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# SOLAR LIQUID-DESICCANT AIR CONDITIONING SYSTEM

FINAL REPORT
PREPARED FOR

DEPARTMENT OF ENERGY

APPROPRIATE TECHNOLOGY GRANTS
#DE-FG44-80R410096.000

SOLAR ENGINEERING COMPANY

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# DISCLAIMER

# I. TABLE OF CONTENTS

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## II. SUMMARY

A design for a closed, diurnal, intermittent absorption chiller for passive solar air-conditioning using liquid sorbents has been constructed and tested.

$LiBr-H_2O$ will not work with this design because of its low vapor pressure at the temperature available. The approach has possibilities using the 2 $LiBr-ZrBr-CH_3OH$ or $H_2O-NH_3$ sorbent refrigerant pairs. The use of $H_2O-NH_3$ appears to be the better candidate because of the lower solution viscosity and less cycle weight, through tank volumes and collector requirements are similar. Further study of other refrigerant pairs such as S-Thiocyanate-ammonia is indicated, however the difficulties encountered in construction and low potential coefficient of performance, and thus large collection area needed, makes commercialization of such a system doubtful in the forseeable future.

## III. A.. INTRODUCTION AND OBJECTIVES

Solar energy is naturally intermittent by its diurnal cycle. Use of this cycle as a means to produce driving temperature differentials for passive cooling has been the subject of much effort and discussion. Such differentials would also seem a perfect match for the requirements of sorbent-refrigerant combinations. Simply stated, in the heat of the day solar energy separates the sorbent-refrigerant pair. In the evening refrigerant vapor is resorbed by the cooled sorbent which provides a reduced vapor pressure that either vaporizes additional refrigerant from its container, which results in cooling, or in an open system, dehumidifies air to be later cooled by evaporative cooling.

The objective of this project involves construction of a 300 square foot solar collector panel for the development of desorption energy, the construction of evaporator and condenser coils and the performance analysis of this proposed disiccant solar air conditioning system.

## III. B.. THE INTERMITTENT MACHINE

The absorption use of the intermittent refrigeration cycle was first demonstrated by Faraday in 1824 who saturated silver chloride with ammonia gas in an enclosed apparatus (Figure 1). In such a devise, if the silver chloride cools, it reabsorbes ammonia vapor causing the previously condensed ammonia liquid to vaporize and in so doing, remove heat from its surroundings.

The use of solar energy to drive an intermittent ammonia-water absorption cycle was developed in 1958 by Williams, Chung, Lof, et al, (1). More recently Tchernev has used a zeolite-water system (2).

A system using liquid desiccant-refrigerant pairs and the three tank principal of Rombe and Foex (3) has been constructed with the intent of producing a totally passive solar incorporation into building construction. A schematic diagram of the system as initially conceived is shown in Figure 2.

## III. C.. THE INTERMITTENT SORPTION CYCLE

The thermodynamic path of the typically ideal intermittent cycle can be demonstrated by a closed loop on a vapor pressure vs. temperature chart for refrigerant-sorbent concentrations. A typical chart is shown for lithium bromide-water in Figure 3.

The regeneration phase of the cycle begins at 1 where heat is provided to the solution. The temperature of the solution is increased to point 2 where the vapor pressure of the refrigant is equal to its condensation pressure in the condenser (at a lower temperature). Continued addition of heat vaporized refrigerant (which travels to the condenser from the solution at constant pressure and also heats the drying sorbent until the final generator temperature is reached at 3 which ends the regeneration phase. For the refrigeration phase, the generator, now the absorber, is cooled to point 4 where the vapor pressure over the sorbent is equal to the vapor pressure of the refrigerant within the condenser/evaporator. The refrigerant vaporizes removing some heat from the remaining refrigerant liquid and the rest from its surroundings. The further cooling of the absorber allows the continued resorption of the refrigerant to its starting point 1. Simultaneously, as the evaporator continues to absorb heat, it returns to its initial temperature and pressure conditions.

The primary thermodynamic penalty to the intermittent cycle is apparent from the functions represented by line 1-2 and line 3-4. Continous cycling of fluids through heat exchangers can reclaim the heat removed in 3-4 to provide the heat needed for 1-2. The intermittent cycle precludes such exchange.

The theoretical refrigeration obtained from each unit weight of refrigerant cycled in the intermittent cycle is equivalent to the change in enthalpy (heat content) of the pure refrigerant from its cooled state of condensation (point 4 in Figure 5) to its final state of evaporation, point 1. Total refrigeration supplied is thus equal to the condensate mass cycled W. multiplied by the enthalpy of vaporization $h_{fg}$.

## III. D..COOLING RATIO

Defining the cooling ratio n for the cycle as the energy removed from the surroundings during the refrigeration phase 4-1 divided by the energy supplied to the generator during the regeneration phase, Sargent and Beckman (4) have shown that the cooling ratio $\eta$ is approximated by :

$$\eta = W_4 h_{fg} \Big/ m_3 h_3 - m_1 h_1 + \int_1^3 h_v \, dm_r$$

Defining operating temperature parameters suitable for an air cooled air conditioner as :

$$T_c = 90°F \qquad T_e = 40°F$$

$$T_1 = 80°F \qquad T_3 = 180°F$$

the above equation provides a theoretical upper limit of the cooling ratio for any particular absorbent-refrigerant pair (if the pair will operate within these parameters).

Assuming a well built house has a requirement of 100,000 Btu cooling per 24 your cycle, the upper portion of the equation provides an estimate of the minimum regrigerant cycle weight to meet this 100,000 Btu/cycle requirement. The cooling ratio further provides an estimate of the heat needed and thus collector area needed to meet this cooling load. The results of such an analysis are shown in Table 1 for four different liquid absorbent-refrigerant pairs.

Table 1. Analysis of Four Refrigerant Pairs to a Cooling Requirement of 100,000 Btu/day.

| | LiBr-$H_2O$ | 2LiBr-1ZnBr-$CH_3OH$ | $H_2O$-$NH_3$ | $H_2O$-$SO_2$ |
|---|---|---|---|---|
| $\eta$ | .78 | .51 | .50 | .44 |
| lbs. refrig/cycle 4-1 | 93.0 | 244 | 48 | 628.0 |
| lbs solution/cycle 1-2 | 490.3 | 1455 | 720 | 6282 |
| Collector area required @ 700 Btu/ft$^2$ · day | 483 | 280 | 286 | 1020 |

## III. E.. NATURAL PUMPING

In the design of Figure 2 the movement of the fluid from tank 1 to the generator/absorber during 2-3 is produced by the difference in pressure between tank 1 and the generator-/absorber. As the temperature increases in tank 1, the vapor pressure difference over tank 1 moves the concentrated fluid through the generator to drain into tank 2 until the reduction of free volume of tank 2 provides a pressure from non-condensables and partial pressure of the refrigerant, equal to that of the partial weights of the refrigerant in the generator.

Assuming the solutions reach equilibrium between the generator cycle 2-3 and the absorption cycle 4-1, then the dp/dt relationship of the solution in tank 1 specifically defines the required tank 1 temperature that will cause the cycling of the fluid weights specified in Table 1 through height $h_1$.

Initial operating conditions define the pressure of the generator/absorber at 2. Assuming the partial pressure of the non-condensable gas plus any partial pressure of refrigerant in tank 2 at the end of absorption 1 is equal to that needed to provide the initial condition generator/absorber pressure of the evaporator plus the required pressure to lift the last of the fluid from tank 2 to the generator/absorber through $h_2$ and using the ideal gas law for the non-condensable gas in tank 2, if $h_2$ is specified, the minimum volume of tank 2 can be calculated.

For the LiBr-$H_2O$ example, during the end of regeneration 3 the pressure needed in the generator/absorber to maintain condensation pressure in the condenser is 36.0 mm Hg. Disregarding friction losses, the pressure to move solution to the generator/absorber is the 36.0 mm Hg plus that needed to lift the solution through $h_1$. Assuming that tank 1 can be maintained at 180°, the LiBr-$H_2O$ solution only provides a vapor pressure of 118 mm Hg. The net 82 mm Hg (118-36) is enough to pump solution through an $h_1$ of 3.6 ft. When pumping from tank 2 to the generator/asorber at 4-1, the minimum pressure needed is equal to the 6.4 mm Hg with tank 2 full (partial pressure of $H_2O$ is zero) provides a lifting net pressure of 29.6 mm Hg--enough for an $h_2$ of 8.8 inches.

If the pressures available are more than sufficient for natural pumping at selected heights, the required tank 1 temperature can be reduced. These figures as well as others are summarized in Table 2.

Table 2. System Requirements for Ideal Cycle of 100,000 Btu/Cycle.

| Refrigerant Pair | Minimum Tank Size in Gal. Tank 1 | Tank 2 | Tank 3 | Minimum Tank 1 Temperature for Natural Pumping | Maximum Natural Pumping Height $h_1$ | $h_2$ |
|---|---|---|---|---|---|---|
| LiBr-$H_2O$ | 40 | 954 | 12 | 180°F | 3.6" | 8.8" |
| 2liBr-1ZnBr-$Ch_3OH$ | 135 | 608<br>(287) | 37 | 151°F | 4' | 4'<br>(2) |
| $H_2O$-$NH_3$ | 120 | 114 | 58 | 145°F | 8' | 8' |
| $H_2O$-$SO_2$ | 720 | 1007 | 56 | 124°F | 4' | 4' |

The LiBr-$H_2O$ system is not suitable for natural pumping because the low vapor pressures of the refrigerant are insufficient to pump naturally at the initial condition temperatures. The volumes and weight of toxic refrigerant and corrosive solution required clearly eliminate the $H_2O$-$So_2$ pair from consideration. For methanol, the requirement for a 287 gallon tank that provides an $h_2$of less than 2 feet could be achieved by a 1.5 diameter x8.2 ft. tank on its side, though the 1,200 lbs. of solution would have to have special supporting roof structure. For natural pumping, the proposed system would probably work best with ammonia-water pair which provides sufficient vapor pressure differentials to easily move the solutions and the temperature of tank 1 and the tank sizes are workable.

## III. F. PROBLEMS ENCOUNTERED

Besides the fact that water based pairs such as LiBr-water and $CaCl-H_2O$ would not provide enough vapor pressures, there was also difficulty with the built in absorbers leaking rain water into the interior of the collector panels This would create a high humidity in the panel air when the panels first began operating in the morning. Sealing the glass covers was never totally, and to my mind, satisfactorally achieved. The best solution we found was the use of silicone sealent at all the edges and joints.

The natural pumping system is self regulating but far from optimum. The pressure required to move solution and the time it begins to move is independent of the required condenser condensation pressure. However, an automatic valve could be placed between the solution collector panels and generator/absorber and another between tank 2 and the generator such that solution will not flow until the temperatures required for condensation or evaporation are available.

To achieve anything near the Sargent-Bechman $\eta$, every pound of solutuion must go fully from 1 through 3, the design itself prevents this. If the solution begins to move for desorption at the temperature just above 2, the solution reaching the generator/absorber will vaporize some refrigerant, but clearly will not continue to vaporize to 3 without the addition of heat. The solution must then be heated to 2' before it is allowed to enter the generator/absorber if heat of vaporization is supplied. Thus for this particular design, the maximum achievable n of the system is appropriately represented.

by:

$$\eta' = W_4 h_{fg} \Big/ m_1 (h_3 - h_1) + \int_1^3 h_v \, dm_v$$

This revises the operational requirements of Tables 1 and 2 to those shown in Table 3.

Table 3. Collector area needed for 100,000 Btu/Day cooling with $\eta'$

| | $\eta'$ | Collector 700 Btu/ft$^2$.day<br>1-2' | 2'-3 (thermosiphon) | Ft$^2$<br>Total |
|---|---|---|---|---|
| 2LiBr-1ZnBr-$CH_3OH$ | .50 | 125 | 160 | 285 |
| $H_2O-NH_3$ | .49 | 115 | 176 | 291 |

There is another consideration. Even if the wet solution enters the generator/absorber at the specified state 2' temp, the vaporization of the refrigerant from the solution removes heat carried only by the solution itself--thus to drain state 3 solution to tank 2 requires the solution be superheated beyond state 2' upon entering the generator/absorber. This superheated solution temperature requirement of 549°F, 308°F and 334°F for the LiBr-$H_2O$, 2LiBr-1ZnBr-$H_2O$ and $H_2O-NH_3$ pairs respectively is unworkable. If the solution is heated to 2' and flashed to the 2-3 pressure with the heat of vaporization provided by only the solution (dotted line of Figure 3) the $\eta$ drops for the LiBr-$H_2O$ system to .7%--far too low as over 33,000 lbs of solution would be required. The addition of a thermosiphon heat exchanger to provide the heat of vaporization to the generator/absorber from a portion of the solar panels is a workable alternative. The fluid of the heat exchanger and panels could be at atmospheric pressure but would require a higher temperature than the initial maximum 180°F temperature specified for heat exchange at the 180° requirement. The heat requirement, $m\ (h_3-h^1)$, could still be provided directly by a dedicated portion of the collector.

With the limited efficiency obtainable from the self-pumping, diurnal intermittent cycle, it is clear that further efficiency penalty of self regulation cannot be tolerated. The addition of three simple temperature on-off switches (snap switch type) could provide an effective means of regulating the cycle. Such a system is shown in Figure 4.

## IV. CONCLUSIONS AND RECOMMENDATIONS

The operation of the self-pumping, diurnal intermittent, liquid sorbent air condition system of Figure 4 is possible using the 2LiBr-1ZnBr-$CH_3OH$ or $H_2O$-$NH_3$ sorbent refrigerant pairs. The use of $H_2O$-$NH_3$ appears to be the better candidate because of lower solution viscosity and less cycle weight, though tank volumes and collector requirements are similar. Further study of other possible refrigerant pairs such as sodium thiocyanate - ammonia is indicated, however commercialization of such a system at this time is doubtful.

# V. DRAWINGS AND MATERIALS

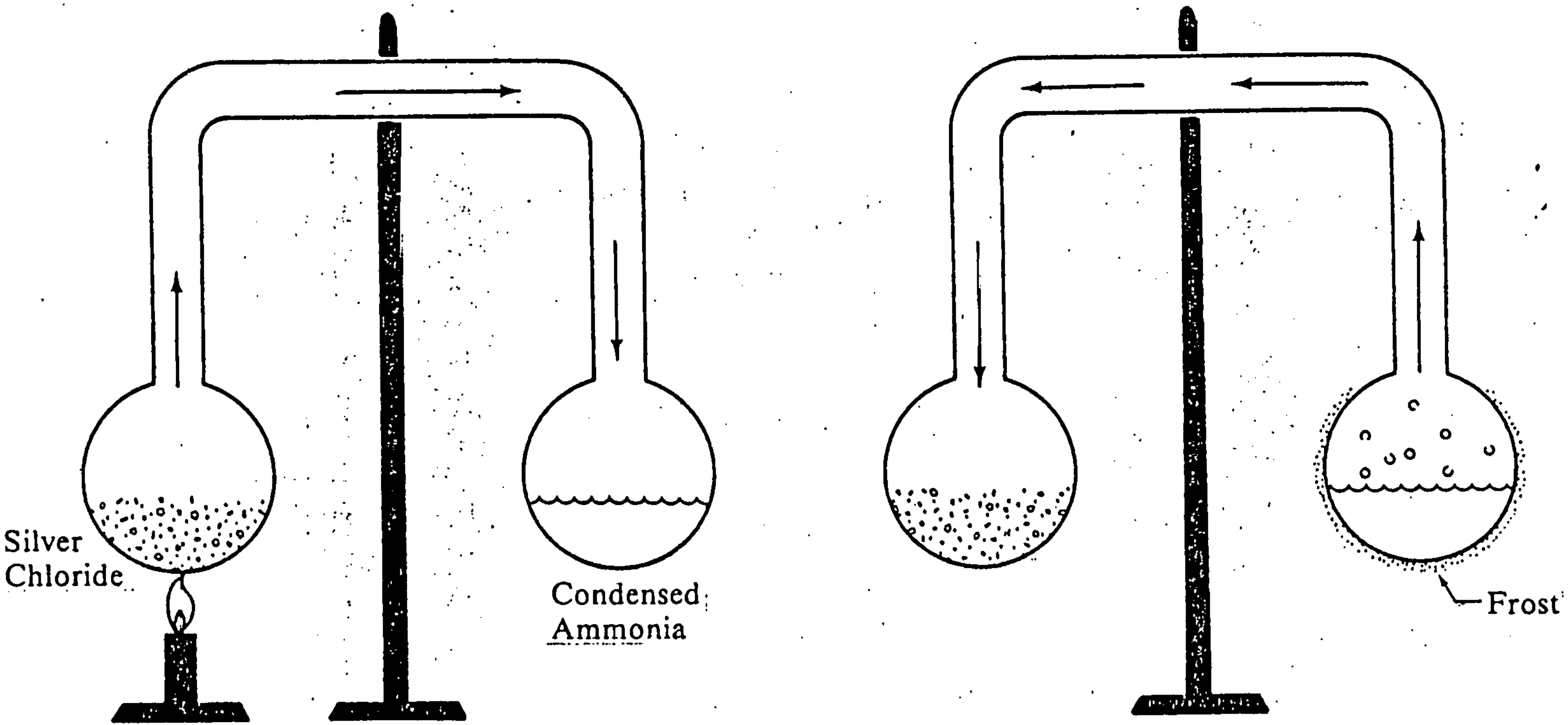

FIGURE 1. Faraday's Apparatus Demonstrating The Intermittent Sorption Cycle

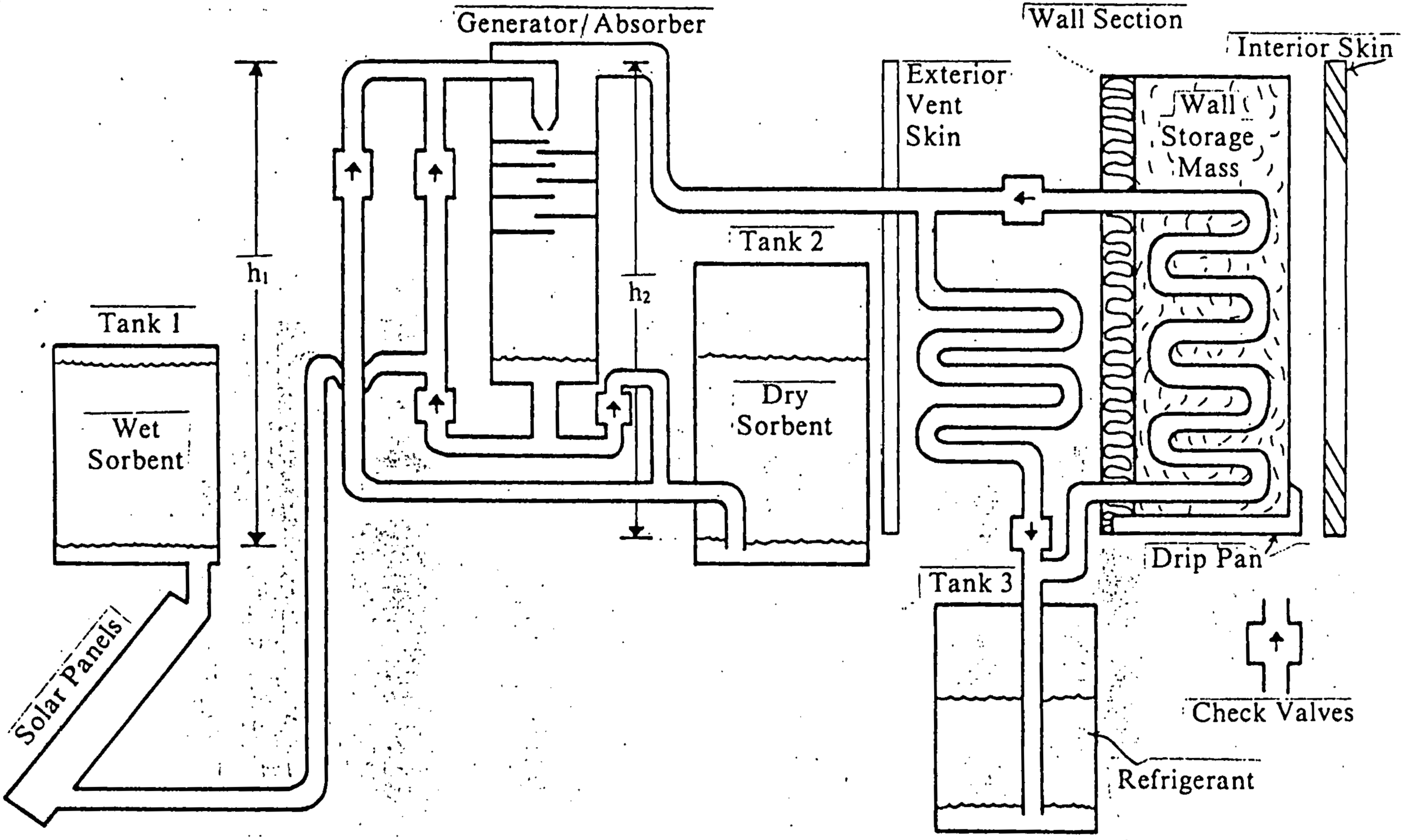

FIGURE 2. Proposed Self-Pumping Passive Solar Air-Conditioning System

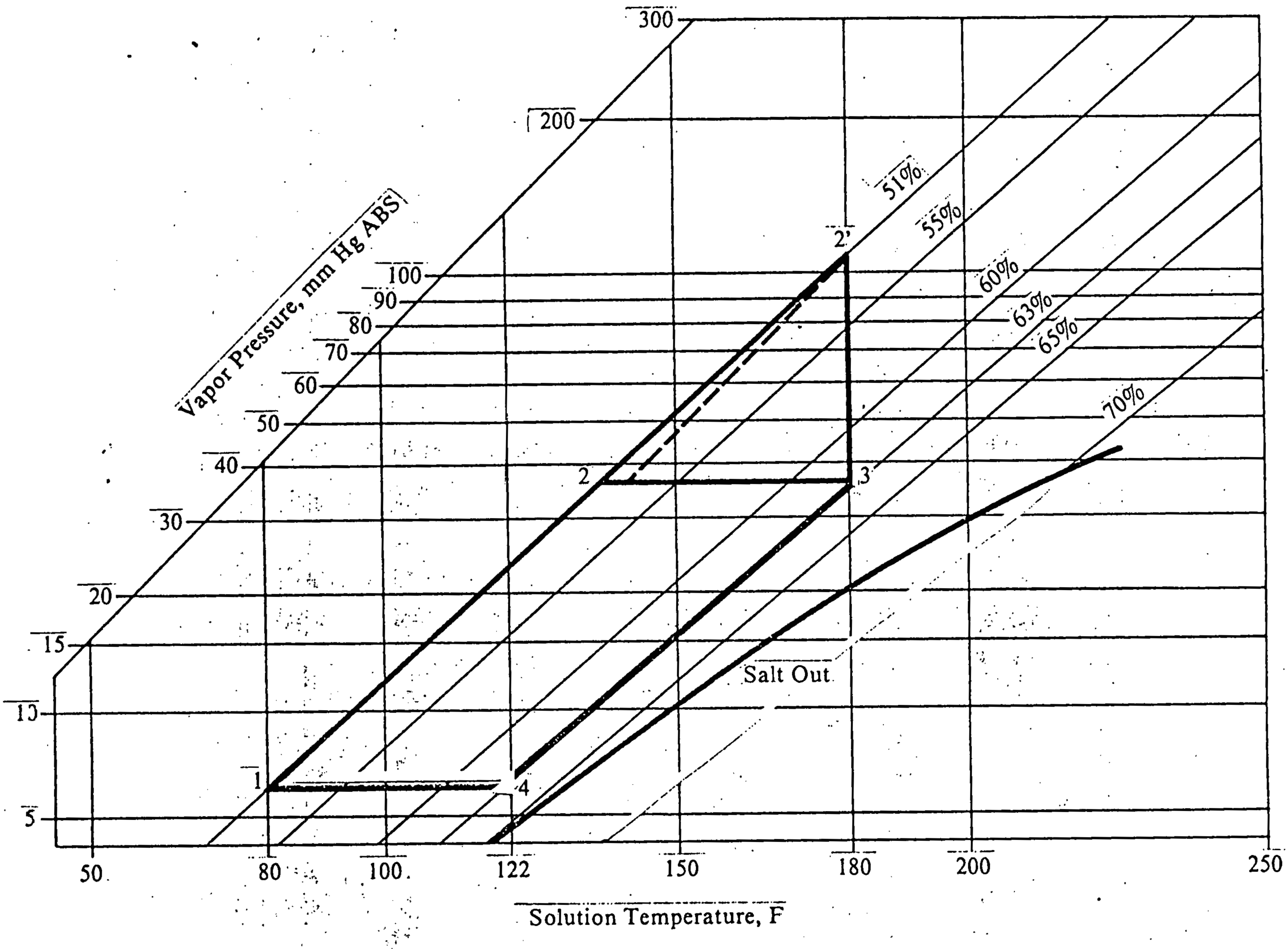

FIGURE 3. $L_iB_r$ - $H_2O$ Vapor Pressure vs. Temperature

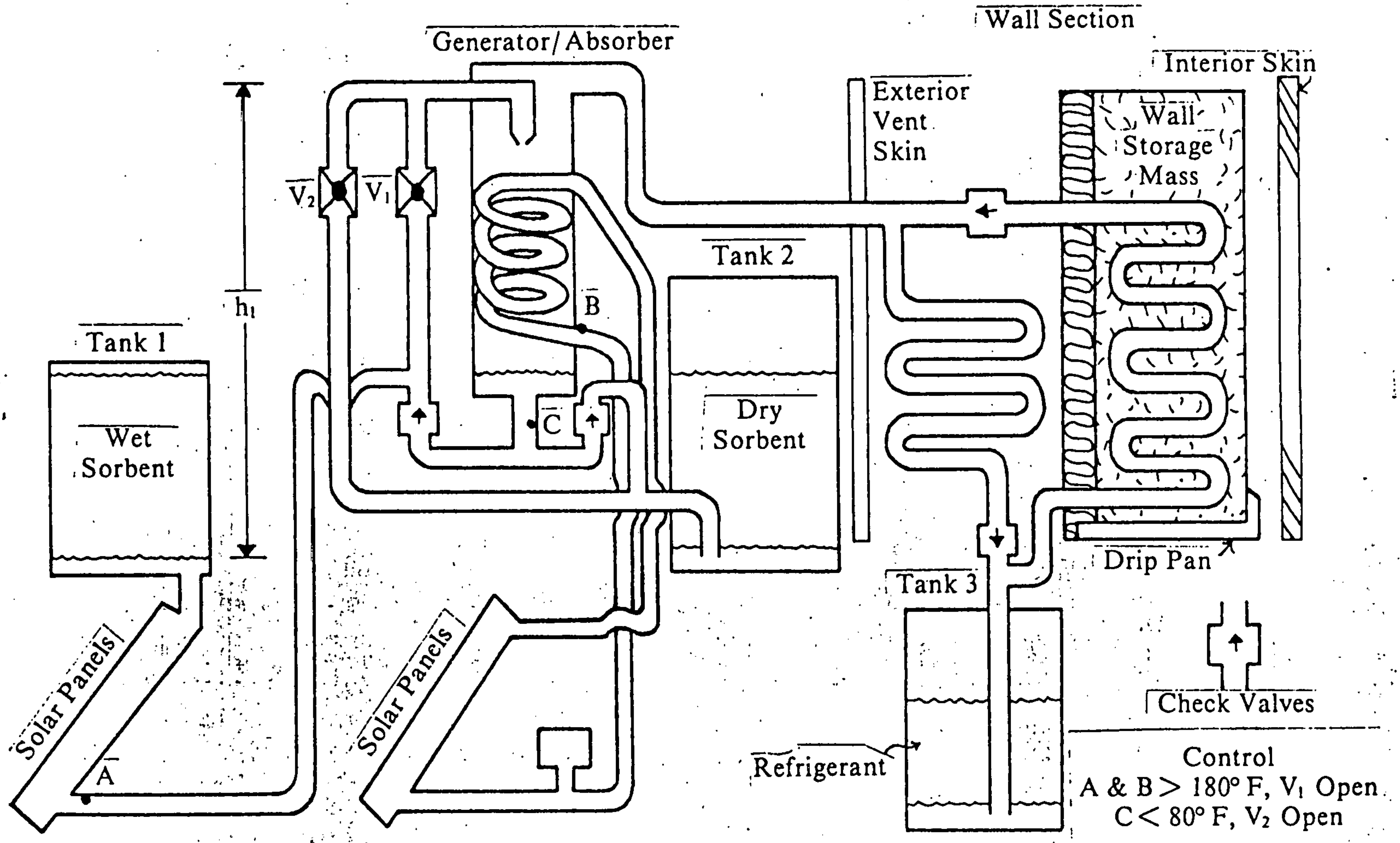

FIGURE 4. Revised Self-Pumping Passive Solar Air-Conditioning System

## V. NOTES ON MATERIALS:

Since the results of this effort indicate that either $H_2O$-$NH_3$ or sodium thiocipanate - $NH_3$ are the best working pairs for this type of system, the materials and equipment used in the furthering of this study would of necessity be of materials not effected by $NH_3$. The copper panels and copper tube used in the construction of the system utilized for this grant are attacked by the $NH_3$ and are thus not suitable for use as an operational system. Materials such as stainless steel or galvanized steel would be necessary for such a system. This would also effect the price of the system and hence the commercialability of such a system with a $NH_3$ working fluid is also in doubt.

## VI. NOMENCLATURE AND REFERENCES

Nomenclature:

$h$ = enthalpy
$h_{fg}$ = enthalpy of vaporization
$m$ = mass
$h$ = condensate mass
$n$ = cooling ratio
Subscripts:
$c$ = condenser
$e$ = evaporator
$v$ = saturated vapor
1,2,3,4, = the status represented by Figure 3.

References:

(1) B.A.Williams, R.Chung, G.O.G. Lof, D.A.Fester, and J.A.Duffee, "cooling Systems Based on Solar Regeneration". Refrigng. Engng. 66,33 (1958).

(2) B.I.Tchernev, "Intergrated Solar Zoolite Collector." _Proceedings of the Annual DOE Active Solar Heating and Cooling Contractors' Review Meeting_, March 1980, pp 50-53.

(3) F.Trombe and M.Foex, _Solar EnergyScience & Engineering_, 1, 51 (1957

(4) S.L.Sargent and W.A.Beckman, "Theoretical Performance of an Ammonia Sodium Thiocyanate Intermittent Absorption Refrigeration Cycle." _Solar Energy_, 12, 137 (1968).