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Design, Evaluation and Recommendation Effort Relating to the Modification of a Residential 3-Ton Absorption Cycle Cooling Unit for Operation with Solar Energy

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by

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FOREWORD

This report represents the results of work performed by the Engineering Department of Arkla Industries Incorporated for the NASA-Marshall Space Flight Center, Alabama, under Exhibit A on Contract ~~[illegible]~~ (Mod. 1).
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The NASA contract monitor for this study was Mr. R. L. Middleton of the MSFC Astronautics Laboratory.

SUMMARY

This report is concerned with the possible use of solar energy powered absorption units to provide cooling and heating of residential buildings. Both the ammonia-water and the water-lithium bromide cycles, the only ones to achieve commercial success, are considered.

With the temperature of the water supplied by the solar collector limited to 93.1°C to 112.3°C (200-240°F) it is shown that the air cooled ammonia water unit does not meet the criteria for COP and pump power on the cooling cycle and the heat obtained from it acting as a heat pump is at too low a temperature. If the ammonia machine is water cooled it will meet the design criteria for cooling but can not supply the heating needs. Code restrictions limit the ammonia unit to water chiller configuration and it must be installed outdoors.

The water cooled lithium bromide unit meets the specified performance for cooling with appreciably lower generator temperatures and without a mechanical solution pump. With no code restrictions, this unit can be installed either indoors or outdoors and can be a direct expansion unit or a water chiller.

It is recommended that in the demonstration project a direct expansion lithium bromide unit be used for cooling and an auxiliary duct coil using the solar heated water be employed for heating.

CONTENTS

INTRODUCTION

With the worsening of the fuel shortage, intensive efforts are being made to employ alternate sources of energy including solar energy. In the past, solar energy has been employed on a small scale for water heating and for heating of buildings. Now it is being studied as a possible major source of energy.

This report is concerned with a subsystem of a complete solar energy system. It involves a study of heat operated refrigeration units (absorption units) as the means of supplying cooling and possibly heating to residential buildings. The absorption unit is well suited for use with solar energy since it can use the stored heat directly without conversion and requires a minimum of electrical energy.

The original objective was a study on the modification of a 3 ton air cooled ammonia absorption unit to serve also as a heat pump. This was subsequently expanded to include the lithium bromide unit and from these studies to recommend a water fired subsystem for demonstrating residential solar heating and cooling.

These units will be considered in the order mentioned and according to the following guide lines.

1. Energy input to the unit will be in the form of water heated to 93.3°-115.5°C (200°-240°F)
2. Heating load 76.0 mJh (72,000 Btuh) at -17.7°C (0°F)
   Cooling load 38.0 mJh (36,000 Btuh) at 35°C (95°F) ambient
3. Minimum electrical power
4. Coefficient of Performance for
   (a) Heating
   COP $>$ 1.5 for 4.4°C (40°F) ambient
   (b) Cooling
   COP $>$ 0.5 at 35°C (95°F) ambient
5. Cost to be such that it is as competitive as possible with conventional compressor equipment
6. The unit should be safe for residential usage
7. Only routine maintenance required for a goal of twenty years life.

AQUA AMMONIA UNIT

The aqua ammonia absorption unit was invented over a hundred years ago and during the late 19th and early 20th centuries was used extensively in ice plants. More recently it has been employed in small tonnage gas fired air conditioners.

Ammonia is a desirable refrigerant due to its very low freezing point and its high latent heat. The latter property results in a small amount of refrigerant for a given load and this in turn results in small amounts of absorbent required to be circulated. In addition the properties of its aqueous solutions are such that the unit can be air cooled.

Offsetting the desirable properties of ammonia is the classification of ammonia in the Refrigeration Code B9.1 (ANSI B9.1-1971, ASHRAE Standard 15-70) as a Group 2 refrigerant. Group 2 refrigerants may not be used in direct expansion units for comfort cooling. They can only be used in indirect systems. Thus present day units are designed as water chillers for outdoor installation and with the chilled water being pumped to a coil indoors. Due to the Group 2 classification of ammonia, the heat obtained from the absorber and condenser also can not be used directly. Thus if an absorption unit is to be used as a heat pump, the absorber and condenser must be converted from direct air cooled to indirect air cooled. This might be accomplished by a closed water cooling circuit. In the cooling mode, then, chilled water produced by the evaporator would be circulated to the coil in the house and heated water from the absorber and condenser circuit would be circulated to an outdoor coil for rejection to the ambient. In the heating mode the chilled water from the evaporator would be circulated to the outdoor coil and the heated water would be sent to the indoor coil. Based on this arrangement, calculations were made using the following assumptions for the cooling mode.

1. An ambient of 35°C (95°F)
2. Cooling water for absorber and condenser in the cooling mode achieves a 5.55°C (10°F) approach to ambient or the water temperature leaving the coil is 40.55°C (105°F)
3. The cooling water flow is split between the absorber and condenser so that both receive 40.55°C (105°F) entering water

4. Temperature rise in the cooling water to the condenser is 4.44°C (8°F) and that the condenser temperature is 1.11°C (2°F) above the leaving water temperature or 46.11°C (115°F)

5. The temperature of the weak absorbent* leaving the absorber is 2.77°C (5°F) above the entering water or the temperature of the solution is 43.33°C (110°F)

6. The liquid heat exchanger is 75% efficient as defined by the equation

$$\mathrm{Eff} = \frac{t_{sg} - t_{sa}}{t_{sg} - t_{wal}} \qquad (1)$$

where t_{sg} = temperature of the strong absorbent from the generator

t_{sa} = temperature of the strong absorbent to the absorber

t_{wal} = temperature of the weak absorbent entering the heat exchanger

7. The vapor leaving the analyzer is in equilibrium with the entering weak absorbent.

8. The vapor leaving the rectifier is 99.5% ammonia and the condensate from the rectifier has the same concentration of ammonia as the weak absorbent.

9. The heat of rectification is imparted to the weak absorbent prior to the weak solution entering the liquid heat exchanger.

10. Heat exchange between liquid refrigerant and vapor raises the temperature of the vapor to 26.8°C (80°F)

11. Chilled water supply of 0.455 liters per second (7.2 gallons per minute) at 7.2°C (45°F) with a 5.55°C (10°F) rise.

*In previous years it was common to refer to the weak absorbent as rich liquid, it being rich in refrigerant. This practice resulted in confusion when referring to the lithium bromide unit and ASHRAE has adopted the system in terms of absorbent.

12. Hot water flow to the generator is such that a 5.55°C (10°F) drop in its temperature occurs in the generator.

Using the previous assumptions, and the chart of properties of ammonia solutions* six cases of operation in the cooling mode were considered and are shown in Table 1.

Case 1 assumes optimum conditions i.e. maximum temperature for the hot water supply, a close approach 5.55°C (10°F) between generator and hot water supplied and attainment of equilibrium conditions in the absorber.

Cases 2 and 3 make the same assumptions except that the hot water supply is progressively cooler. This requires an increase in the ammonia concentration of the strong absorbent, resulting in increases in solution circulated, pump power, heat exchanger load and heat input to the generator and hence a decrease in COP.

Case 4 uses the same conditions as Case 1 but more realistically assumes that equilibrium is not attained in the absorber.

Case 5 is similar to Case 4 with the exception of a lower temperature hot water supply.

Case 6 uses the same conditions as Case 5 but assumes an 11.1°C (20°F) differential between generator temperature and hot water supply. It is felt that this represents a more practical condition.

For comparison, data on a commercial gas fired unit are listed as Case 7. To make the comparison equivalent to the other cases, the COP does not include flue loss.

The data in Case 6, where 110°C (230°F) water powers the generator, are not encouraging. Heat input to the generator and hence heat rejection to the ambient together with heat exchanger load and pump power are all high and the coefficient of performance is poor. If 104.4°C (220°F) water, the average of the 200-240°F range of hot water specified, had been used, the results would have been even poorer.

In addition as shown in Appendix A for a 4.45°C (8°F) rise in the temperature of the condenser cooling water, a flow rate of 0.589 L/sec (9.4 gpm) is required. Likewise if a constant approach of 2.8°C (5°F) of the solution temperature to cooling water temperature throughout the absorber is assumed, the temperature rise will be 6.1°C (11°F) and will require a flow rate of 1.286 L/sec (20.3 gpm)

When the unit is used as a heat pump with a -6.65°C (20°F) evaporator, if the same concentrations of 43.5 and 46% as shown in Case 6 are employed the equilibrium temperatures of the solutions are 37.8°C (100°F) and 33.3°C (92°F) and obviously the water cooling the absorber must be at lower temperatures.

*Appendix A

Assume instead that the cooling water flow, split during cooling operation, is combined and enters the condenser first thereby giving the lowest condenser temperature and that a concentration control of some kind permits a weakening of the ammonia content of the solution. The increased water flow will result in approximately a 1.47°C (2.6°F) rise in temperature of the water through the condenser and 4.21°C (7.6°F) through the absorber at full load. Now assume that the water entering the condenser has a temperature of 37°C (98.5°F). The water leaving the condenser will be 38.2°C (101°F) and the condenser temperature 39.4°C (103°F). With a generator temperature of 98.9°C (210°F) a strong absorbent concentration of 39.5% ammonia is obtained and using the same circulation rate as on the cooling cycle the weak absorbent will be 42.2% ammonia. Equilibrium temperatures of these solutions for a -6.67°C (20°F) evaporator are 45°C (113°F) and 40°C (104°F). The cooling water entering the absorber at 38.2°C (101°F) will rise to 42.1°C (107.8°F).

These water temperatures are far too low to permit use in a fan coil. The only possible application would be in a radiant system using several thousand square feet of surface.

As an alternative to the air cooled ammonia unit, consider the case where the unit is water cooled. Assume that the generator temperature is 87.7°C (190°F), that condensing water is supplied at 29.4°C (85°F), the weak absorbent leaves the absorber at 32.2°C (90°F), the condenser is at 40.5°C (105°F) and that the weak absorbent instead of being in equilibrium at 55% actually leaves at 53.5%. With these conditions calculations show a COP of 0.578 and a pump power of 0.094 kW (0.126 H. P.) This meets the specification for cooling performance but since the ammonia concentrations are still higher than those for the air cooled unit heating performance will be worse.

TABLE 1

COOLING PERFORMANCE OF AIR COOLED AMMONIA UNIT POWERED BY SOLAR ENERGY

	Entering Hot Water °C	(°F)	Max. Gen. Temp. °C	(°F)	Hourly Heat Input to Gen. x 10 -3 mJ	(Btu)	Hourly Heat Rejection x 10 -3 mJ	(Btu)	Conc. Strong Abs. $\%NH_3$	Conc. Weak Abs. $\% NH_3$	Pump Work kWatts	(HP)	COP
Case 1	115.5	(240)	110	(230)	70.7	(67.2)	109.0	(103.2)	37.5	47.5	0.098	0.131	0.534
Case 2	110	(230)	104.4	(220)	75.2	(71.4)	113.3	(107.4)	40.5	47.5	0.132	0.178	0.505
Case 3	104.4	(220)	98.9	(210)	87.4	(83.0)	124.5	(119.0)	43.5	47.5	0.219	0.294	0.435
Case 4	115.5	(240)	110	(230)	80.3	(76.2)	118.4	(112.2)	37.5	46.0	0.128	0.172	0.472
Case 5	110	(230)	104.4	(220)	86.0	(81.5)	123.9	(117.5)	40.5	46.0	0.164	0.226	0.430
Case 6	110	(230)	98.9	(210)	119.2	(113.1)	157.2	(149.1)	43.5	46.0	0.352	0.472	0.317
Case 7*			187.2	(370)	61.1	(58.0)	99.1	(94.0)	9.0	41.0	0.056	0.075	0.623

*Data for standard gas operated unit.

WATER-LITHIUM BROMIDE UNIT

The water-lithium bromide combination is the only other cycle to enjoy any degree of commercial success. It was used for a number of years in gas fired residential air conditioners both as direct expansion units and as water chillers and is still used in steam and hot water fired commercial water chillers.

Water has the highest latent heat of any refrigerant (over double that of ammonia) and hence the smallest amount is required to produce a given refrigeration effect. It is a safe refrigerant and has no code limitations; these units can be installed indoors or outdoors. The units operate under vacuum and the pumping power (if a mechanical pump is used) is quite low. In addition the units have a good COP, 0.65, and maintain this over a range of partial inputs, thus minimizing the amount of auxiliary heat that may be required. A drawback is that water freezes at 0°C (32°F) and thus the units can not be used as heat pumps. Also the units must be water cooled; however this is not without benefit in that this results in generator temperatures that are low enough to be operated with hot water at temperatures below the normal boiling point. This reduces the performance requirements of the solar collector and also reduces losses from the hot water storage tank.

A tentative specification based on actual data for a water-lithium bromide unit with specifed capacity is given in Table 2.

TABLE 2

TENTATIVE SPECIFICATION OF A WATER FIRED, WATER-LITHIUM BROMIDE AIR CONDITIONING UNIT

Input	58.0 mJh (55,000 Btuh)	0.693 L/sec @ 98.9°C inlet water (11 gpm @ 210°F)
Output	38.0 mJh (36,000 Btuh)	5.66 m^3/sec of 26.7°C DB, 19.46°C WB inlet air (1200 cfm of 80°F DB, 67°F WB)
Heat Rejection	96.3 mJh (91,000 (Btuh)	0.63 L/sec inlet water (10 gpm @ 85°F)
Generator Temp.	86°C (187°F)	

Water-lithium bromide units in the specified tonnage are available as gas fired units. They can be converted to water fired units by changing the generator.

CONCLUSIONS

In view of the foregoing, it is apparent that until appreciably higher water temperatures are available, the air cooled ammonia-water unit is not an attractive candiate for operation with solar energy.

A water cooled ammonia unit would meet the specified conditions for cooling but would require an auxiliary coil using the solar heated water for heating.

The water-lithium bromide unit meets the cooling requirements, has a higher COP than the ammonia unit and unlike the latter does not require a mechanical solution pump. It will require an auxiliary heating coil.

Under these conditions the lithium-bromide unit is obviously preferred.

RECOMMENDATIONS

or the proof of concept experiment, it is recommended that a direct expansion ater cooled lithium bromide (3 ton) unit, modified for operation on hot water, e employed for the cooling function. For the heating function, a suitable uct coil can be employed using the solar heated water.

or simplicity of controls, it is recommended that the cooling unit be installed ndoors.

APPENDIX A
NOMENCLATURE

C_{sa}	concentration of strong absorbent wt. % NH_3
C_{wa}	concentration of weak absorbent wt. % NH_3
C_{va}	concentration of NH_3 in vapor from analyzer wt. %
C_{vr}	concentration of NH_3 in vapor from rectifier wt. %
h_{er}	heat content of condensate from rectifier mJ/kg
h_{va}	heat content of vapor leaving analyzer mJ/kg
h_{vr}	heat content of vapor leaving rectifier mJ/kg
h_{lc}	heat content of liquid NH_3 from condenser mJ/kg
h_{vrx}	heat content of vapor leaving refrig. heat exchanger mJ/kg
h_{sg}	heat content of strong absorbent leaving generator mJ/kg
h_{sa}	heat content of strong absorbent entering absorber mJ/kg
h_{wa}	heat content of weak absorbent leaving absorber mJ/kg
h_{wa1}	heat content of weak absorbent leaving rectifier mJ/kg
h_{wa2}	heat content of weak absorbent leaving liquid heat exchanger mJ/k
t_{sg}	temperature of strong absorbent from generator °C
t_{sa}	temperature of strong absorbent entering absorber °C
t_{wa}	temperature of weak absorbent leaving absorber °C
t_{wa1}	temperature of weak absorbent leaving rectifier °C
t_{wa2}	temperature of weak absorbent leaving liquid heat exchanger °C

SAMPLE

CALCULATIONS OF PERFORMANCE OF AIR COOLED AMMONIA-WATER UNIT - CASE 1

Given Conditions:	Condenser Temperature	46.1°C (115°F)
	Evaporator Temperature	4.44°C (40°F)
	Generator Temperature	110°C (230°F)
	Vapor Leaving Refrig X	26.6°C (80°F)
	Weak Absorbent Temp. from Abs.	43.2°C (110°F)

For other assumptions, see p. 2.

From the 1972 ASHRAE Handbook of Fundamentals

h_{vrx} = 1495 mJ/kg (646 Btu/#)
h_{lc} = 402 mJ/kg (173 Btu/#)

The net refrigerating effect of 1 kg of NH_3 under the given conditions is 1495-402 = 1093 mJ (473 Btu/#). A ton of refrigeration requires 12650 mJ per hr and therefore 3 tons requires 37,973 mJ/hr (36000 Btu/hr).

$$\therefore \frac{37973}{1093} = 34.7 \text{ kg } NH_3\text{/hr. } (76.2\#\text{/hr})$$

Given the condenser and generator temperatures, the concentration C_{sa} of the strong absorbent is determined from the chart of properties of ammonia solutions. From the same chart given the temperature of the evaporator and of the solution leaving the absorber, the concentration C_{wa} of the weak absorbent is determined. Thus C_{sa} is found to be 37.5% and C_{wa} is 47.5%. From these values, the relative pumping (weight of strong absorbent circulated per unit weight of refrigerant) is calculated using the equation

$$\text{Relative pumping} = \frac{1- C_{wa}}{C_{wa}-C_{sa}}$$

$$\text{R.P.} = \frac{1- 0.475}{0.475-0.375} = 5.25$$

From the chart the following values are then determined.

h_{va} = 1450 mJ/kg (626 Btu/#)

C_{va} = 0.976

h_{vr} = 1360 mJ/kg (586 Btu/#)

C_{vr} = 0.995

C_{lr} = 0.475

h_{lr} = 165 mJ/kg (71 Btu/#)

If for each unit of refrigerant vapor leaving the rectifier (1 + X) units of vapor enter and X units of condensate are produced the following material balance can be made

$$(1 + X)\ (0.976) = (1)\ (0.995) + (X)\ (0.475)$$
$$X = 0.0378 \text{ kg}$$

A heat balance for the rectifier can now be made to determine the heat Q removed from the vapors

$$(1.0378)\ (1450) = (1)\ (1360) + (0.0378)\ (765) + Q \text{ rect.}$$
$$Q \text{ rect} = 144 \text{ mJ/kg (62 Btu/\# refrig to cond.)}$$

Since 6.25 kg of weak absorbent (5.25 kg of strong absorbent plus one kg of refrigerant) receive this heat of rectification 144/6.25 = 23.1 mJ/kg (10 Btu/lb) are added to the weak absorbent.

From the chart

$$h_{wa} = -74.4 \text{ mJ/kg } (-32 \text{ Btu/\#})$$

Then $h_{wa1} = -74.4 + 23.1 = -51.3 \text{mJ/kg } (-22 \text{ Btu/\#})$

$$t_{wa1} = 45.6°\text{C } (114°\text{F})$$

Based on equation #1, the temperature of the strong absorbent leaving the liquid heat exchanger can now be calculated.

$$0.75 = \frac{110 - t_{sa}}{110 - 45.6}$$

$$t_{sa} = 61.8°\text{C } (143°\text{F})$$

Then $h_{sa} = 41.8 \text{ mJ/kg } (18 \text{ Btu/\#})$

also $h_{sg} = 271.5 \text{ mJ/kg } (117 \text{ Btu/\#})$

The amount of heat given up by a kilogram of the strong absorbent in the liquid heat exchanger is given by the equation

$$Q/\text{kg SA} = h_{sg} - h_{sa}$$

$$Q/\text{kg SA} = 271.5 - 41.8 = 229.7 \text{ mJ/kg } (99 \text{ Btu/\# SA})$$

The total heat transferred in the heat exchanger in mJ/hr can be calculated knowing the relative circulation rate and the total weight of ammonia required per hour. Thus $Q_{1he} = (34.7)\ (5.25)\ (229.7) = 41800$ mJ/hr (39600 Btu/hr)

This amount of heat is transferred to the weak absorbent. The heat content of each kg of weak absorbent is increased by an amount Δh given by

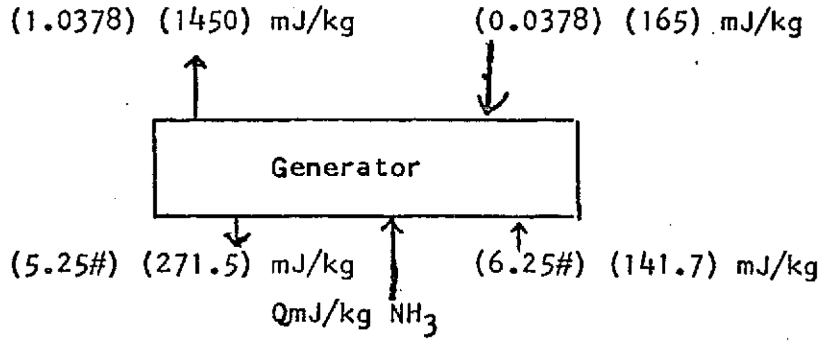
$$\Delta h = \frac{41600}{(34.6)\ (6.25)} = 193 \text{ mJ/kg } (83 \text{ Btu/\#})$$

The heat content h_{w2} of the weak absorbent entering the generator is

$$h_{w2} = h_{w1} + \Delta h = -51.3 + 193 = 141.7 \text{ mJ/kg } (61 \text{ Btu/\#})$$

From the chart t_{w2} is 86°C (187°F)

A heat balance around the generator can now be made based on a kilogram of refrigerant to the condenser



Q gen/kg NH_3 + (6.25) (141.7) + (0.0378) (165) = (1,0378) (1450) + (5.25) (271.5)

Q gen/kg NH_3 = 2040 mJ/kg (881 Btu/#)

Q gen/hr = (34.7) (2040 = 70790 mJ/hr (67200 Btu/hr)

$$COP = \frac{37793}{70790} = 0.534$$

$$\text{ump head} = \Delta p \ \frac{kg}{m^2} \ \frac{1}{d \ 47.5\% \text{ solution}}$$

$$= 13700 \ \frac{kg}{m^2} \ \frac{1}{823 \ kg/m^3} = 167 \text{ meters}$$

$$W = (167 \text{ meters}) \ (0.06 \ \frac{kg}{sec}) = 9.8 \ \frac{kg \ m}{sec} = 98 \text{ watts}$$

or in English units

$$h = \frac{(195\#/in) \ (144)}{(0.825) \ (62.4)} = 543 \text{ feet}$$

$$H.P. = \frac{(543) \ (7.93)}{33000} = 0.131$$

COOLING WATER FLOW CALCULATIONS

ondenser Load = 34.64 kg NH_3/hr. (1361-221) mJ kg = 39500 mJ/hr (37400 Btuh)

For a 4.45°C (8°F) temperature rise

$$2.39 \times 10^{-1} \frac{39500}{3600} = (4.45)\ L$$

L = 0.589 liters/sec (9.4 gpm)

bsorber Load = Total Load - Condenser Load

= (149,100) (1.054) - 39500

= 157300-39500 = 117,800 mJ

$$2.39 \times 10^{-1} \frac{117800}{3600} = (6.1)\ L$$

L = 1.286 liters/sec. (20.3 gpm)