/Hr-Ft ²)	Wet Bulb Temperature (°F)	Net Power Supplied To Grid (kw)	Total Heat Input (Btu/Hr)
17 Tonnage = 27.50 tons	150	66.5	-7.30	190500
	175	65.6	-5.30	222500
	200	64.9	-3.45	254000
	225	64.0	-1.75	286000
	250	63.2	-0.20	318500
	275	62.4	1.25	350500
	300	61.5	2.70	383500
	325	—	—	—
	350	—	—	—
19 Tonnage = 26.82 tons	150	72.4	-8.80	192000
	175	71.6	-6.85	224000
	200	70.9	-4.90	256000
	225	70.1	-3.10	288500
	250	69.3	-1.45	319000
	275	68.4	0.05	350000
	300	67.6	1.50	381500
	325	—	—	—
	350	—	—	—
21 Tonnage = 25.44 tons	150	78.3	-9.95	—
	175	77.5	-8.00	223000
	200	76.7	-6.05	253000
	225	75.9	-4.15	286000
	250	75.1	-2.40	319500
	275	74.3	-0.80	352000
	300	73.5	0.70	383000
	325	—	—	—
	350	—	—	—
23 Tonnage = 23.08 tons	150	84.5	-11.0	—
	175	83.4	-8.45	222000
	200	82.6	-6.55	252500
	225	81.8	-4.60	286000
	250	81.0	-2.80	318500
	275	80.3	-1.15	350000
	300	79.4	0.45	381000
	325	—	—	—
	350	—	—	—

APPENDIX B

Air Cooled Condenser Calculations

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

HERM-695

SUBJECT

Air cooled
condenserBY
S.O.DATE
2-22-82

High temp. HERM (INPUT)

$$Q = 269,738$$

$$k_{er} = 0.0485$$

$$t_c = 100.1^\circ\text{F}$$

$$M_{LR} = 0.894$$

$$S_L = 90.2$$

$$S_v = .586$$

$$\dot{W}_R = 3434 \text{ Lb/hr}$$

Core geometry

fin material - aluminum
 $k_m = 100$

$$\text{MOTOR HP} = 3 \text{ HP}$$

$$\text{FAN-MOTOR efficiency} = .5$$

$$C_{p_{A.R}} = .24$$

$$K_{air} = .015$$

$$M_{air} = .045$$

$$\text{fouling factor } R = .001$$

$$\text{Tube O.D.} = 5/8 \text{ inch}$$

$$\text{Tube I.D.} = .555 \text{ inch}$$

NOTE: CONVERT ALL INPUT

VARIABLES TO FT, °F, Lb, HR, BTU units

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

HERM-695

SUBJECT

Air cooled
condenser

BY

S.O

DATE

2-22-82

1. $A_f^* = (S_1 \cdot S_2 - \pi \cdot O.D.^2/4)$
2. $A_T^* = A_f^* + \pi \cdot O.D. (P - \delta_{fin})$
3. $A_{C_1}^* = (S_1 - O.D.) (P - \delta_{fin})$
4. $A_{C_2}^* = (\sqrt{S_1^2/4 + S_2^2} - O.D.) (P - \delta_{fin})$
5. $A_c^* = A_{C_1}^*$ minimum
6. $A_{fin}/A_{TOT} = \delta_{fin}/A_T^*$
7. $G = A_c^*/P \cdot S_1$
8. $\ell_{fin} = \sqrt{\frac{S_1 \cdot S_2}{\pi}} - O.D./2$
9. $r_o/r_i = \frac{2 \ell_{fin} + O.D.}{O.D.}$
10. $r_h = S_2 \cdot A_c^*/A_T^*$
11. $A_{out}/A_{in} = A_T^*/\pi \cdot P \cdot I.D.$
12. $j_a = .431 \left(\frac{S_1 - O.D.}{S_2 - O.D.} \right)^{-.139} \left(\frac{P}{\delta_{fin}} \right)^{-.223} \left(\frac{P}{d} \right)^{.581}$
13. $f_a = .0814 \left(\frac{S_1 - O.D.}{S_2 - O.D.} \right)^{.35} \left(\frac{P}{\delta_{fin}} \right)^{.476} \left(\frac{P}{d} \right)^{.162}$
14. $\alpha = G/V_h$
15. $V = HBL$
16. $A_{out} = \alpha V$
17. $h_c = .4561 \left(\frac{q'' \cdot P_L (P_L - P_U) k_L^3 A_{out}}{M_L \cdot I.D. A_{out}/A_{in} \cdot W_R} \right)^{1/3}$
- 17A. $A_c = GHB$

4.18E8

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

HERM-695

BY

S.O.

DATE

2-22-82

SUBJECT

Air cooled
condenser18. Assume w_{AIR}

$$19. P_{AIR} = \frac{C_{PA} \cdot M_A}{k_A}$$

$$20. G_A = \frac{w_{AIR}}{A_C}$$

$$21. Re_A = \frac{4 \rho_A G_A}{\mu_{AIR}}$$

$$22. j = j_A Re_A^{-.364}$$

$$23. h_A = C_{PA} G_A \cdot d / P_{AIR}^{2/3}$$

$$24. C_{\#ROWS} = \left[.6 \cdot (SGN \# ROWS) + .7 \cdot (SGN(\#-1)) + .9 \left(\frac{SGN(\#-2.5)+1}{2} \right) + \right. \\ \left. + (\#-3) \left(\frac{SGN(\#-3.5)+1}{2} \right) \right] / \#ROWS$$

NOTE: $SGN \ i = -1$ if $i < 0$
 0 if $i = 0$
 1 if $i > 0$

$$25. m_e = \sqrt{\frac{2 h_A}{k_m \cdot \delta_{FIN}}} \ell_{FIN}$$

$$26. \eta_f = \frac{\tanh(m_e)}{m_e} \left\{ 2 - (r_o/r_i)^{-.035} + \left[(r_o/r_i)^{-.19} - 1 \right] (m_e) \right\}$$

$$27. \eta_o = 1 - A_{FIN} / A_{TOT} (1 - \eta_f)$$

$$28. U = \frac{1}{\left(\frac{1}{h_c} + R \right) \frac{A_{WT}}{A_{FIN}} + \frac{1}{h_A \cdot \eta_o \cdot C_{\#ROWS}}}$$

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

HERM-695

BY

S. J.

DATE

2-22-82

SUBJECT

Air cooled
condenser

$$29. \quad NTU = \frac{U \cdot A_{out}}{C_{PA} \cdot \dot{W}_{AIR}}$$

$$30. \quad \epsilon = 1 - e^{-NTU}$$

$$31. \quad t_{AIR,IN} = t_c - \frac{Q}{\epsilon \cdot C_{PA} \dot{W}_A}$$

$$32. \quad t_{AIR,OUT} = t_{AIR,IN} + \frac{Q}{C_P \cdot \dot{W}_{AIR}}$$

$$33. \quad V_{AIR} = \frac{1545 (t_{AIR,OUT} + 460)}{29 \cdot P_{ATM}(\text{PSIA}) \cdot 144}$$

$$34. \quad \Delta P_{AIR}(\text{PSID}) = \frac{G_A^2 \cdot V_{AIR}}{29 \cdot 144} f_a K_{EA} \left(\frac{A_{out}}{A_c} \right)^{-0.282}$$

$$35. \quad \dot{W}_{AIR}^* = \frac{35,930 \cdot P_{ATM}(\text{PSIA}) \cdot \eta_{FM} \cdot HP}{\Delta P_{SID} (t_{AIR,IN} + 460)}$$

$$36. \quad \text{IF } \dot{W}_{AIR}^* \neq \dot{W}_{AIR} \pm \frac{\dot{W}_{AIR}}{1000} \quad \text{RETURN TO STEP 18}$$

$$37. \quad \text{PRINT } t_{AIR,IN}$$

BARBER-NICHOLS
ENGINEERING CO.

PROJECT

HERM-695

SUBJECT

Air cooled
Condenser

BY

S.O.

DATE

2-22-82

Sample calculation

1. $A_f^* = .0228$
2. $A_r^* = .0240$
3. $A_{c1}^* = .0729$
4. $A_{c2}^* = .0729$
5. $A_c^* = .0729$
6. $\frac{A_{fin}}{A_{tot}} = .95$
7. $G = .513$
8. $\ell_{fin} = .03959 \text{ FT}$
9. $r_o/r_i = 2.52$
10. $r_h = .002412 \text{ FT}$
11. $\frac{A_{out}}{A_{in}} = 19.823$
12. $j_a = .08933$
13. $f_a = .1818$
14. $\alpha = 212.85$
15. $V = 21.65$
16. $A_{out} = 4608.4 \text{ FT}^2$
17. $h_c = 391.23$
- 17A. $A_c = 25.67 \text{ FT}^2$
18. $\dot{W}_{air} = 122,025 \text{ lb/hr}$
19. $Pr_{air} = .72$
20. $G_A = 4753.3$
21. $Re_A = 1019$
22. $j = .00536$
23. $h_A = 10.236$
24. $C_{\# \text{ row}} = .8$
25. $ml = .258$
26. $\eta_t = .851$
27. $\eta_o = .8589$
28. $U = 4.702$
29. $NTU = .74$
30. $\epsilon = .5229$
31. $t_{air \text{ in}} = 82.48$
32. $t_{air \text{ out}} = 91.63$
33. $V_{air} = 13.773$
34. $\Delta P_{air} = .01197 \text{ psid}$
35. $\dot{W}_{air}^* = 122,053 \text{ lb/hr}$