ENGINEERING DESIGN CONSTRUCTION
AND TESTING OF A SALT-WATER
ABSORPTION UNIT OPTIMIZED FOR USE
WITH A SOLAR COLLECTOR HEAT SOURCE

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ABSTRACT

This report covers the development of a 3 ton direct evaporatively cooled LiBr chiller and the construction of three operable field test units suitable for use with flat plate collectors.

Compared to conventional LiBr chillers using shell and tube heat exchangers and a separate cooling tower this approach aims to reduce first cost, installation cost and parasitic power.

The unit is packaged into a size approximately 94 cm square and 2 meters tall. It produces 7.2°C (45F) chilled water with a .72 COP when fired with 89.4°C (193F) hot water while rejecting heat to 25.6°C (78F) wb ambient air. Power to operate the condenser fan, solution pump and surface wetting pump is 450 watts.

Air cooled units are frequently proposed to eliminate the nuisance of water. Their penalty is higher hot water firing temperature and lower COP. This unit deals with water freezing by making the sump and wetting pump freeze resistant so that seasonal draining is not required. Low heat flux through the wetted surfaces yields performance insensitivity to the accumulation of scale.

This approach may be the best ultimate residential solar air conditioning device so work should continue to advance its development further.

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INTRODUCTION

The purpose of the work covered by this report is to accelerate availability of an evaporatively cooled unit package, optimized for solar firing, which can be integrated into a unitary solar heating and cooling system package. Currently available equipment utilizes a shell and tube heat exchanger concept as opposed to evaporative cooling.

Evaporatively cooled absorption equipment has some important advantages over conventional water cooled equipment. Conventional systems require a water cooled chiller plus a cooling tower to provide and recirculate the cooling water. The evaporatively cooled system combines the chiller and cooling tower into one package. By doing this, the labor and material for installing the cooling water and electric lines to the tower are eliminated and the pump work for recirculating cooling water is minimized. The decrease in pump work helps lower the parasitic power requirements which is an important consideration in a solar installation.

Other advantages include the use of a freeze resistant concept and increased absorber and condenser heat transfer surface areas. The freeze resistant concept eliminates the service calls required in draining the cooling water circuit in the fall and filling it in the spring. The heat flux on the larger heat transfer surfaces would be lower, thus allowing a larger build-up of scale before affecting unit performance. Also, the time for scale build-up to the point requiring removal would be increased, which would reduce maintenance expenses.

The feasibility of an evaporatively cooled salt-water absorption cooling unit was demonstrated under NSF Grant GI-44095. The concept of the unit, shown in Figure 1, was a cylindrical condenser which in turn was surrounded by succeeding annular vessels spaced apart from the condenser and each other, the latter vessels serving as the absorber. A test unit was built utilizing this concept.

The test unit proved the feasibility of the concept but to produce field test equipment, further refinement was necessary, using this crude test unit (prototype #1) as a basis. This report covers development of a unit suitable for field testing and eventually production.

.. Development of the unit was approached with two general objectives:

- 1) Refinement of absorber and condenser surfaces and other sealed-unit components to produce a unit of minimum size which has good reliability.
- 2) Combine the refined sealed-unit with the necessary external components in such a way that the end result is a sophisticated evaporatively cooled field test unit compatible with the unitary solar heating and cooling system package being developed by Arkla under Contract EG-77-C-02-4593.

The specific objectives were:

- 1) Optimization of absorber and condenser surfaces resulting in unit size reduction.
- 2) Selection of effective absorber and condenser air side corrosion protection.
- 3) Development of reliable hermetic solution pump and pump package.
- 4) Development of the evaporative cooling components including the water sump, water collector, water distributor, fan and fan venturi.
- 5) Construction of three field test units.

Due to the extra time required for formulation and development of ideas to accomplish compatibility with the unitary package, performance testing of the units was limited to the laboratory. However, in on-going Contract No. EG-77-C-4593-*000, extensive field testing of a retrofit installation is planned.

TECHNICAL

ABSORBER CONDENSER DEVELOPMENTSize Reduction

The evaporatively cooled absorber and condenser require large heat transfer surfaces. The surfaces of prototype #1 (NSF unit) were not optimized for heat transfer. Additional testing was needed to supply data necessary for determining how much, if any, the surfaces could be reduced.

The absorber consisted of two annular vessels with the inner vessel containing approximately 40% of the total absorber surface. See Figure 1. The solution flow to the outer vessel was cut off leaving only 40% of the total surface operative. The resulting capacity was 8.7 KW (30,000 Btuh). This was 83% of full capacity and gave a good indication that a reduction in surface would be possible. For further verification, the inner vessels solution flow was cut off and the outer vessel used as the absorber. Then additional evaporator surface was added using the total absorber surface. Also capacity tests were performed using higher heat inputs with performance optimized by adjustment of solution concentration. After analyzing the results of these tests, the absorber and condenser were redesigned with a 23% decrease in their surface areas. See Figures 2 and 3 for details. The additional annular vessel was added to make up the loss in condenser surface due to the elimination of the vertical fins. The vertical fins were replaced with a new horizontal sinusoidal configuration which was also used on the absorber and condenser annular vessels. See Figure 5. The advantages of the sinusoidal configuration are:

- 1) Reduced fabrication problems
- 2) Makes a more rigid assembly which allows the use of thinner material

1. Generator Shell & Transfer Surface	9. Liquid Refrigerant Line
2. Solution Pre-Heater	10. Drive Motor for Pumps
3. Solution Heat Exchanger	11. Solution Pump
4. Evaporatively Cooled Plate Absorber	12. Evaporative Water Pump
5. Evaporatively Cooled Plate Condenser	13. Chilled Water Pump
6. Absorber Solution Drippers	14. Fan
7. Evaporator Refrigerant Drippers	15. Solution Eliminator Baffle (Carryover)
8. Evaporator Coil	

FIGURE 1

SCHEMATIC OF SOLAR UNIT

FIGURE 2

EVAPORATIVELY COOLED SOLAR CHILLER
 ABSORBER/CONDENSER ASSEMBLY
 PROTOTYPE NUMBER 1

FIGURE 3

EVAPORATIVELY COOLED SOLAR CHILLER
ABSORBER/CONDENSER ASSEMBLY
PROTOTYPE NUMBER 2

- 3) Induces turbulence in the air flow between the surfaces which should improve heat transfer.

Performance of the test unit (prototype #2), built with the reduced surfaces, indicated an additional 14% reduction was possible. See Figure 4. The final design reflects a 34% decrease in the absorber and condenser surfaces.

Corrosion Protection

The heat transfer surfaces of an evaporatively cooled unit are exposed to a warm, humid atmosphere. This creates a serious corrosion problem. A coating or material was needed that would eliminate the corrosion problem but also meet the requirements of good surface wettability, ease of application or construction and reasonable cost. Three approaches for protection were considered:

- 1) Stainless Steel
- 2) Galvanized Mild Steel
- 3) Mild Steel with Organic Coating

The obvious advantage to stainless steel is its inherent anti-corrosion properties. It would also eliminate the problems involved in applying coatings. A section of absorber surface was fabricated of stainless steel and after proper treatment provided good surface wetting. However, the extra cost involved in heat treating after welding and the high cost of the material made stainless an unattractive choice.

Galvanized mild steel would also provide good corrosion resistance. The concern was application problems due to the high temperature involved and the size and construction of the parts to be coated. To check this out, a full size outer absorber was galvanized. The results were good, however, as with stainless steel, the cost was too high to make it economically feasible.

FIGURE 4

EVAPORATIVELY COOLED SOLAR CHILLER
 ABSORBER/CONDENSER ASSEMBLY
 PROTOTYPE NUMBER 3

FIGURE 5

ABSORBER/CONDENSER SHELL ASSEMBLY DETAIL

Mild steel with an organic coating turned out to be the best choice. A total of eleven different organic coatings were tested in salt spray and humidity. These included various epoxy coatings, zinc-rich coatings and plastic coatings. The coating best meeting the desired requirements was a zinc-rich epoxy coating called Devcon "Z". This coating passed 2,000 hour salt spray and 4,000 hour humidity tests. The material is air dry and the resulting surface is rough which is conducive to good wetting. Optimum thickness is approximately 0.076 mm (.003) and the proposed application method is flow coating. Devcon "Z" was used on the three units constructed for field testing.

HERMETIC SOLUTION PUMP DEVELOPMENT

The solution pump is the only part of the sealed absorption system that has moving parts. Obviously the reliability of the pump is an important factor in the overall life of the unit. How reliable the pump is depends largely on bearing life. Drawing on experience gained in the development of magnetic drive pumps for existing large tonnage equipment, a design was conceived utilizing a face to face magnetic drive with enough bearing surface to assure long life. The magnetic drive was chosen because it eliminates the need for seals. A prototype was built using a carbon-graphite bearing and permanent ceramic magnets. The magnets were glued to their mounting plates with epoxy. The diaphragm between the magnets, the two thrust bearings, and the pump shaft were made out of 316 stainless steel. The impeller and pump body were made out of mild steel. The pump was designed to provide 0.063 L/S (1 GPM) at a 3.65 M (12 foot) head with a speed of 1725 RPM. A bench test was constructed, with a fixed 3.65 M (12 foot) head, to check the pumps.

After some refinements, mainly reduction of clearance between the impeller and pump body, the proper flow was achieved. However, there was a problem with the magnets decoupling on start-up. This was attributed to the 10 cm (4 in) diameter impeller requiring too much torque. Further testing and analysis led to the decision to increase the speed to 3450 RPM. The increase in speed required a reduction in impeller diameter from 10 cm (4 in) to 5.4 cm (2-1/8 in) to maintain the same flow. The reduced impeller diameter required less torque which solved the decoupling problem. Also the bearing thrust was reduced because weaker magnets could be used. See Figures 6 and 7.

At this point two prototype pumps were built for life testing. The life tests simulate, as closely as possible, the conditions in the unit. After eight months of continuous operation, the pumps continue to perform satisfactorily.

FIGURE 6

SOLUTION PUMP DETAIL - ARMY

FIGURE 7
SOLUTION PUMP ASSEMBLY / SECTION DETAIL - ARKI A

UNIT PACKAGE DEVELOPMENT

Solution Pump and Cooling Water Pump Development

The initial pump mount design was a belt drive system using one 48 frame motor to operate the solution pump, condensing water pump and the chilled water pump. In reviewing the application of the cooling unit in relation to the unitary Solar Package, it was decided the chilled water pump should be located elsewhere in the system. It was also recognized that minimum power usage was an important criteria for designing the pump system. These considerations led to a pump mount design which utilized a double ended 33 frame motor driving both the solution pump and condensing water pump. The use of the smaller frame motor and elimination of the belt-pulley system reduced power requirements considerably. A direct drive sump pump was used for the condensing water. See Figure 8a for Arkla design pump system.

For comparison purposes, a double ended motor with magnetic drive on both the solution and water pump was obtained from Gorman-Rupp Industries, a pump manufacturer. See Figures 8c and 9. This system was evaluated with regard to flow, power usage and reliability. Flow and power usage were quite acceptable. The reliability of the condensing water pump was marginal due to build up of magnetic particles on the magnets. This information supported the original idea of using a sump pump for the condensing water.

Further testing was performed after converting the magnetic drive water pump to a direct drive sump pump. Satisfactory results were obtained. Upon comparing the Arkla and GRI pump systems, it was found that the GRI system used approximately one-half the amount of power that the Arkla system used. The direction of the Arkla pump design was basically fixed before this comparison was made. Even though considerable time was spent in developing the Arkla pump, it was decided in the interest of low parasitic power usage

that the GRI pump system should be used on the field test units. See Figure 8b. Life tests similar to those for the Arkla solution pump were set up for the Gorman-Rupp pump. The GRI pumps have not been on test long enough yet to make an accurate comparison with the Arkla pump.

Use of the more efficient Gorman-Rupp pump, along with eliminating the belt pulley system and going to a 33 frame drive motor, reduced power requirements of the pump package from 275W to 100W.

Evaporative Cooling Component Development

The components necessary for evaporative cooling and how they fit together largely determines the final configuration of the overall unit package. In this case the desire for the unit package to fit the solar systems package dictated the cooling components design.

For the unit to be compatible with the solar package, two criteria had to be met. The hot water and chilled water circuits had to have automatic drain down capabilities and the unit had to physically fit the system skid without extensive alteration to the skid. To provide drain down and match the solar systems tank connections the unit had to be increased in height. Fitting the unit to the system skid, along with the desire to reduce its bulkiness, led to some major changes in its configuration.

The air flow system on the first prototype unit incorporated a .56M (22" side mounted fan which forced air over the absorber and condenser surfaces. This type of system required an extension of the unit housing for mounting the fan, which increased the area requirements of the total unit. To eliminate the side extension it was decided to locate the fan either on the top or the bottom of the unit. The bottom location was the logical choice because the increase in height required in matching the system skid left excess space under the unit. A top mount would have increased the height even more. So the fan was mounted under the unit and forced air up through the annular spaces between the absorber and condenser shells.

Figure 8a
Arkla Vertical Pump System

Figure 8b
Gorman-Rupp Vertical Pump System

Figure 8c
Gorman-Rupp Horizontal Pump System

FIGURE 9

A one piece fiberglass water collector was used to collect the cooling water and drain it into a closed sump. This concept as opposed to an open sump with the water draining directly into it, was used to keep the fan motor in a drier atmosphere. The collector also served as the fan venturi opening which was reduced to 0.5M (20 in) for the bottom mount. The V-shape of the collector trough, along with a water baffle directly over the fan and the reduced diameter of the opening helped minimize water splash out of the fan opening when the fan was off. See Figure 10.

The cooling water was distributed over the surfaces by a rotating header mounted at the top center of the condenser. Motion of the header is accomplished through reaction as water leaves the header.

Use of the bottom mount fan and the reduction in unit diameter via the decrease in absorber and condenser surfaces, reduced the unit base dimensions from 0.99M x 1.49M (39" x 59") to .92M x .92M (36-1/2" x 36-1/2"). That puts the unit size in relation to the skid within acceptable limits. Reducing the fan diameter from 0.56M (22") to 0.5M (20") reduced power requirements from 360W to 300W.

Unit Development

There were five units involved in completing the requirements of the contract covered by this report. They will be referred to as prototypes 1, 2, 3, 4, and 5. Prototype #1 was the crude unit resulting from the feasibility study done under NSF Grant GI-44095. Prototype #2 reflected the first absorber and condenser size reduction and prototypes 3, 4, and 5 were the final test units.

Prototype #1

This unit was the starting point for the development work previously discussed. Performance testing of this unit provided the information necessary for the first absorber and condenser surface reduction.

FIGURE 10
SCHEMATIC OF WATER COLLECTOR & SUMP SYSTEM

Prototype #2

This unit reflected the first decrease (23%) in absorber and condenser surface area. Results of performance tests led to a further reduction in absorber and condenser surface. The feasibility of the new absorber and condenser surface configuration was demonstrated by this unit and it was the first to utilize Devcon "Z" for corrosion protection. Following is a list of the unit components and their status compared to prototype #1.

Generator - No Change

Condenser - Reduced surface area (23%) and additional annular vessel added as condenser surface. Also incorporated new sinusoidal surface configuration. Rollers were developed to fabricate the sinusoidal configuration without excessive stretching of the metal. The corrugations were rolled one at a time.

Preheater - Coiled tighter due to reduction in condenser diameter.

Absorber - Reduced surface area (23%) and incorporated new surface configuration.

Evaporator - No Change

Liquid Heat Exchanger - Same basic design but larger.

Storage Chamber - Minor change in configuration.

Pump System - For convenience used separate motors for the solution and water pump. Used Crane solution pump and Gorman-Rupp water pump.

Cooling Fan - 0.56M (22") side inlet

Cooling Water Collector System - Water drained directly into open sump.

Water Distributor - No Change

Prototype #3

This unit reflected the second and final decrease (14%) in absorber and condenser surface and the work done under the unit package development. Following is a list of the unit components and their status compared to prototype #2.

Generator - No Change

Condenser - Surface area reduced 14%

Preheater - Coiled tighter due to decrease in inner condenser diameter.

Absorber - Surface area reduced 14%

Evaporator - 7.62 cm (3") reduction in diameter via 14% decrease in absorber surface area.

Liquid Heat Exchanger - No Change

Storage Chamber - The external storage chamber was eliminated and the annular absorber vessels served as solution storage volume.

Pump System - Used the Gorman-Rupp horizontal double ended system.
See Figure 8c.

Cooling Fan - 0.5M (20") bottom mount top discharge

Cooling Water Collector System - Used water collector which drained into closed sump.

Water Distributor - No Change.

Purge Pump - The function of the purge pump is to pump non-condensable gases produced in the unit into a storage chamber where they will not affect performance. Prototype #3 was equipped with such a pump.

Prototypes 4 & 5

The major changes in these units as compared to #3 were changing from the horizontal motor and pump configuration with a magnetic drive water pump to a vertical configuration with a direct drive sump pump (See Figure 8b) and lengthening the outer absorber vessel 12.7 cm (5") to provide more solution storage volume. The final unit package dimensions were 0.92 M x 0.92 M x 1.97 M (36-1/2" x 36-1/2" x 77-1/2"). See Figures 11, 12, 13 and 14 for completed unit. See Figure 15 for schematic.

The units were tested in a room where outside air is brought through ducts where its temperature and humidity are automatically adjusted to the desired values $\pm 0.28^{\circ}\text{C}$ (0.5°F) before it is introduced into the room. The exhaust is located in the ceiling and the unit is located directly under it. Air volume through the unit was determined by a velocity traverse of the annular openings between the absorber and condenser vessels. Hot water was supplied to the generator of the unit by a hot water boiler located outside the test room. The flow rate of the hot water was measured by a Rotameter and the temperatures in and out were measured with calibrated thermometers graduated to 0.11°C (0.2°F) and readable to 0.05°C (0.1°F). Chilled water flow and temperature were measured likewise. The evaporator load was supplied by a fan coil. The ambient design conditions for the unit are 35°C (95°F) dry bulb and 25.6°C (78°F) wet bulb.

Typical test results for prototypes 1, 2 and 3, at design conditions are shown in Table I. Performance at off-design conditions should follow the curves shown on page 30. Appendix "A" contains calculations to help illustrate the heat transfer characteristics of the absorber and condenser surfaces.

Figure 11
Front View
Evaporatively Cooled Solar Chiller

Figure 12
Back View
Evaporatively Cooled Solar Chiller

Water Collector

Water
Sump

Figure 13
Lower Front View
Evaporatively Cooled Solar Chiller

Figure 14
Top View
Evaporatively Cooled Solar Chiller

FIGURE 15¹

SCHEMATIC OF EVAPORATIVELY COOLED SOLAR CHILLER

Table 1
PROTOTYPE #1

<u>DB</u>	<u>Ambient WB</u>	<u>Vol</u>	<u>Evap</u>	<u>In</u>	<u>Chilled Water Out</u>	<u>Rate</u>	<u>Cooling Capacity</u>	<u>In</u>	<u>Hot Water Out</u>	<u>Rate</u>	<u>Heat Input</u>	<u>COP</u>
35°C (95°F)	25.6°C (78°F)	0.99 m ³ /s (2100 CFM)	5°C (41°F)	13°C (55.4°F)	7.3°C (45.2°F)	0.45 L/S (7.2 GPM)	10.7 KW (36,720 Btuh)	89.7°C (193.4°F)	84.1°C (183.4°F)	0.69 L/S (11 GPM)	15.5 KW (53,394 Btuh)	0.687

PROTOTYPE #2

35°C (95°F)	25.6°C (78°F)	.868 m ³ /s (1840 CFM)	6°C (43°F)	12.7°C (55°F)	7.2°C (44.9°F)	0.45 L/S (7.2 GPM)	10.6 KW (36,360 Btuh)	88.5°C (191.3°F)	83.3°C (181.9°F)	0.69 L/S (11 GPM)	14.7 KW (50,580 Btuh)	0.719
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PROTOTYPE #3

35°C (95°F)	25.6°C (78°F)	.625 m ³ /s (1325 CFM)	6°C (43°F)	12.7°C (54.9°F)	7.2°C (44.9°F)	0.45 L/S (7.2 GPM)	10.5 KW (36,000 Btuh)	89.7°C (193.4°F)	84.4°C (184°F)	0.69 L/S (11 GPM)	14.5 KW (49,992 Btuh)	0.720
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CAPACITY VS GENERATOR INLET WATER TEMPERATURE SOLAR FIRED CHILLER (XWF 3600)

Generator Input Flow 11 GPM
Evaporative Cooling Water Flow 6 GPM
Chilled Water Temperature Out 45°F

CONCLUSION

The result of this project is an evaporatively cooled absorption chiller that is optimized for use in a unitary solar heating and cooling system. Historically, air cooled equipment is favored over water cooled equipment because the need for dealing with the cooling water is eliminated. However, air cooled equipment requires approximately 112.7°C (235°F) firing water, where water cooled equipment can operate with 90.5°C (195°F) firing water. This means that the air cooled unit would require a pressurized solar collector system. Also the COP of an air cooled unit is lower, approximately 0.60 as compared to 0.72 for water cooled. When these facts are considered the inconveniences of water cooling do not seem so objectionable. In fact, the probability is that water cooled equipment will turn out to be the most practical.

In conclusion, it is felt that the unit developed in this project is a practical, manufacturable unit and that work should continue to keep the technology developed to date, current with the needs of the overall energy conservation program.

APPENDIX A

Calculations

The air flow across the annular vessels is $.625 \text{ m}^3/\text{s}$ (1325 CFM). Using a total heat rejection rate of 90730 kJ/hr (86000 Btu/hr), the increase in air enthalpy may be calculated.

$$\Delta H_{\text{air}} = \frac{Q_r}{m_{\text{air}}}$$

$$\Delta H_{\text{air}} = \text{Increase in air enthalpy} \quad \text{kJ/kg}$$

$$Q_r = \text{Total heat rejected} \quad \text{kJ/hr}$$

$$m_{\text{air}} = \text{Mass flow rate of the air} \quad \text{kg/hr}$$

$$\Delta H_{\text{air}} = \frac{90730}{2469.9} = 36.7 \text{ kJ/kg (16 Btu/lb)}$$

Using the value of ΔH_{air} above, figure A-1 was constructed. The average enthalpy difference between the surfaces and the air, and the equation involving heat and mass transfer were used to calculate the value of the Lewis Relationship.

$$Q = KA \Delta H$$

$$Q = \text{Heat rejected kJ/hr}$$

$$K = \text{Lewis Relationship} = h/S$$

$$h = \text{Heat Transfer Coefficient} \quad \text{kJ/hr m}^2 \text{ } ^\circ\text{C}$$

$$S = \text{Humid Heat of Air} \quad \text{kJ/kg } ^\circ\text{C}$$

$$A = \text{Heat Transfer Surface Area m}^2$$

$$\Delta H = \text{Driving Force or Difference In Enthalpy kJ/kg}$$

$$90730 = k(13.3)(47.7)$$

$$k = 143 \text{ kg/hr m}^2 \text{ (29.2 lb/hr ft}^2\text{)}$$

The heat transfer coefficient may be determined by multiplying the Lewis Relationship by the humid heat of the air.

FIG. A-1

SPECIFICATIONS AND PHYSICAL DATA

XWF 3600

Design Delivered Capacity, Btuh	-	36,000
Design Delivered Capacity Tons IME	-	3.0

Energy Requirements

Design Hot Water Input, Btuh	-	50,000
Design Hot Water Inlet Temp °F	-	195
Design Hot Water Outlet Temp °F	-	185.9
Permissible Range of Inlet Temp °F	-	170 - 205
Design Hot Water Flow, GPM	-	11.0
Permissible Range of Flow, GPM	-	5 - 22
Electrical Voltage, 60 HZ, 1 PH	-	115
Maximum Wattage Draw	-	400

includes
solution pump
condensing
water pump a
condenser fa

Chilled Water Data

Design Inlet Temp °F	-	55
Design Outlet Temp °F	-	45
Design Flow, GPM	-	7.2
Permissible Range of Flow, GPM.	-	4 - 13

Evaporative Cooling Data

Design Heat Rejection, Btuh	86,000
Design Water Flow, GPM	6
Total Water Consumption, GPH	20
Bleed Off Rate, GPH	10
Evaporation Rate, GPH	10
Water Sump Capacity, Gal	11.5

Service Connections

Hot Water Inlet and Outlet	1" NPT
Chilled Water Inlet and Outlet	1" NPT
Sump Make-Up	1/2" NPT
Sump Overflow	1" NPT

Physical Data

Approximate unit weight, including charge, pounds	1,020
Unit Dimensions, Inches	36.5W, 36.5D, 83H

Heat Transfer Surface Area

Absorber	-	88.3 Ft ²
Condenser	-	55.3 Ft ²

Materials of Major Components

Generator Hot Water Coil	-	Stainless Steel
Evaporator Coil	-	Copper
Solution Pump	-	Stainless Steel
Absorber	-	Mild Steel
Condenser	-	Mild Steel
Liquid Heat Exchanger	-	Mild Steel
Purge Pump and Storage	-	Mild Steel
Preheater	-	Stainless Steel

PURGE SYSTEM OPERATION

The purge pump acts as a miniature absorber - see Figure C-1 purge pump schematic. A small portion of strong-absorbent from the absorber-solution-in-circuit is supplied to the upper chamber of the purge pump. The solution is cooled by trickling down a copper coil which is supplied with cooling water from the condensing water circuit. A purge tube runs from the purge pump to the bottom of the inner absorber where the non-condensable gases are most likely to collect. Water vapor with entrained gas is drawn from the absorber through this tube. Once in the purge pump, the water vapor is absorbed by the solution. The remaining gas is picked up and pumped by an intermittent syphoning action to the lower purge pump chamber. From the lower chamber, the gas rises into the gas storage chamber where it is isolated from the rest of the unit and remains until removed through the purge valve. The solution collected in the lower pump chamber flows back to the unit solution sump.

Figure C-1

SCHEMATIC OF PURGE PUMP

FREEZE PROTECTION SYSTEM

Three areas require protection from freezing. These are - the generator water coil, the evaporator chilled water coil and the evaporative cooling water sump and pump.

The elevations of the generator and evaporator coils are set so that when the unit is mounted on an Arkla solar system skid, the coils will drain down any time the chiller is off. With no water in the coils, no freeze-up can occur. The generator hot water drains into the system tank and the chilled water drains into the fan coil tank. The system tank water is always hot and the water in the fan coil tank will be hot during the heating season, when the potential for freeze-up exists. See schematic of solar system on following page.

The evaporative cooling water always drains back into the sump can on shut down. See Figure 10 on Page 20 of the final report. Depending on where the unit is located (inside or outside) the sump water may freeze. The flexibility of the fiberglass or plastic sump, plus the space that is available for the water to expand, allows the water to freeze without causing damage to the sump. The sump pump, which is submerged in the water, has an internal volume small enough so that the expansion that occurs during freezing does not exceed the limits of the pump housing material. The water in a fiberglass sump with a sump pump submerged in the water was frozen, with no resulting damage to the sump pan or the pump.

HEATING MODE

COOLING MODE

Figure C-2

SEQUENCE AND OPERATION OF CONTROL SYSTEM - XWF 3600

The XWF 3600 Control Box contains the following components

- 1) 115V-24V Transformer
- 2) Chilled Water Pump Relay (CWPR) 24V
- 3) Condenser Fan Relay (CFR) 115V
- 4) Time Delay Relay (TDR) 24V
- 5) Fan Temperature Switch 115V
- 6) Low Temperature Safety Switch 24V
- 7) High Temperature Safety Switch 24V
- 8) Fan and Pump Motor Capacitors

When the house thermostat demands cooling, power is supplied to the 24V heater coil in the TDR. There is a time delay of approximately 45 seconds before the TDR contacts close. When this occurs, power is supplied to the CWPR coil and the CFR coil which starts the chilled water pump, condenser fan, solution/condensing water and generator hot water pump motors. The fan coil relay, which is external to the XWF 3600 box, is also energized at this time. When the thermostat is satisfied, power to the TDR heater coil and the generator hot water pump is cut. The rest of the motors continue to run until the TDR heater coil cools down and the contacts open. This takes approximately 3 minutes. The purpose of the 3 minute spin down period is to use some of the residual refrigerant in the unit piping and flash chamber.

Since the refrigerant in the XWF 3600 is water, provisions have to be made to prevent its temperature from falling below 32°F. When the unit is in operation, the fan temperature switch (FTS) and the low temperature safety switch (LTS) sense the refrigerant temperature at a location in the unit where it reaches its lowest point. The FTS is calibrated to open at approximately 37°F. This switch cuts power to the condenser fan only. The FTS allows the unit to operate in mild ambients without experiencing frequent LTS shut downs. The LTS is calibrated to open at approximately 34°F. This switch cuts power to everything but the chilled water pump and fan coil. Both switches close on a temperature rise of 4 - 6°F.

The high temperature safety switch (HTS) senses the top outside surface temperature of the inner condenser cylinder. This location is directly above the generator vapor outlet. The switch is set to open at 120°F and close at 110°F. When the HTS opens, power is cut to the generator hot water pump, chilled water pump and fan coil.

A schematic of the control circuit may be found on the following page.

Figure C-3
Controls Schematic

APPENDIX D
SUMMARY OF HEAT EXCHANGER COATING TESTS

COMPANY NAME	COATING NAME	CURING SCHEDULE	MIL THICKNESS	SALT SPRAY	HUMIDITY
B. L. Downey	Fluidized Bed PVC Coating	Bake	3-5	696 hrs., red rust spots.	4,896 hrs. O.K.
B. L. Downey	Epoxy Primer Polyester Topcoat	Bake	3-5	264 hrs., red rust spots	2,832 hrs. two red rust spots.
3-M Corp.	EC-2290 Epoxy Resin Base	Bake	2-4	283 hrs., 1 red rust spot. 672 hrs. generalized red rust and black oxide under coating	120 hrs. many red rust and black oxide areas under coating.
3-M Corp.	XC-3935 Epoxy Resin Base	Bake	2-4	120 hrs., 2 red rust spots. 360 hrs. many red rust and black oxide areas under coating.	96 hrs., 1 red rust spot. 120 hrs., 2 red rust spots. 360 hrs., red rust and black oxide spreading under coating.
3-M Corp.	EC-2216 Modified Epoxy Base	Bake	2-4	670 hrs., coating peeling with generalized red rust.	670 hrs., coating peeling with generalized red rust.
Glidden	Epoxy Primer Epoxy Topcoat	Air Dry	4-5	4,762 hrs., 1 red rust spot on 1 of 2 panels. 7,704 hrs., same.	4,704 hrs. blistering, no red rust. 7,704 hrs., same
Dupont	-----	Air Dry	2-4	40 hrs., red rust forming.	no test.
Devcon Corp.	Devcon "Z"	Air Dry	3	2,224 hrs., small red rust spots.	4,224, hrs. ZnCO ₃ crystal formation, no rust. 7,080 hrs., same.
Devcon Corp.	Devcon "Z" Zinc Filled Epoxy	Air Dry	1.5	Slight rusting after 400 hrs.	2,904 hrs., red rust appearing.
Orville Products	CRS #23 Metallo Organic Complex Phosphate Inhibited	Air Dry	3-5	1,368 hrs., O.K.	1,368 hrs. O.K.
Orville Products	CRS #23 Metallo Organic Complex Phosphate Inhibited	Bake	3-5	When material baked, coating became brittle and flaked.	
Red Spot	2 Part Zinc rich primer	Air Dry	3	Material has slick surface which is not acceptable.	

Equilibrium Chart for Aqueous Lithium Bromide Solutions

EQUATIONS

1. $i = Ai' + B$
2. $i' = (i - B)/A$
3. $A = -2.00735 + .16976X - (3.133362 E-3)X^2 + (1.97668 E-5)X^3$
4. $B = 321.128 - 19.322X + .374382X^2 - (2.0637 E-3)X^3$
5. $\log P = C + D/(i' + 459.72) + E/(i' + 459.72)^2$
6. $i' = (-2E/(D + (D^2 - 4E(C - \log P))^{.5})) - 459.72$

TEMP. RANGE (REFRIGERANT) $0 \leq i' \leq 230^\circ F$
 TEMP. RANGE (SOLUTION) $40 \leq i \leq 350^\circ F$
 CONCENTRATION RANGE $45\% \leq X \leq 70\%$

$C = 6.21147$
 $D = -2386.373$
 $E = -337269.46$
 $i' = \text{REFRIGERANT TEMP. } ^\circ F$
 $i = \text{SOLUTION TEMP. } ^\circ F$
 $X = \text{PERCENT LiBr}$
 $P = \text{PSIA}$

Cycle Operating Points