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STUDY OF TWO-PHASE TURBINE ENGINE FOR SOLAR SPACE COOLING
Final Report

By
William E. Amend

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Biphase Energy Systems
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F I N A L R E P O R T

For

STUDY OF TWO-PHASE TURBINE ENGINE
FOR SOLAR SPACE COOLING

Contract No. EG-77-C-03-1542

Prepared for the
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Prepared by:

William E. Amend
Project Manager

Approved by:

Lance Hays
President

Biphase Energy Systems
Santa Monica, California

TABLE OF CONTENTS

<u>Section</u>	<u>Title</u>	<u>Page</u>
I	ABSTRACT	1
II	INTRODUCTION	3
III	DESCRIPTION OF TWO-PHASE TURBINE	6
IV	CYCLE DESCRIPTIONS	14
V	CYCLE STUDIES	18
VI	DESIGN STUDY	42
VII	CONCLUSIONS.	52
APPENDIX A -	Cycle Analysis	53
APPENDIX B -	Design Calculations.	59
APPENDIX C -	Fluid Compatibility Tests.	69
APPENDIX D -	Applications Study	71
APPENDIX E -	Nomenclature	76
REFERENCES		78

LIST OF FIGURES

<u>Figure</u>	<u>Title</u>	<u>Page</u>
1	Exit Velocity of Two-Phase Nozzles	7
2	Schematic of Flow in Two-Phase Turbine	8
3	Schematic of Two-Phase Turbine Wheel and Blading	9
4	Separator-Turbine with Radial Inflow Liquid Turbine.	11
5	Separator-Turbine with U-Tube Liquid Turbine	12
6	Combined Cycles.	15
7	Schematic of Separator Cycle	16
8	Performance of Combined Cycle for Various Turbine Fluids with R113	20
9	Performance of Combined Cycle for Various Turbine Fluids with R216	21
10	Effect of Compressor Efficiency on Performance of Integrated Cooling Cycle	22
11	Effect of Turbine Efficiency on Performance of Integrated Cooling Cycle.	23
12	Effect of Nozzle Entrance Temperature on Performance of Integrated Cycle for Therminol 66 + R216.	25
13	Effect of Nozzle Entrance Temperature on Performance of Integrated Cycle for Therminol 66 + R113.	26
14	Effect of Condenser Temperature on Performance of Integrated Cycle.	27
15	Effect of Evaporator Temperature on Performance of Integrated Cycle.	28
16	Effect of Nozzle Entrance Temperature on Performance of Separate Cycle.	30
17	Effect of Condenser Temperature on Performance of Separate Cycle.	31
18	Effect of Evaporator Temperature on Performance of Separate Cycle.	32
19	Effect of Compressor Efficiency on Performance of Separate Cycle.	33
20	Effect of Turbine Efficiency on Performance of Integrated Cycle.	34
21	Effect of Condenser Temperature on Performance of Separate Cycle.	35
22	Projected Performance for Integrated Cooling Cycle	36

LIST OF FIGURES (Cont'd)

<u>Figure</u>	<u>Title</u>	<u>Page</u>
23	Projected Performance for Separate Cooling Cycle.	37
24	Conceptual Configuration of Biphasic Turbine for Solar Cooling Applications.	39
25	Typical Performance Curves for a Small Induction Motor/Generator	41
26	Conceptual Layout of Biphasic Solar Cooling System	43
27	Design of Biphasic Solar Cooling Turbine	44
28	Mixer Design.	45
29	Nozzle Definition	47
D-1	Capital Cost (OEM) of Biphasic Solar Cooling System as a Function of Collector Cost.	74
D-2	Annual Operating Expenses of 3 Ton Cooling System Assuming an Annual Interest Rate of 8%	75

STUDY OF TWO-PHASE TURBINE ENGINE FOR SOLAR SPACE COOLING

I. ABSTRACT

Solar cooling can be achieved by Rankine-powered vapor compression, absorption or dessicant processes. The performance of all three systems are projected to be at about the same level and all three systems require complex and expensive auxiliary power and/or energy storage units. The Biphasic turbine offers an alternative prime mover for the vapor-compression system that promises higher performance, longer life, lower cost and cheaper and simpler auxiliary power systems.

In conventional vapor-compression systems, electric motors rotating at 1800 to 3600 RPM are used to drive the compressor. The Biphasic turbine is a new type of turbine that has a high torque and low shaft speed that can correspond directly to the compressor speed. Thus the problem of matching the speeds of the prime mover and the compressor is eliminated. In addition a synchronous motor can be directly connected to the turbine output shaft to supply auxiliary power.

Detailed mathematical description of two promising Biphasic refrigeration cycles were developed and programmed on the computer (all known first-order irreversibilities were accounted for). Extensive parameter sweeps were made to identify the most effective working-fluid combinations and to determine the sensitivity of cycle-performance levels.

A Cycle configuration was established for a nominal 3-ton air cooled refrigeration system and the design parameters were determined from the computer code. A two-phase turbine as a prime

mover for this system was designed to operate at a constant nominal shaft speed of 3600 rpm with supplementary power supplied by a synchronous AC motor.

Finally a series of fluid compatibility tests were run to weed out potential fluid combinations that are reactive.

II. INTRODUCTION

During the past few years significant efforts have been under way to develop active solar-powered cooling systems for buildings. Most of the work has centered around Rankine-powered compression systems, absorption chillers, and dessicant systems. The projected performance levels of all three of these concepts are about the same level.

The Biphase turbines form a new class of prime movers that produce high torque at low rpm. It is possible to design a Biphase turbine that can operate at 1800 or 3600 rpm and thus can be directly coupled to the compressor in a conventional vapor-compression refrigeration system. In addition supplementary power can be supplied to the turbine by connecting an induction electric motor to the turbine shaft. Thus, when the solar insolation decreases below the design level the synchronous motor automatically picks up the power deficit to keep the cooling system operating at the design load. Furthermore, the low speed and low-temperature design makes it possible to use conventional materials, bearings and seals and precludes the need for close clearances and tolerances. This makes commercialization more feasible because off-the-shelf hardware can be used.

First cost is always a consideration in commercially feasible products. Elimination and/or combination of components is therefore very important. Typical solar systems use solar energy to heat a low-vapor-pressure fluid which remains liquid in the collector. Because conventional Rankine cycle type power generators must use a vapor, the heat must be transferred to a higher-vapor pressure fluid so that boiling can occur. The Biphase cycle, however, uses both a low-vapor-pressure and a higher-vapor-pressure fluid so that the collector fluid can be used directly

in the engine, in effect providing heat transfer between the fluids when they are mixed. The boiler is thereby eliminated reducing capital costs and complexity.

The majority of the capital cost of a solar-powered cooling system is tied up in the collector and energy-augmentation systems. Thus it is important to have a system with a high COP (Refrigeration/Energy input) and simple auxiliary power-input system. The Biphase solar-cooling system has the potential to satisfy these two important criteria. In addition the Biphase system has the potential to be highly reliable since the Biphase turbine is a low-speed prime mover requiring no close tolerances. Thus the Biphase solar cooling system has a potential clear-cut advantage over the other solar cooling systems.

This report describes the results of a study that was undertaken to evaluate the Biphase turbine for solar cooling applications. The objectives of this work were as follows:

1. Identify and evaluate Biphase Solar Cooling cycles and working fluid combinations.
2. Develop a conceptual configuration of a 3-ton Biphase Solar Cooling system.
3. Design a Biphase turbine to power the Solar Cooling System.
4. Perform an application study including commercialization potential and economic analysis.

The operation of the Biphase turbine is described in Section III along with its salient features. The two most promising cycles for the Biphase solar cooling system are described in Section IV (the mathematical analysis of these cycles is presented in Appendix A), while the results of parameter studies performed on these cycles are presented in Section V.

Section VI deals with the conceptual configuration of a nominal three ton air-cooled Biphasic solar cooling system. The important features of the system are discussed and the detail design of a Biphasic turbine for a prime mover in this system is presented (the design procedures and detailed calculations are summarized in Appendix B, the results of fluid compatibility tests are presented in Appendix C and an application study is summarized in Appendix D). Finally the conclusions drawn from this work are presented in Section VII.

III. DESCRIPTION OF TWO-PHASE TURBINE

The two elements responsible for the unique characteristics of the two-phase turbine engine are the two-phase nozzle and the turbine wheel. The nozzle produces efficient conversion of liquid enthalpy to vapor and liquid kinetic energy. This is achieved by mixing a finely dispersed hot liquid of low vapor pressure (such as oil) with a liquid possessing higher vapor pressure (such as water or Freon). The high-vapor-pressure fluid vaporizes at high pressure and is subsequently expanded to low pressure and high velocity, dragging the oil droplets to a high velocity. The presence of large amounts of finely dispersed liquid has a similar effect on the nozzle-exit velocity as a large increase in molecular weight of the vapor would, i.e. a high mass flow rate and low spouting velocity are obtained. Figure 1 shows typical nozzle exit velocities as a function of the ratio of liquid to gas. Starting at a value of about 4,000 ft/s for all gas, the exit velocity is reduced to less than 1,000 ft/s for a mass ratio of 30. Thus, a much lower turbine-wheel speed can be used for maximum efficiency than for the vapor alone.

The relation of the nozzle and turbine wheel for a single stage, impulse circumferential configuration is shown in Figures 2 and 3. The geometry can be very similar to a simple Pelton Wheel. The only difference is that the initial impingement zone is inclined at a more gradual angle than the Pelton Wheel to avoid two-phase impact losses; and the surface area can be made less than that of a Pelton Wheel because of the concentration of the liquid phase after impingement. Another geometry that can be used is very similar to a conventional axial-flow impulse turbine.

A variation of the above turbine, which offers elimination of the primary oil pump and may have higher efficiency for some applications, is a rotary-separator turbine. Here the two-phase mixture is impinged on a rotating separator surface to obtain a high-velocity liquid annulus. The liquid flows into a radial

Figure 1. Exit Velocity of Two-Phase Nozzles

Figure 2. Schematic of Flow in Two-Phase Turbine

Figure 3. Schematic of Two-Phase Turbine Wheel and Blading

inflow turbine, which in its simplest case, is a rotating pitot tube and diffuser. The liquid is returned through the heat exchanger to the nozzle. Another version is the use of a simple "U-Tube" to recover the liquid energy.

Figures 4 and 5 illustrate these turbine designs, consisting of nozzle(s), rotating separator, and liquid turbine. The liquid and gas mixture comprising the working fluid is supplied at high pressure to the nozzle inlets. The mixture expands to low pressure at the nozzle exits, and the resulting high-velocity two-phase jet impinges on the inner surface of the rim of the rotating separator. The liquid becomes concentrated in a layer on the inner surface due to the inertia of the liquid and centrifugal force, while the gas flows radially inward through passages and enters the gas discharge pipe through ports in the stationary housing. The rotating separator is supported by bearings mounted in the housing. The rotation of the separator is impeded only by windage and bearing-friction losses which can be very small. Thus, only a very small relative velocity between the impinging jets and the surface, aided by the torque imparted to the rotating separator by the inward flow of the gas through passages, serves to maintain the speed of the liquid layer at a value nearly equal to that of the jets.

The liquid flows from the liquid layer through passages in the rotating separator into an annular chamber that forms an integral part of the separator wheel. As a result another liquid layer is formed and held against the surface by centrifugal force. This layer furnishes the fluid-energy source for the turbine rotating concentrically within the separator wheel and having the turbine inlet passages immersed in the liquid layer.

The turbine may be of any conventional type where the blades or passages can be arranged to intercept the liquid layer. Figure 4 shows a radial in-flow type of turbine while a "U tube" pickup is illustrated in Figure 5. The turbine wheel rotates at a lower speed than the separator wheel causing liquid from the layer to

Figure 4. Separator-Turbine with Radial Inflow Liquid Turbine

Figure 5. Separator-Turbine with U-Tube Liquid Turbine

enter the inlets, flow radially inward through passages, and flow to the liquid outlet pipe through apertures in the wall of the turbine shaft that is connected to the load to be driven. The turbine is supported on bearings.

The turbine passage can optionally incorporate a diffuser in which the velocity of the liquid entering the inlet can be partially converted to pressure such that, even allowing for the pressure drop in the radial passages due to centrifugal force, the liquid pressure in the discharge pipe is substantially higher than the pressure in the turbine casing and, in fact, greater than the pressure at the nozzle inlets. Thus, the diffusers can supply the necessary pumping of the liquid, eliminating the need for a separate pump to return the liquid to the nozzles.

For operation with high pressure at the discharge, the leakage of liquid between the shaft and the housing can be reduced by labyrinth seals and drains which return liquid leakage to the bottom of the housing where the liquid from this and other internal leakage sources is picked up by slinger blades and thrown back into the jets. Leakage to the outside of housing is prevented by a shaft seal. It should be emphasized that the two-phase turbine does not require close tolerances to be maintained and the seals and bearings are conventional off the-shelf items.

IV. CYCLE DESCRIPTIONS

Two versions of the Biphasic cycle have been identified as having potential in solar cooling applications. One is a completely integrated cycle that uses the same fluid as both the thermodynamic fluid in the prime mover, and the refrigerant in the cooling loop. The other has a separate power cycle with the turbine shaft tied into the compressor in a conventional compression-refrigeration loop and uses different working fluids in each cycle.

The integrated cooling system is shown schematically in Figure 6 along with the T-S diagram. The need for a boiler is circumvented by using the low-vapor-pressure turbine fluid as the heat-transfer fluid in the solar collector. The hot turbine fluid from the solar collector is mixed with the condensed working fluid and expanded through the nozzle into the separator/turbine where the fluids are separated and power is extracted from the liquid. The vapor phase from the turbine is combined with the portion of working fluid used as refrigerant and flows through the regenerator to the common condenser. In this cycle the Biphasic turbine is directly integrated into the refrigeration cycle. That is, the same fluid is used as both refrigerant and working fluid, thus eliminating the need for two separate condensers and further reducing capital costs. The condensed thermodynamic-fluid flow is split into two streams. The refrigerant portion (x lb/sec) is expanded to the low refrigeration pressure and temperature where it is vaporized by absorbing heat and subsequently recompressed. The turbine portion of the working fluid (1 lb/sec) is returned to the mixer via the regenerator.

The separate cycle is shown schematically in Figure 7. The operation of this cycle is basically the same as for the integrated cycle except there are separate condensers for the working

Figure 6. Combined Cycles

Figure 7. Schematic of Separate Cycle

fluid in the power cycle and the refrigerant in the refrigeration cycle, since different fluids are used for the two roles. The main difference in operation between the two cycles is that the separate cycle cannot conserve any of the compression energy as is done in the regenerator of the integrated cycle. Thus the separate cycle has an inherently lower performance than the integrated cycle.

V. CYCLE STUDIES

The mathematical models of the two Biphase Cycles developed in the previous section were programmed on the digital computer and parameter sweeps were run to determine the sensitivity of the cycle performance to the independent variables and to bracket the performance levels. The parameters that were varied and the range of values are shown in Table I. In addition different combinations of working fluids were studied to determine the optimum fluids from a performance point of view.

Many potential working fluids for the integrated cycles can be eliminated by the requirement that the thermodynamic fluid in the nozzle should operate considerably below the critical temperature. The remaining prime working-fluid candidates for this cycle are R11, R113, R216, isobutane and water. The main candidates for the turbine fluid are Krytox, Therminol 66, Dowtherm A, and Caloria. Water gives the best performance for the cycle, but the extremely large volumetric flow rates in the refrigeration section (particularly the compressor) make it an uneconomical fluid. Thus water has been eliminated from further consideration. All fluid combinations were compared over a range of temperatures from 220 to 350°F for the base case of component performance shown in Table II. Typical results for R113 and R216, which give the highest performance, are shown in conjunction with the various turbine fluids in Figures 8 and 9. Krytox has the highest performance and the flattest performance curve for both working fluids. Therminol 66, Dowtherm and water all give slightly reduced performance levels, while the performance curves for Caloria are the lowest, but still acceptable.

The dependence of the integrated cycle performance upon component efficiencies is summarized in Figures 10 and 11. These curves show that the coefficient of performance is strongly dependent upon both the turbine and compressor performance levels.

TABLE I

Summary of Range of Parameter Values
Used In Systems Study of Biphase
Solar Cooling Systems

Temperatures

$$350 \geq T_{\text{source}} \geq 220$$

$$115 \geq T_{\text{con}} \geq 90$$

$$45 \geq T_{\text{evap}} \geq 35$$

Component Performance

$$0.9 \geq \eta_t \geq 0.7$$

$$0.9 \geq \eta_c \geq 0.6$$

TABLE II

Component Performance Parameters Used In
Fluid Evaluation Studies

$$\eta_n = 0.9$$

$$\eta_t = 0.9$$

$$\eta_c = 0.9$$

$$\eta_p = 0.85$$

$$\epsilon_{\text{reg}} = 0.9$$

Trichlorotrifluorethane
Working fluid: R113

COEFFICIENT OF PERFORMANCE

1.0

0.9

0.8

0.7

0.6

0

4

8

12

16

20

Mass Flow Ratio

Krytox

Dowtherm A

Therminol 60 & Water

Caloria

$T_3 = 220^{\circ}\text{F}$

$T_7 = 100^{\circ}\text{F}$

$T_9 = 40^{\circ}\text{F}$

Figure 8. Performance of Combined Cycle for Various Turbine Fluids With R113

hexafluoropropane
Working Fluid: R216

Figure 9. Performance of Combined Cycle for Various Turbine Fluids With R216

Figure 10. Effect of Compressor Efficiency on Performance of Integrated Cooling Cycle

Figure 11. Effect of Turbine Efficiency on Performance of Integrated Cooling Cycle

The effect of operating temperatures are summarized in Figures 12 to 15. As the temperature at the nozzle exit is increased from 680°R the cycle performance increases (c.f. Figures 12 and 13) until the critical temperature is approached. Near the critical point the specific volume decreases and the amount of work done by the expanding gas on the liquid is significantly diminished, thus decreasing the power output of the turbine. The cycle performance for the R216 begins to fall off significantly at 300°F nozzle inlet temperature, while the R113 gives increasing performance with nozzle inlet temperature above 350°F . This behavior is a manifestation of the higher critical temperature of R113 (417.4°F for R113 vs 355.98°F for R216).

Decreasing the condenser temperature from 110 to 90°F causes the performance to increase by 30 to 50% (Figure 14). This is indicative of the potential performance gains to be accrued by switching from air-cooled to water-cooled condenser. The evaporation temperature also influences the performance of the integrated cycle as shown in Figure 15. Decreasing the evaporator temperature from 45 to 40°F causes the COP to decrease by 8 to 12%.

The effect of working fluid combinations and component performance levels were also studied for the separate cycle. The results of this phase of the study showed that water as the high-vapor pressure fluid in the power cycle gives much higher cycle efficiencies than any of the other candidates. The effects of the turbine fluid were found to be slight as for the integrated cycle. The parameter sweeps were made using Therminol 66 since it is readily available and a substantial amount of information exists on its properties and applications. Freon 22 was selected as the refrigerant since it is the working fluid used in most conventional vapor-compression air conditioners.

Figure 12. Effect of Nozzle Exit Temperature on Performance of Integrated Cycle for Therminol 66 + R216

THERMINOL 60 + R113

$T_7 = 560^\circ\text{R}$

$T_3 = 680^\circ\text{R}$

$T_9 = 500^\circ\text{F}$

$T_3 = 710^\circ\text{R}$

$T_3 = 760^\circ\text{R}$

$T_3 = 810^\circ\text{R}$

Figure 13. Effect of Nozzle Exit Temperature on Performance of Integrated Cycle for Therminol 66 + R113

THERMINOL 60 + R216

$T_3 = 250^\circ\text{F}$

$T_9 = 45^\circ\text{F}$

$\eta_t = 0.8$

COEFFICIENT OF PERFORMANCE

Figure 14. Effect of Condenser Temperature on Performance of Integrated Cycle

Figure 15. Effect of Evaporator Temperature on Performance of Integrated Cycle

The impact of cycle temperatures and component efficiency levels on system performance are similar to those for the integrated cycle. Typical results are shown in Figures 16 to 20. Two important differences in these findings are:

1. The COP increases with increasing mass ratio.
2. The COP is lower than for the integrated cycle.

The increasing COP with mass flow ratio is caused by the significantly higher specific volume of steam than other candidate working fluids (typically two orders of magnitude greater than for other fluids). Thus the expansion (first) term in the nozzle equation, Eq. (A-2) in Appendix A, dominates the second term which is proportional to the low-vapor pressure fluid pumping power. Consequently the turbine-power output increases faster than the pumping power and the power-cycle efficiency increases with r . The higher COP for the integrated cycle results from the conservation of part of the compression power in the regenerator.

The projected performance limits of the two cycles were determined by running the cycle simulation with the expected upper and lower performance limits of the compressor and turbine. Previous experimental work indicates that the projected efficiency of the Biphase turbine (exclusive of the nozzle performance) will be in the range of 75 to 85%,⁽¹⁾ while high-performance compressors can currently be designed and built with efficiencies in the same range.⁽²⁾ The projected brackets on system performance are shown in Figure 22 and 23 for the two cycles.

A very important consideration in a solar-cooling system is the auxiliary power system to drive the compressor, and supply parasitic power when the solar insolation drops below the design

Figure 16. Effect of Nozzle-Exit Temperature on Performance of Separate Cycle

Figure 17. Effect of Condenser Temperature on Performance of Separate Cycle

Figure 18. Effect of Evaporator Temperature on Performance of Separate Cycle

Figure 19. Effect of Compressor Efficiency on Performance of Separate Cycle

Figure 20. Effect of Turbine Efficiency on Performance of Integrated Cycle

Figure 21. Effect of Condenser Temperature on Performance of Separate Cycle

Figure 22. Projected Performance for Integrated Cooling Cycle

Figure 23. Projected Performance for Separate Cooling Cycle

level. This power can be supplied by stored energy in the form of sensible or latent heat or chemical energy. This aspect of solar-cooling systems has received much attention, but the only real conclusion thus far reached is that there are many options to be evaluated and that they add complexity and increase the cost of the systems. The Biphase Systems offer an important advantage in the method in which auxillary power can be supplied.

The Biphase turbine is a low-speed, high-torque prime mover whose shaft speed can be varied by changing the mass flow ratio. Thus the Biphase turbine can be designed to operate at 3600 RPM, which enables it to drive conventional pumps and compressors directly. More importantly, this shaft speed means that the system can also be driven by an induction motor connected to the turbine shaft as shown in Figure 24. In this configuration the power deficit caused by a decrease in solar insolation can be supplied directly by the synchronous motor without any gearing. If the solar insolation increases above the design value the motor will act as a generator, producing power already conditioned to be fed into the grid.

The operation of an induction motor in this application is well known.⁽³⁾ If the rotor speed slows slightly below the synchronous speed of the rotating field (i.e., 3600 rpm) a torque will be produced. As less power is produced by the turbine, the shaft and hence the rotor tends to slow down resulting in a larger slip between the rotor speed and the field. Thus a larger torque is produced. Similarly if the power output of the turbine exceeds the design value, the turbine speed increases above the synchronous speed of the field and the directions of magnetic field interactions and induced rotor currents reverse. As a result the mechanical power input to the shaft is converted into electric power which flows into the connected electrical system. The only requirement for the generator action is that the connected power system be able to supply enough kilovolt-amperes at lagging power factor to excite the generator by supplying the rotating magnetic field.

Figure 24. Conceptual Configuration of Biphase Turbine for Solar Cooling Applications

Typical performance curves for an induction machine acting both as a motor and a generator are presented in Figure 25.

The integrated Biphasic cooling cycle is projected to have higher performance levels than the separate cycle when compared for component having the same performance level. However, the three working fluids identified for the integrated cycle (i.e., R11, R113 and R216) have extremely high specific volumes compared to R12 and R22 which are the suggested working fluids for conventional cooling systems. These larger specific volumes can be translated directly into larger and more expensive compressors (by more than an order of magnitude). In addition rotational speeds in excess of 40,000 rpm would be required to attain higher compressor performance levels on the scale of three to five hp for these working fluids. Since the two-phase turbine is a low-speed high-torque machine, this would result in a mismatch of shaft speeds.

Consequently the separate cycle was selected as the best of the two concepts for this study. Although the performance of the separate cycle will be below that of an integrated cycle it is projected that the performance will exceed that of current absorption units. In addition a large degree of flexibility is gained with respect to the cooling system. That is, the separate cycle can be used to power a variety of air-conditioning systems, thus giving it potential for retrofit applications.

Figure 25. Typical Performance Curves for a Small Induction Motor/Generator

VI. DESIGN STUDY

As a result of detailed system studies it was decided that the configuration most suited for the Biphase Solar Cooling unit is the separate or dual cycle (see previous section). The computer simulation of this system described in Appendix A was run to determine the mass flow rates, thermodynamic state points, power output of turbine and power consumption for the system components. The design procedure is described in Appendix B and the cycle schematic with the design parameters is shown in Figure 25. The design constraints are summarized in Table III. The unspecified collectors were assumed to deliver Therminol 66 coolant at 300°F. The cooling system was specified by DOE as a 3-ton nominal unit cooled by air.

The matching of the collector and power system constraints resulted in an upper cycle temperature of 275°F (temperature of two-phase mixture entering the nozzle with saturated steam) and a mass flow ratio of 104. The efficiency of the power cycle was projected to be 15.2% with the component performance levels shown in Table IV. The COP of the overall system was projected to be 0.54 based on the power added to the coolant in the solar collectors.

The conceptual layout of the Biphase solar cooling system is shown in Figure 26. The design of the two-phase turbine that acts as the prime mover in the Biphase solar cooling system is shown in Figure 27, while the mixer design is shown in Figure 28. The design procedure is summarized in Appendix B.

The mixer-design concept selected for this application consists of a multiple-staged-orifice-target design, which incorporated two mechanisms for oil atomization, namely, jet breakup and obstacle impact or impingement. A two-stage design is shown in Figure 28, with a transition section for flow delivery to three nozzles. Four mixers, rather than one, were selected to provide for more gradual

Figure 26. Conceptual Layout of Biphase Solar Cooling System

FIG. 27 DESIGN OF BIPHASE SOLAR COOLING TURBINE

Rev.	By	Date	Description
1	W. H. H.	10/1/60	DESIGN
2	W. H. H.	10/1/60	REVISED
3	W. H. H.	10/1/60	REVISED
4	W. H. H.	10/1/60	REVISED
5	W. H. H.	10/1/60	REVISED
6	W. H. H.	10/1/60	REVISED
7	W. H. H.	10/1/60	REVISED
8	W. H. H.	10/1/60	REVISED
9	W. H. H.	10/1/60	REVISED
10	W. H. H.	10/1/60	REVISED

Rev.	By	Date	Description
1	W. H. H.	10/1/60	DESIGN
2	W. H. H.	10/1/60	REVISED
3	W. H. H.	10/1/60	REVISED
4	W. H. H.	10/1/60	REVISED
5	W. H. H.	10/1/60	REVISED
6	W. H. H.	10/1/60	REVISED
7	W. H. H.	10/1/60	REVISED
8	W. H. H.	10/1/60	REVISED
9	W. H. H.	10/1/60	REVISED
10	W. H. H.	10/1/60	REVISED

Rev.	By	Date	Description
1	W. H. H.	10/1/60	DESIGN
2	W. H. H.	10/1/60	REVISED
3	W. H. H.	10/1/60	REVISED
4	W. H. H.	10/1/60	REVISED
5	W. H. H.	10/1/60	REVISED
6	W. H. H.	10/1/60	REVISED
7	W. H. H.	10/1/60	REVISED
8	W. H. H.	10/1/60	REVISED
9	W. H. H.	10/1/60	REVISED
10	W. H. H.	10/1/60	REVISED

BIPHASE TURBINE
SOLAR COOLING
D/BE-397

PAGE 1 OF 1

- 2 STAGE ORIFICE TARGET DESIGN

Figure 28. Mixer Design

flow turning between the mixer exit and nozzle inlets, thus reducing the centrifugal force which causes droplet drift. Excessive drift will cause wetting of the tube walls and buildup of oil films which can lead to coalescence, slug flow of large droplets, and degradation in nozzle efficiency.

The water enters the mixer at the center and is diffused radially to the periphery. It then flows in an annulus across the hot oil jets, which impinge almost at 90° into the water film. Rapid vaporization of the water will occur with local small-scale turbulent mixing. The vaporization process should breakup the oil jets into ligaments from which droplets will be formed. Further breakup will occur due to droplet impact and breakup on the targets. In theory, as the number of stages are increased, exit droplet size should decrease as only the smallest sizes will be carried by the steam to the exit. Larger droplets will not follow the streamlines produced by the orifice/target structure. An equilibrium condition should be reached whereby exit mass ratio will equal inlet mass ratio. The design has flexibility so that additional stages can be added to provide further reduction in droplet size and increase in mixer pressure drop which must be made up by the liquid and water pumps. Consequently, an optimum condition will exist for which cycle efficiency is maximized.

Because the droplet size requirement is small, the properties of the flow will be well represented by the homogeneous equilibrium model, for which the effect of nozzle profile on efficiency is minimal. Consequently, a straight wall conical design was baselined. This is a lower cost design relative to a profiled contour.

The turbine is a single-staged unit in which a two-phase mixture of Therminol 66 and saturated steam is expanded through four nozzles to a velocity of 348 ft/sec (see Figure 29 for nozzle-rim configuration). This stream impinges on the inside surface of a 10.5" diameter free wheeling drum (separator) at an average angle

Figure 29. Nozzle Definition

TABLE III

Power System Design Constraints for Solar Powered
Cooling System

Collector-Unspecified

Temperature of coolant
exiting collector = 300°F

Air-Cooled Condenser

Condenser operating
temperature = 115°F

Nominal cooling
capacity = 3 tons

Compressor power:

R22: 4hp

Pumping power:

Low vapor
pressure fluid: 0.36hp

High vapor
pressure fluid: 0.0033hp

Turbine output power: 4.37hp

System COP = 0.54

TABLE IV

Component Performance Values Used in Design of
Biphase Turbine For Solar Cooling System

$$\eta_{coll} = 0.50$$

$$\eta_n = 0.85$$

$$\eta_t = 0.80$$

$$\eta_p = 0.80$$

$$\eta_{comp} = 0.80$$

of 20° , thus driving the drum at a velocity of 327 ft/sec (7138 rpm). The liquid portion of the stream separates out due to the large centrifugal force field and forms a layer on the drum and flows through holes in the drum to the opposite side of the support hub. This layer drives the turbine shaft as the liquid is collected by a U-Tube submerged in the layer. The discharge from the U-Tube falls into the hot-well under the force of gravity. The separator drum free wheels on its own shaft that is concentric with the turbine output shaft and is supported on bearings and held in place by clips. The two U-Tube pickups are attached to the output shaft which is supported on another set of bearings. The turbine is designed for liquid kinetic-energy recovery only since over 99% of the total kinetic energy resides in the liquid phase at the high mass ratio used in this application.

The separated steam is drawn off through a three-inch diameter hole in the casing by the condenser. This results in a stream velocity of about 100 ft/sec through the condenser-feed line.

Note that static fluid compatibility tests were run (see Appendix C) and indicated that Therminol 66 and water should be compatible as a choice of fluid combinations. However, additional dynamic testing is required before a final decision can be made regarding fluid compatibility for Biphase power systems.

SALIENT FEATURES OF DESIGN

An important feature of the design is the "piggy-back" mounting of the two bearing sets which reduces the length of the assembly, but more importantly reduces the speed differential between races (of the turbine bearings) to 3600 rpm.

The bearings are standard "off-the-shelf" items. They operate in a grease-packed cavity and are lightly preloaded by adjacent wave spring washers. Concentricity of interdependent parts is favored

by a minimum of machined diameter variations. Sealing at the chamber side of the bearings is accomplished by teflon lip seals. The separator is a medium-carbon-steel forging machined to size. An alternative to this might be a centrifugally cast aluminum rotor. The pick-up or turbine comprises stock-sized tubing. The nozzles (4) are investment castings, a process which lends itself to accurate fabrication especially in large quantities. The 4 independent nozzles would be fed by respective fluid and steam manifolds external to the housing.

The housings are castings - either iron or aluminum and would be sand, permanent mold or die cast depending on production quantities.

VII. CONCLUSIONS

Detailed cycle calculations over ranges of values for independent variables have shown the solar-powered cooling systems driven by a Biphase turbine have potentially high performance levels.

The high system performance can be translated directly into low system cost through reduced requirements in solar collector area. In addition since the Biphase turbine can be designed to operate at 3600 rpm, supplementary power will be supplied by a 3600 rpm synchronous motor mounted directly on the output shaft.

In addition, it is likely that the Biphase turbine will be cheaper to fabricate and maintain than conventional turbines because no close tolerances are required. It is a low rpm machine and conventional off-the-shelf bearings and seals can be used.

A conceptual configuration was developed for a 3-ton Biphase solar-powered cooling system based on the results of detailed cycle studies. A biphase turbine was designed for this system to give the required 4.35 hp shaft power at 3600 rpm. The detailed design demonstrated that a simple compact two-phase turbine could be designed and fabricated using off-the-shelf components and conventional mass-production techniques.

APPENDIX A

CYCLE ANALYSIS

The two cycles studied in detail are shown schematically in Figures 6 and 7. Both of these cycles are comprised of the same basic components. These are as follows:

1. Two-Phase nozzle
2. Turbine/Separator
3. Regenerator
4. Condensers
5. Pumps
6. Evaporator

The mathematical descriptions of the components and the calculational method for the cycles are presented in this section. All calculations are based on one pound per second of working fluid flowing through the power cycle.

The integrated cycle (See Figure 6) is presented first. The calculations begin with the nozzle for which the following simplifying assumptions have been made.

1. r pounds of low-vapor-pressure liquid (turbine fluid) and one pound of high-vapor-pressure fluid (thermodynamic working fluid) expand from pressure p_3 to p_4 .
2. The liquid is uniformly dispersed as spherical drops (all of same diameter).
3. No external forces act on the two-phase mixture other than the pressure and wall shear.
4. No heat transfer to or from the mixture (expansion proceeds adiabatically).
5. Flow is quasi-one dimensional.
6. Liquid droplets are isothermal.
7. Gas mixture obeys additive pressure law.
8. No relative velocity between the phases.

The conservation of mass between locations 3 and 4 gives (1)

$$\int_{T_3}^{T_4} C_{pg} dT + \frac{(V_4)_s^2}{2gJ} + r \left[C_{pl}(T_4 - T_3) + \frac{(p_4 - p_3)}{\rho_l} \right] = 0 \quad (1)$$

The work done by the liquid and gas in expanding from pressure P_3 to P_4 is equal to the isentropic gain in kinetic energy:

$$(V_4)_s^2 \frac{(1+r)}{2g} = - \int_{P_3}^{P_4} v_g dP + r \frac{(p_4 - p_3)}{\rho_l} \quad (2)$$

Equations (1) and (2) can be combined with the polytropic expansion relation:

$$p_3 v_{g3}^n = p_4 v_{g4}^n \quad (3)$$

to give:

$$\int_{T_3}^{T_4} C_{pg} dT + \frac{n}{n-1} p_3 v_{g3} \left[1 - \left(\frac{p_4}{p_3} \right)^{\frac{n-1}{n}} \right] + r C_{pl}(T_4 - T_3) = 0 \quad (4)$$

Assuming that $C_{pg} \cong \text{constant}$ (the temperature drop through nozzle turns out to be very small for the large mass ratios used in the Biphase system) we obtain:

$$n = \frac{-A}{1-A} \quad (5)$$

where:

$$A = \frac{(C_{pg} + r C_{pl}) T_3}{p_3 v_{g3}} \quad (6)$$

Define the nozzle efficiency as

$$\eta_n = \frac{V_4^2}{(V_4)_s^2} \quad (7)$$

and the turbine efficiency as:

$$\eta_t = \frac{P_t(2gJ)}{V_u^2(1+r)} \quad (8)$$

or:

$$P_t = \frac{\eta_t \eta_n (V_u)_s^2 (1+r)}{2gJ} \quad (9)$$

The temperature at the turbine exit is found by applying the first law of thermodynamics across the nozzle and turbine/separator as:

$$T_s = T_3 + \frac{rv_\ell(p_3 - p_s) - P_t}{rC_{p\ell}} \quad (10)$$

The power required to pump the condensed working fluid is given by:

$$P_{p1} = \frac{(p_{12} - p_7)}{\rho_{gl} \eta_p} \quad (11)$$

and the temperature of the fluid exiting the pump is:

$$T_{12} = T_7 + \frac{P_{p1}(1-\eta_p)}{C_{pg\ell}} \quad (12)$$

The pumping power required to circulate the turbine fluid is given by:

$$P_{p2} = \frac{r(p_3 - p_u)}{\rho_\ell \eta_p} \quad (13)$$

The compression power per pound of refrigerant in the refrigerant loop is given by:

$$P_c = \frac{C_{pg} T_9}{\eta_c} \left[\left(\frac{p_{10}}{p_9} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad (14)$$

Then the flowrate of working fluid and through the refrigerant loop per unit flow rate of working fluid in the power cycle is obtained by matching the turbine power with the pumping requirements as:

$$P_t = P_{p1} + P_{p2} + xP_c \quad (15)$$

The temperature out of the compressor is :

$$T_{10} = T_9 + \frac{P_c}{C_{pg}} \quad (16)$$

while the temperature T_{11} is obtained from an energy balance as:

$$T_{11} = \frac{xT_{10} + T_5}{1+x} \quad (17)$$

The regenerator performance is represented in terms of the effectiveness of the device, which is directly related to the surface area. The regenerator temperatures are given by:

$$T_{13} = T_{12} + \epsilon(T_{11} - T_{12}) \quad (18)$$

$$T_6 = T_{11} + \frac{(T_{12} - T_{13})}{C_{pg}(1+x)} C_{pgl} \quad (19)$$

The heat rejection is given by:

$$Q_r = (1+x)(h_6 - h_7) \quad (20)$$

and the refrigeration is given by:

$$Q_{cool} = x(h_9 - h_7) \quad (21)$$

Since the turbine power balances the pumping power requirements (c.f. eg. 15), the heat addition to the cycle is given by:

$$Q_a = Q_r - Q_{cool} \quad (22)$$

and the coefficient of performance for the system is:

$$\text{COP} = \frac{Q_{\text{cool}}}{Q_a} \quad (23)$$

In the separate cycle the turbine and pumping power requirements are calculated exactly the same as in the integrated cycle (c.f. equations (1) to (14) above).

However the flow rates through both the hot and cold sides of the regenerator are the same for this system (the refrigerant has its own separate condenser). Thus the temperature at the exit of the cold side of the regenerator is now:

$$T_{13} = T_{12} + \epsilon(T_5 - T_{12}) \quad (24)$$

and at the exit of the hot side:

$$T_6 = T_5 + (T_{12} - T_{13}) \frac{C_{pgl}}{C_{pg}} \quad (25)$$

The heat rejection in the condenser is:

$$(Q_r)_p = (h_6 - h_7) \quad (26)$$

and the efficiency of the power cycle is given by:

$$\eta = \frac{P_t - (P_{p1} + P_{p2})}{Q_r + P_t - (P_{p1} + P_{p2})} \quad (27)$$

The compression calculations in the refrigeration cycle are given by equations (14) and (16). The heat rejection is given by:

$$(Q_r)_r = x(h_{10} - h_7) \quad (28)$$

and the cooling rate is:

$$Q_{\text{cool}} = x(h_9 - h_7) \quad (29)$$

where X is found as in the integrated cycle (c.f. eq. 15).

The total heat added to the system is then:

$$Q_A = (Q_r)_r + (Q_r)_p - Q_{cool} \quad (50)$$

and the overall coefficient of performance is:

$$COP = \frac{Q_{cool}}{Q_A} \quad (51)$$

APPENDIX B

DESIGN CALCULATIONS

The schematic of the Biphasic dual solar cooling system is shown in Figure 7. The heat added in the collectors is given by:

$$Q_A = C_{p_\ell} (T_5 - T_4) \dot{m}_\ell \quad (1)$$

while an energy balance on the mixer gives:

$$\begin{aligned} \dot{m}_\ell C_{p_\ell} (T_a - T_3) \\ = \dot{m}_g \left((T_2 - T_8) (C_{p_\ell}) + h_{fg} + C_{p_v} (T_3 - T_2) \right) \end{aligned} \quad (2)$$

where:

$$T_a = T_s + \frac{W_{k_p}}{\dot{m}_\ell C_{p_\ell}} \quad (3)$$

The free parameters T_2 , T_3 and T_s were selected from selected from parameter sweeps as follows:

$$T_2 = T_3 = 735^\circ\text{R}$$

$$T_s = 760^\circ\text{F}$$

The mass ratio r is selected by matching the value of T_3 given by equations (1) to (3) with that from the cycle calculations (system of equations (A-1) to (A-31)). Simultaneous solution of these two sets of equations with the system constraints give the design mass ratio $r = 104$.

The isentropic velocity for the design conditions expanding the two-phase mixture with saturated steam from 45.5 psia to 1.5 psia is obtained from equation (A-2) as:

$$V_s = 367 \text{ ft/sec}$$

The resulting cycle performance for the component performance levels given in Table I was obtained from the computer code that models the separate cycle was 15.2% (see Figure 26 for summary) while the overall coefficient of performance was projected to be 0.54.

A flow rate of one lb/sec of R22 through the refrigeration cycle was calculated to give 64.435 B/sec of refrigeration. Since a nominal three-ton cooling unit must deliver 10 B/sec the mass flow rate of coolant is found to be:

$$\dot{m}_c = \frac{10}{64.435} = 0.155 \text{ lb/sec}$$

One lb/sec of steam flow rate in the power cycle supplies enough power to handle 10.15 lb/sec of R22 in the refrigeration loop. Thus the mass flow rate of steam in the power cycle is:

$$\dot{m}_{st} = \frac{0.155}{10.15} = 0.0153 \text{ lb/sec}$$

and the mass flow rate of Therminol 66 is:

$$\dot{m}_{T66} = r\dot{m}_{st} = 104 \times 0.0153 = 1.592 \text{ lb/sec}$$

The volumetric flow rate of steam at the nozzle exit is:

$$\dot{Q}_{st} = v_{st}\dot{m}_{st} = 332.3 \times 0.0153 = 5.07 \text{ ft}^3/\text{sec}$$

and the volumetric flow rate of Therminol 66 is:

$$Q_{T_{66}} = \frac{\dot{m}_{T_{66}}}{\rho_{T_{66}}} = \frac{1.592}{60.03} = 0.027 \text{ ft}^3/\text{sec}$$

the total volumetric flow rate at the nozzle exit is:

$$Q_T = Q_{st} + Q_{T_{66}} = 5.10 \text{ ft}^3/\text{sec}$$

The nozzle-separator design chosen is illustrated conceptually in Figure 28. In this arrangement these two-phase mixture is accelerated to its high final velocity through nozzles located in the same radial plane as the separator rim and the resulting fluid stream impinges upon the rim at an average angle θ to the tangent. In this design four nozzles with a thickness t of $7/8$ " were used. This resulted in an exit width given by:

$$W_o = \frac{Q_T / 4}{V_n t} = 0.603 \text{ in}$$

The ratio of the nozzle throat width to the exit width is given by:

$$\frac{W^*}{W_o} = \frac{\left(\frac{p_o}{p_t}\right)^{1/n} \frac{2n}{n-1} \left(1 - \frac{p_o}{p_t}\right)^{\frac{n-1}{n}}}{\left(\frac{2n}{n+1}\right)^{\frac{n+1}{n-1}}}$$

yields $W^* = 0.140 \text{ in.}$

The half angle of the diverging part of the nozzle is obtained from geometric considerations as:

$$\delta = \tan^{-1} \left(\frac{W_o}{2L} \left(1 - \frac{W^*}{W_o} \right) \right)$$

Rearrangement of this relationship yields $L=1.20$ in. for design value of $\delta = 10^\circ$.

The design rim speed is related to the fluid velocity and geometry by:

$$N_{\text{rim}} = \frac{60xV_n \cos\theta}{2\pi R}$$

A separator radius of 5.25 and impinging angle of 20° yields a rim speed of 7138 rpm.

The liquid pickup acting as an impulse turbine yields a maximum efficiency at a speed ratio of $1/2$. Thus this turbine will have its peak performance at a speed of 3568 rpm. The turbine is to be designed with two liquid pickups for ease of balancing. Thus at design conditions each pickup must handle about 0.8 lb/sec or $0.0133 \text{ ft}^3/\text{sec}$ of Therminol 66.

Experience has shown that the U-Tube pickups should not flow full.

The design capacity of a U-Tube pickup given by:

$$Q = 0.5V_n \cos 20 \quad (\text{Wh})$$

shows that a 0.2 in. x 0.15 in. rectangular tube will have a capacity of $0.034 \text{ ft}^3/\text{sec}$ flowing full, which should suffice.

The total power output of the turbine will be 4.35 hp operating at design conditions.

The important turbine parameters are summarized in Table B-I, with important numerical calculational procedures and results are presented in Table B-II:

TABLE B-I

Turbine Parameters

Flow Rates

Steam = 0.015 lb/sec
Therminol 66 = 1.592 lb/sec

Nozzles

Entrance Pressure = 45.5 psia
Number = 4
Exit velocity = 348 ft/sec

Dimensions

Length of
diverging section = 1.20 in.
Throat = 0.875 in. x 0.140 in.
Exit = 0.875 in. x 0.603 in.
Half angles of
diverging section = 10°

Separator

Inner diameter = 10.5 in.
rpm = 7138
Speed = 332 ft/sec
Pressure = 1.5 psia
Temperature = 115°F

Turbine

rpm = 3568
Power Output = 4.35hp

TABLE B-II

Summary of Turbine Design Calculations

1. Estimated rotor weight

11.74#

Bearing size = 15mm (both bearings)
 Shaft = 0.5906
 Effective Shaft diameter = 0.80 in.

2. Rotor Unbalance Load

Rotor = 7200 rpm
 Maximum imbalance = 0.05 ounce inches

 Force = $\frac{0.05}{6172} (.1045 \times 7200)^2$
 = 4.59 lb

3. Rotor Deflection

$$\delta = \frac{CW(L+C)^2}{3EI}$$

where:

W = rotor load, lb.

E = 30×10^6 psi

I = $\frac{\pi(d)^4}{64}$

d = Effective shaft diameter, inches

C = Rotor overhang, inches

L = Bearing spacing

Thus I = $.049(.5906)^4 = 0.006$

$$\delta = \frac{16.33(1.18)^2(1.18+1.86)}{3 \times 30 \times 10^6 \times 0.006} = 0.00013 \text{ in}$$

TABLE B-II (Continued)

Critical Speed

$$N_c = \frac{30}{\pi} \left(\frac{g}{\delta} \right)^{1/2} = 16471 \text{ rpm}$$

Stress in rotor shaft

ASME Power Code Equation

$$S_s = \frac{16}{d^3} \left((K_b M_b)^2 + (K_t M_t)^2 \right)^{1/2}$$

where:

S_s = Shear Stress-Torsion

d = Shaft diameter, inches

K_b = Combined shock and fatigue factor for bending.

M_b = Bending moment in-lb

K_t = Combined shock and fatigue factor for torsion
-1.5 for minor shock.

M_t = Torsional moment, in-lb.

Neglect bending moment

$$S_s = \frac{16}{(0.5906)^3} \frac{1.5 \times 5 \times 33,000 \times 12}{2\pi \times 7,200}$$

$$= 629 \text{ psi}$$

Stress in rotor (separator)

TABLE B-II (Continued)

$$\begin{aligned}\text{Weight of Rim} &= 2.5 \text{ lbs.} \\ \text{Velocity of Rim, 7200 rpm} &= \frac{7200 \times \pi}{30} = 754 \text{ radius/sec} \\ \text{Centrifugal Force} &= \frac{W(\omega^2)R}{g}\end{aligned}$$

Where

W = Weight of rim

ω = Velocity, radius/sec

R = Rim Radius, in.

$$= \frac{2.5(754)^2 6}{32.2 \times 10.5} = 25222.1 \text{ lbs.}$$

$$\text{Tangential or Hoop Stress} = \frac{25222}{.28 \times 2} = 45040 \text{ psi}$$

$$\begin{aligned}\text{Weight of Liquid, @ max depth in rim} &= .20 \times 1.75 \times 10.5 \times \pi \times .036 \\ &= .42 \text{ lbs.}\end{aligned}$$

$$\text{Centrifugal Force} = \frac{.42(754)^2 6}{32.2 \times 10.5} = 4193\#$$

$$\text{Increase in hoop stress} = \frac{4193}{25222} = .166$$

$$\text{Hoop Stress, Max} = 45040 \times 1.17 = 52692 \text{ psi}$$

TABLE B-III

ROTOR SHAFT BEARINGS

Bearing Size	15mmx35mm	AFBMA #202
Bearing Span	1.86"	
Rotor Overhang	1.18"	
Rotor (Inner Race) Speed	7200 rpm	
Housing (Outer Race) Speed	3600 rpm	
Speed Differential (Inner to Outer Race)	3600 rpm	

Radial Load = Rotor WT(@Bearing) = 19.20

Thrust Load = Wave Washer Preload = 15-20 lbs.

Reference: - New Departure Hyatt Bearing Catalog #3202 Single
Row Radial Series Brg, Capacity @ 500 rpm = 325 lbs.

$$\text{Thrust/Radial Load} = T/R = 17.5/19.2 = .91$$

Load Conversion Factor, FC	= 1.61
Equivalent Radial Load, P = FCxR	= 30.91
Speed Factor, For 3600, SF	= .61
Capacity @ Operating Speed	= .61x325 = 198.25
Life Factor	= 198.25/30.91 = 6.41

This value results in a B-10 life greater than that
tabled (i.e., 100,000 hrs.)

$$\begin{aligned}
 \text{Centrifugal Load Per Inch of circumference} &= \frac{29415}{10.5 \times \pi} \\
 &= 891 \text{ lb.} \\
 \text{Rim Load per inch Width} &= \frac{891}{2.25} \# \\
 \text{or, Unit Load @ Rim} &= 396 \#/\text{inch} \\
 \text{If Disc Width @ Rim} &= .20 \\
 \text{Then disc (@rim) Stress} &= \frac{891}{.2} = 4455 \text{ psi}
 \end{aligned}$$

Assuming uniform thickness of .2"

Max $\sigma_{\text{tangential}}$ @ Hub, Hole Diameter of .75"

$$= \frac{1}{4} \frac{\delta w^2}{386.4} \left[(3+v)R^2 + (1-v)R_o^2 \right]$$

where δ = Density, lb/in³

w = Angular velocity, Radi/sec

v = Possons Ratio

R = Disc Outside Radius, in.

R_o = Hub hole Radius, in.

$$\begin{aligned}
 \sigma_t &= \frac{1}{4} \frac{(.283)(754)^2}{386.4} \left[(3+.3)6^2 + (1-.3).375^2 \right] \\
 &= 12377 \text{ psi}
 \end{aligned}$$

$$\begin{aligned}
 \sigma_{\text{tangential}} \text{ @ Disc rim} &= \frac{1}{8} \frac{\delta w^2}{386.4} \left[(3+v)(R^2 + R_o^2 + \frac{R^2 R_o^2}{r^2}) \right. \\
 &\quad \left. - (1+3v)r^2 \right]
 \end{aligned}$$

where r = Radial Distance of Selected Section in.

this case, $r = R$

$$\begin{aligned}
 \sigma_t &= \frac{1}{8} \frac{(.283)(754)^2}{386.4} \left[(3+.3)(6^2 + .375^2 + \frac{6^2 \times .375^2}{6^2}) \right] \\
 &= 2672 \text{ psi}
 \end{aligned}$$

$$\text{Rim stress} = 3490 + 2672 = 6162 \text{ psi}$$

CONCLUSIONS: The Disc material of steel is adequate, however the rim hoop stress would require a medium carbon steel of 60,000 psi yield. Strength i.e., A.1.5.1 1045-1050 Group.

APPENDIX C

FLUID COMPATIBILITY TESTS

Several series of static fluid tests were run to determine the compatibility of the various working fluid combinations that showed promise for the Biphasic Solar powered cooling system. Preliminary fluid screening was based upon cycle screening and projected performance levels.

Since the separate cycle was the preferred configuration for the 3-ton Biphasic cooling system water was selected as the working fluid (see Section VI). Equal volumes of various low-vapor pressure fluids and water (4cc each) were poured into 300 series stainless steel capsules with about 2cc of ullage between the liquid level and a Bourdon tube gage.

In the first series of tests the ullage was evacuated and the capsules were inserted into a furnace and maintained at 400°F for 500 hours. The procedure was modified slightly in the second series of tests. The samples were inserted into the furnace as in the first tests but the temperature was reduced to 300°F. At the end of 250 hours the samples were removed and the vacuum was broken allowing air into the ullage so information could be obtained about oxidation effect on the fluids. The samples so treated were returned to the furnace for an additional 250 hours.

Low vapor-pressure fluids listed in Table C-I were tested as described above and evaluated for evidence of any chemical reaction as indicated by an increase in vapor pressure with time, solute formation or discoloration. Inspection of all fluids showed no evidence of chemical reaction.

TABLE C-1

Low-Vapor Pressure Fluids Tested For
Compatability With Water

<u>FLUID</u>	<u>Manufacture</u>
THERMINOL 44	Monsanto
THERMINOL 60	Monsanto
THERMINOL 66	Monsanto
CALORIA HT43	Exxon
KRYTOX 143 AC	Dupont
DOWTHERM A	Dow

APPENDIX D

APPLICATIONS STUDY

Solar powered cooling systems generally require two unique components that conventional cooling systems do not have. These are a bank of solar collectors and a means of supplying or storing supplementary energy for the system when solar insolation falls below the nominal level for which the system is designed to operate. Unfortunately in most solar cooling systems these two components make up the majority of the initial capital cost. This is true for both new units and retrofitting of existing units.

Assuming a solar collector efficiency of 50% and an overall COP of 0.5 for an air-cooled condenser, a three-ton solar-powered cooling unit would require 480 ft^2 of active collector surface. At today's costs of about $\$20/\text{ft}^2$ this can be translated to about $\$10,000$ for solar panels. Thus a ten percent improvement in performance level would result in a cost reduction approximately equal to the cost of a conventional three-ton unit. The Biphase solar-powered system has the potential to offer such a performance improvement.

Thus the Biphase solar-powered cooling system has the potential to have a significant economic advantage over other proposed solar-powered cooling systems. Despite this potential advantage, a three-ton Biphase system is projected to cost $\$11,911$ in 1978 dollars which is far in excess of the nominal installed cost of conventional systems. (See Table D-I and Figure D-1). Even if the collector cost were to come down to $\$5/\text{ft}^2$ the cost of a Biphase Solar cooling system is projected to be $\$4,711$.

The annual operating costs of the Biphase solar cooling system are compared to those of a conventional three-ton electric powered cooling system in Figure D-2.

Solar powered cooling system can be competitive only when the cost of electrical power becomes high, the cost of solar collectors become cheap and the annual operating time becomes large. On the other hand government subsidies for solar-powered cooling can completely change the whole picture.

Current collector costs tend to be closer to the \$20/ft² than \$5/ft². However the cost of solar collectors is projected to come down as production volume goes up. For example, the General Electric TC-100 collectors, which could operate with the Biphase system currently sell for about \$20/ft² FOB Valley Forge, PA in quantities of 1600 ft² or more. However, General Electric expects that they could build these collectors for \$5/ft² if they could tool up for 1,000,000 ft²/yr. (5) Retrofit possibilities were also considered in this study. It would be possible to link the output of the Biphase turbine directly to the compressor shaft by means of a V-belt if the compressor-motor unit is not hermetic. However, most present-day systems are hermetically sealed which would negate this possibility. In addition the cost of the air-conditioning portion of the system makes up only a fraction of the cost of the complete solar-powered unit so that it would probably end up being cheaper to fabricate the complete Biphase system at the factory and completely replace existing conventional air conditioning systems.

TABLE D-1

Cost Estimate of Three Ton Biphas Solar Cooling System
1978 Dollars

<u>Component</u>	<u>OEM Cost (\$)</u>
Biphase Turbine	200
Compressor/Motor	50
Compressor Seal	50
Compressor Clutch	62
Therminol Pump	158
Condensate Pump	116
Regenerator	240
Water Condenser	200
R22 Evaporator	250
R22 Condenser	200
Air Blower/Motor	145
Controls & Valves & Plumbing	640
Collectors @ \$20.ft ²	<u>9600</u>
	\$11,911

Figure D-1. Capital Cost (OEM) of Biphase Solar Cooling System as a Function of Collector Cost

Figure D-2. Annual Operating Expenses of 3 Ton Cooling System Assuming an Annual Interest Rate of 8%

APPENDIX E

NOMENCLATURE

C_{pg}	-	Specific heat of high-vapor-pressure fluid in vapor phase, B/lbm-°R.
C_{pgl}	-	Specific heat of high-vapor-pressure fluid in liquid phase, B/lbm-°R.
C_{pl}	-	Specific heat of low-vapor-pressure fluid in liquid phase.
COP	-	Coefficient of performance, B/lbm-°R.
g	-	32.2 lbm-ft/lb+-Sec ²
h	-	Specific enthalpy, B/lbm
J	-	FF8 ft-lb+/B
$\dot{m}$	-	Mass flow rate, lb/sec.
p	-	Pressure, psia
P	-	Power, B/sec
Q	-	Heat transfer rate, B/sec
r	-	Ratio of low-vapor-pressure fluid to high-vapor-pressure fluid by excess
T	-	Temperature, °R
v	-	Specific volume, ft ³ /lbm.
V	-	Velocity, ft/sec.
X	-	Fraction of total flow of working fluid that goes through refrigeration loop in combined cycle.

Greek Letters

- γ - Ratio of specific heats
- ϵ - Heat exchanger effectiveness.
- η - Isentropic efficiency
- ρ - Density, lbm/ft^3

Subscripts

- a - Quantity added
- A - Quantity added
- c - Compression
- Cool - Quantity for cooling purposes.
- g - Gas
- l - Liquid
- n - Nozzle
- p - Pump
- r - Regeneration
- s - Isentropic
- reg - Recuperation
- t - Turbine

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