

## THERMODYNAMIC BASIS FOR THE CHOICE OF WORKING FLUIDS FOR SOLAR ABSORPTION COOLING SYSTEMS

G. ALI MANSOORI and VINOD PATEL

Department of Energy Engineering, University of Illinois at Chicago Circle (P.O. Box 4348) Chicago, IL 60680, U.S.A.

**Abstract**—Through the application of the first and second laws of thermodynamics upper and lower limits for the coefficient of performance (COP) of absorption cooling cycles are derived. These upper and lower limits, besides being dependent on the environmental temperatures of components of the cycle, are also dependent on the thermodynamic properties of refrigerants, absorbents, and their mixtures. With the use of these upper and lower limits of COP it is now possible to make a quantitative comparative study of different refrigerant-absorbent combinations. The technique developed is applied for the comparative evaluation of  $\text{NH}_3 + \text{H}_2\text{O}$ ,  $\text{NH}_3 + \text{NaSCN}$  and  $\text{H}_2\text{O} + \text{LiBr}$  combinations which are the favorable candidates used in solar absorption cooling cycles.

### INTRODUCTION

Due to the critical shortage of fossil fuels in the recent years heating and cooling through the direct use of solar energy has become more attractive. Solar heating is already economical in several parts of the world while solar cooling is presently in its research and developmental stages. Cooling is needed most when the sunshine is strongest. This may be the primary motivating factor in the continuing development of solar cooling systems.

Solar energy can be utilized in two different ways for cooling purposes: One is through the use of the Rankine cycle; and the other is through the absorption cycle. The present discussion concerns the latter. The absorption refrigeration system has been in use for many years, however only recently has it been considered a means of cooling through the use of solar energy. The concept of the absorption refrigeration system was first studied by Faraday early in 1824. Eventually, this scheme of refrigeration became attractive through the use of LPG or natural gas as heat source for localities far from electricity. Beginning in the 1950s, due to abundance of electrical energy and the boom in the production of compression-expansion refrigeration systems, the absorption cooling system lost its market. However, because of the compatibility of the absorption cycles with the low-grade heat generated by flat-plate solar collectors (temperatures below boiling point of water) its application is expected to rise.

The major question in the design of solar absorption cooling cycles is the choice of working fluids (combination of refrigerant and absorbent). In the present report through the application of the laws of thermodynamics upper and lower bounds for the coefficient of performance (COP) of the absorption cycle are produced. It is shown that these upper and lower bounds in addition to being dependent on operating temperatures of the components of the cycle, are also dependent on the properties of the refrigerant, absorbent and their mixtures. As a result, it is now possible to make accurate

comparative studies of the different refrigerant-absorbent combinations which are candidates for use in solar absorption cooling cycles.

Absorption cooling systems can be classified into two types: intermittent and continuous [1-5]. Both of these types include four basic components: generator, condenser, evaporator and absorber. The intermittent type consists of two major operations which are regeneration and refrigeration. In the regeneration process, the refrigerant-absorbent solution is heated to drive the refrigerant vapor off the solution. The necessary heat can be provided by the use of solar collectors (flat-plate, cylindrical, or any other kind depending on the temperature range needed). Traveling from the generator into the condenser, the refrigerant vapor is condensed and stored. Then, the refrigeration process takes place during which the liquid refrigerant vaporizes and produces a cooling effect in the evaporator, where as the refrigerant vapor is re-absorbed by the absorbent (solvent) in the absorber. These two operations occur one after another and the cooling effect is therefore discontinuously produced. The fact that solar energy is an intermittent heat source forced the majority of the earlier investigators to use the intermittent absorption cycle which produces a less effective cooling than the continuous cooling system. Refrigeration (or air-conditioning) is generally needed in a continuous fashion. As a result, it is more advantageous to develop a continuous cooling effect. The emphasis in this report will be on continuous absorption cycles.

In a continuous absorption cycle, refrigeration and regeneration processes occur at the same time, and a continuous cooling effect is produced. The flow diagram is shown by Fig. 1. Weather conditions such as insulation ambient temperature are important factors in a solar heating and cooling system design. Performance of solar systems may change during the day and be different between a clear day and a cloudy day. Due to these factors and the lack of solar energy at night, con-

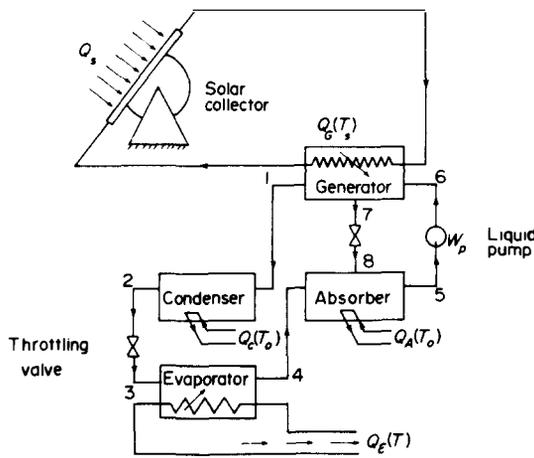


Fig. 1. Schematic diagram of a continuous solar absorption cooling system.

sideration of a storage unit for any continuous solar utilization system is essential. The solar heat is added to the generator for the purpose of continuous vaporization of the refrigerant from the solution. After the vapor refrigerant passes through the condenser, heat is removed from the refrigerant vapor and the pure refrigerant is liquefied. The liquid refrigerant then goes through an expansion valve into the evaporator wherein cooling is achieved by continuous vaporization of the refrigerant at low temperature. The vaporized refrigerant is then dissolved in the weak refrigerant-solvent solution present in the absorber which forms a strong refrigerant solution. This rich solution is then pumped back into the generator through a small liquid pump and the cycle continues. Since the recombination of the refrigerant with the solvent in the absorber is exothermic, heat must be removed from the absorber in order to maintain a sufficiently low temperature so that a high chemical affinity between the refrigerant and the weak solution can be assured. For the operation of an absorption cycle, cooling water (or air) used in the condenser and absorber should be at room temperature (or at any other available heat sink temperature). The pressure in the absorber which is equivalent to the pressure in the evaporator, and the concentration of refrigerant in the absorber, determine the temperature of the absorber. For an ideal absorption cycle, the refrigerant should be liquefied at its saturation temperature in the condenser. Therefore, the temperature of the condenser depends on its pressure. The generator pressure, which is identical to the condenser pressure, and the concentration of the refrigerant in the generator determine the minimum temperature of the generator which has to be maintained for the continuous vaporization of the refrigerant. Theoretically, the refrigerant concentration in the generator must be less than that in the absorber. In principle, four factors—evaporator temperature, condenser temperature and the concentrations of the refrigerant in the generator and absorber will determine the operating conditions of an ideal absorption cooling cycle.

In an actual absorption cycle there exists a heat-exchanger (economizer) between the generator and ab-

sorber for the purpose of recovery of heat in the flow from generator to absorber and preheating of the flow from absorber to generator. In certain cases of absorption cooling cycles, there may exist a secondary generator for a better separation of refrigerant and absorbent. However, in the present discussion the basic absorption cooling cycle, Fig. 1, is used. This is because addition of heat-exchangers, or a secondary generator, to the system introduces a number of extra unknowns which only complicate the comparative study of working fluids which is the purpose of this report.

**THERMODYNAMIC ANALYSIS OF THE SOLAR ABSORPTION COOLING SYSTEM**

The coefficient of performance (COP) of an absorption cycle is defined as the ratio of the cooling effect by the evaporator and the heat input to the generator

$$(COP)_{\text{cycle}} = |Q_E|/|Q_G| \tag{1}$$

The over-all COP of a solar absorption cooling system is defined as the ratio of cooling effect by the evaporator and the solar energy input to the collector:

$$(COP)_{\text{system}} = |Q_E|/|Q_S| \tag{2}$$

If we define the efficiency of the solar collector as the heat output divided by input heat:

$$\eta_{\text{collector}} = |Q_G|/|Q_S| \tag{3}$$

The following relation will exist between the two COP's

$$(COP)_{\text{system}} = (COP)_{\text{cycle}} \cdot \eta_{\text{collector}} \tag{4}$$

According to the first law of thermodynamics the following balance equation exists for the absorption cycle:

$$Q_G + Q_E - Q_C - Q_A + W_P = 0 \tag{5}$$

Also, according to the second law of thermodynamics the following relation can be written for the absorption cycle:

$$\sum |Q_i|/T_i \leq 0$$

or,

$$\frac{|Q_G|}{T_s} + \frac{|Q_E|}{T} - \frac{|Q_C|}{T_o} - \frac{|Q_A|}{T_o} \leq 0 \tag{6}$$

By assuming  $W_p$ , the work input to the liquid pump, negligible as compared to the other terms in eqn (5) and joining the resulting relation with Inequality (6) we get

$$\frac{|Q_G|}{T_s} + \frac{|Q_E|}{T} - \frac{1}{T_o} (|Q_G| + |Q_E|) \leq 0$$

now by consideration of the definition of  $(COP)_{\text{cycle}}$  as given by eqn (1) the above inequality can be rearranged to the following form

$$(COP)_{\text{cycle}} = \frac{|\dot{Q}_E|}{|\dot{Q}_G|} \leq \frac{T}{T_s} \left( \frac{T_s - T_o}{T_o - T} \right). \quad (7)$$

This inequality for COP can be also derived through the definition of Carnot cycle and its efficiency. The equality sign in eqn (7) holds for the COP of a Carnot cycle[1]. The right-hand side of inequality (7) gives the maximum COP which may be achieved for an absorption cooling cycle with known temperatures for the environments of the components of the cycle. The maximum COP can be achieved only when the cycle operates reversibly.

In order to relate the characteristics of the cycle to the properties of the working fluids (refrigerant and absorbent) the laws of thermodynamics for flow systems (open-systems) can be utilized[6]. According to the first law of thermodynamics for flow systems the following relations exist between the heat and work transfer rates and the properties of the working fluids in a steady state absorption cycle.

$$|\dot{Q}_G| = \dot{M}_r(h_1 - h_7) + \dot{M}_p(h_7 - h_6), \quad (8)$$

$$|\dot{Q}_C| = \dot{M}_r(h_1 - h_2), \quad (9)$$

$$|\dot{Q}_E| = \dot{M}_r(h_4 - h_3), \quad (10)$$

$$|\dot{Q}_A| = \dot{M}_r(h_4 - h_8) + \dot{M}_p(h_8 - h_5), \quad (11)$$

$$|\dot{W}_p| = \dot{M}_p(h_6 - h_5). \quad (12)$$

In the above equations  $h_1$  is the specific enthalpy of point 1 in Fig. 1,  $\dot{M}_r$  is the mass flow rate of refrigerant passing through the throttling valve and  $\dot{M}_p$  is mass flow rate of the strong solution passing through the liquid pump. Also, the following relation exists between  $\dot{M}_r$  and  $\dot{M}_p$

$$\dot{M}_r/\dot{M}_p = (X_A - X_G)/(1 - X_G) \quad (13)$$

where  $X_A$  is mass fraction of the refrigerant in the strong liquid phase coming out of absorber and  $X_G$  is for the liquid phase coming out of the generator. In deriving this equation it is assumed the vapor which is coming out of the generator is the pure refrigerant.

The following relations also exist for the isenthalpic expansion valves in the cycle

$$h_2 = h_3 \quad \text{and} \quad h_7 = h_8 \quad (14)$$

by assuming  $\dot{W}_p$ , the power input to the liquid pump, negligible the following relation will be also derived

$$h_6 = h_5. \quad (15)$$

Based on the above equations the COP of the cycle can be defined with respect to flow properties by the follow-

ing relation

$$(COP)_{\text{cycle}} = \frac{|\dot{Q}_E|}{|\dot{Q}_G|} = \frac{h_4 - h_2}{(h_1 - h_7) + \left( \frac{1 - X_G}{X_A - X_G} \right) (h_7 - h_5)}. \quad (16)$$

According to the second law of thermodynamics for open systems[6] the following relations also exist between the heat transfer rates and the properties of the working fluids in a steady-state absorption cycle.

$$|\dot{Q}_G| \leq T_s(\dot{M}_r(s_1 - s_7) + \dot{M}_p(s_7 - s_6)) \quad (17)$$

$$|\dot{Q}_C| \geq T_o\dot{M}_r(s_1 - s_2) \quad (18)$$

$$|\dot{Q}_E| \leq T\dot{M}_r(s_4 - s_3) \quad (19)$$

$$|\dot{Q}_A| \geq T_o(\dot{M}_r(s_4 - s_8) + \dot{M}_p(s_8 - s_5)). \quad (20)$$

In the above relations the equality sign is for the reversible case and the inequality sign is for the irreversible case. By joining eqns (17) and (19), we get

$$|\dot{Q}_G| + |\dot{Q}_E| \leq \dot{M}_r\{T_s(s_1 - s_7) + T(s_4 - s_3)\} + \dot{M}_p T_s(s_7 - s_6). \quad (21)$$

Also, by joining eqns (18) and (20) we get

$$|\dot{Q}_C| + |\dot{Q}_A| \geq \dot{M}_r\{T_o(s_1 - s_2) + T_o(s_4 - s_8)\} + \dot{M}_p T_o(s_8 - s_5). \quad (22)$$

Again, by assuming that  $\dot{W}_p$  is negligible as compared to the other terms in eqn (5) we can write

$$|\dot{Q}_G| + |\dot{Q}_E| = |\dot{Q}_C| + |\dot{Q}_A|. \quad (23)$$

From eqns (21)–(23) we conclude that

$$\dot{M}_r\{T_o(s_1 - s_2) + T_o(s_4 - s_8)\} + \dot{M}_p T_o(s_8 - s_5) \leq |\dot{Q}_G| + |\dot{Q}_E| \leq \dot{M}_r\{T_s(s_1 - s_7) + T(s_4 - s_3)\} + \dot{M}_p T_s(s_7 - s_6). \quad (24)$$

By dividing relation (24) by  $|\dot{Q}_G|$  and consideration of the definition of  $(COP)_{\text{cycle}}$  as given by eqn (1) we get the following relation

$$LL \leq (COP)_{\text{cycle}} \leq UL \quad (25)$$

where the lower limit,  $LL$ , is in the following form

$$LL = \frac{(\dot{M}_r/\dot{M}_p)\{T_o(s_1 - s_2) + T_o(s_4 - s_8)\} + T_o(s_8 - s_5)}{(\dot{M}_r/\dot{M}_p)(h_1 - h_7) + (h_7 - h_5)} - 1, \quad (26)$$

and the upper limit,  $UL$ , is in the following form

$$UL = \frac{(\dot{M}_r/\dot{M}_p)\{T_s(s_1 - s_7) + T(s_4 - s_3)\} + T_s(s_7 - s_6)}{(\dot{M}_r/\dot{M}_p)(h_1 - h_7) + (h_7 - h_5)} - 1. \quad (27)$$

Relations (25)–(27) can be used to calculate the upper and lower limits of the  $(COP)_{\text{cycle}}$  at different working

conditions. This means that the  $(COP)_{\text{cycle}}$  will not go out of the bounds specified by eqn (25). However it should be pointed out that the actual COP of the cycle is even less than the COP as calculated by eqn (16). Also with the understanding that inequality (7) gives the upper limit of COP regardless of the working fluids under consideration and since it is for a Carnet cycle, it is always higher than  $UL$  as given by eqn (27). As a result, we can write

$$LL \leq (COP)_{\text{actual}} \leq (COP)_{\text{eqn (16)}} \leq UL \leq \frac{T}{T_s} \left( \frac{T_s - T_o}{T_o - T} \right) \quad (28)$$

Inequalities (28) can be used for evaluation and comparison of different working fluid combinations. Before doing this, a number of preliminary criteria should be specified for eliminating those working fluid combinations which do not possess the basic and necessary physiochemical and environmental characteristics as refrigeration working fluids.

#### GENERAL REQUIREMENTS FOR THE CHOICE OF WORKING FLUIDS

##### Refrigerants

Almost all organic and inorganic compounds with normal boiling points between  $-50$  and  $100^\circ\text{C}$  are eligible for refrigerant in an absorption solar cooling cycle. The next criteria to consider in screening the refrigerants are toxicity, chemical stability and corrosivity. Also, availability and cost should be taken into account. As a criteria for toxicity the threshold limit value (TLV), which represents the maximum amount of a substance to which healthy persons can be exposed for 8 hr a day (5 days per week) without danger to health may be used. For corrosion and chemical stability of refrigerants, one should estimate the potential reactivity of these substances under typical conditions in the cycle in order to define the potential problems. The above criteria should be used generally for excluding unfavorable refrigerants. Preferably, the pressure levels in the cycle should be kept slightly above atmospheric pressure for safety and economic reasons. Also, it is preferred to choose a refrigerant with a high latent heat of vaporization per unit mass for the purpose of reducing the flow rate of refrigerant circulation. Between the compounds which qualify as refrigerants for absorption solar cooling systems  $\text{H}_2\text{O}$  and  $\text{NH}_3$  have relatively high heats of vaporization per unit mass around atmospheric pressure.

##### Absorbents

In screening of absorbents for absorption solar cooling systems stability, toxicity, corrosivity and mutual solubility with refrigerant should be considered as the criteria. The criteria for selection of absorbents should be the boiling point and melting point of the candidate. Generally normal boiling points above  $100^\circ\text{C}$  should satisfy the low volatility requirement of the absorbent. Criteria for maximum melting point of absorbent allowable depends on the crystalization possibility of the absorbent. However, some absorbents form eutectic

solutions with the refrigerants even at temperatures below the melting point of the absorbent.

##### Refrigerant-absorbent combinations

The combination of a refrigerant and an absorbent should have the following characteristics in order to qualify for use in absorption solar cooling systems: (i) Equilibrium solubility should be high at the required temperature and pressure in the absorber. (ii) Absorption should be rapid and the actual concentration of rich liquid should easily approach the equilibrium value. (iii) Pure refrigerant vapor should be obtained from rich solution as easily as possible. (iv) The absorbent should be non-volatile or very much less volatile than the refrigerant. (v) The viscosities of solutions should be low under operating conditions. (vi) Freezing points of liquids should be lower than the lowest temperature in the cycle.

In general, strong polar compounds with small molecules are preferred for absorbent-refrigerant pairs. This is because polar compounds enter reversible complex formations easily and small molecules can be more polar and tend to have higher heats of vaporization. This is why ammonia-water and water-lithium bromide pairs have been preferred over most of the other refrigerant-absorbent combinations for use in solar absorption cooling systems. Based on the above requirements for the choice of working fluids the following three refrigerant-absorbent combinations are recommended more often by different investigators[7-10]:

Table 1. Working fluid combinations examined

Refrigerant	Absorbent
1 Ammonia( $\text{NH}_3$ )	Sodium Thiocynate ( $\text{NaSCN}$ )
2 Ammonia	Water
3 Water	Lithium Bromide ( $\text{LiBr}$ )

Ammonia- $\text{NaSCN}$  combination possesses many desirable features: inexpensive, non-explosive, chemical stability, safety and non-corrosiveness in steel containers. Liquid ammonia has a high dielectric constant which would permit extensive solubility of an ionic substance like  $\text{NaSCN}$  in itself. Also, ammonia has a high heat of vaporization and a low normal boiling point. The solubility of  $\text{NaSCN}$  in ammonia is so large that it is possible to have their solution at room temperature with less than one atmosphere pressure, while the vapor pressure of liquid ammonia at the same temperature is ten atmospheres. Experimental thermodynamic properties of ammonia- $\text{NaSCN}$  solutions which include solubility data, vapor pressures, temperature-concentration chart, viscosity data, heat of solution and heat capacity data are available[11].

Ammonia-water system is one of the oldest combinations successfully employed in industrial air-conditioning and refrigeration, powered by high-temperature gas burners as the heat source[1].

Ammonia possesses very desirable properties as a refrigerant. Water is a proper absorbent due to its low

cost, availability, non-toxicity. However, because of the high affinity between ammonia and water, although their boiling points are spread apart by about 133°C, some water always vaporizes from the generator along with ammonia which will require a secondary generator (or rectifying column) to be trapped. Thermodynamic properties of this mixture is abundantly available [1].

Water–LiBr combination is another favorite candidate as working fluid for solar absorption cooling cycles. Water in this case is the refrigerant which because of its high heat of vaporization is considered as an appropriate refrigerant. However, this combination is corrosive to the materials of construction of the cycle. Hydrogen is the product of the corrosion reaction. Hydrogen increases the systems pressure which lowers the COP of the cycle. Thermodynamic properties of this mixture are also available [1].

In the present study three working fluid combinations of Table 1 are used and the thermodynamic characteristics of the absorption cycles possessing these working fluid combination are examined.

#### THERMODYNAMIC REQUIREMENTS FOR THE CHOICE OF WORKING FLUIDS

The major thermodynamic parameters affecting the performance of an absorption solar cooling system are the following:

(i) Solar collector efficiency as defined by eqn (3).

(ii) Coefficient of performance (COP) of the cycle as defined by eqn (1). Equation (16) can be used to calculate  $(COP)_{\text{cycle}}$  provided that working fluids characteristics at different parts of the cycle are known. Also, inequalities (28) can be used in order to calculate the upper and the lower bounds of the actual (COP) of the cycle at different working conditions.

(iii)  $\dot{M}_i/\dot{M}_p$  the ratio of the mass flow rate of refrigerant throttled per unit mass of solution pumped from low pressure to high pressure through the liquid pump. Larger values of this ratio are sought in the optimum design of absorption cooling cycles. In an actual absorption cycle  $\dot{M}_i/\dot{M}_p$  is an independent variable, which can be as large as desired if adequate pumping and heat exchange is provided. In any case eqn (13) for  $\dot{M}_i/\dot{M}_p$ , which is a result of mass balance around the generator, will hold. In the present study eqn (13) is used in order to calculate the maximum value of  $\dot{M}_i/\dot{M}_p$  for each working fluid combination. This is achieved by choosing the largest possible value for  $X_A$  and the smallest possible value for  $X_G$  based on equilibrium or solubility limitations at the pressure and temperature conditions of the absorber and generator. With this procedure for calculation of  $X_A$  and  $X_G$  and application of eqn (13) the upper limit of  $\dot{M}_i/\dot{M}_p$  will be derived which is a dependent variable of the working fluids under consideration. Consequently  $\dot{M}_i/\dot{M}_p$  can be used as a criteria for the comparison and choice of working fluids. It should be however pointed out that the lower limit of  $\dot{M}_i/\dot{M}_p$  is zero.

(iv) The pumping work of the strong solution from low pressure in the absorber ( $P_L$ ) to the high pressure in the generator ( $P_H$ ) per ton of refrigeration.

$$\frac{\dot{W}_p}{\dot{Q}_E} = \frac{\dot{M}_p}{\dot{M}_i} \cdot \frac{1}{\rho_A} \cdot \frac{(P_H - P_L)}{(h_4 - h_3)} \quad (29)$$

In this relation  $\rho_A$  is the density of the strong solution which is pumped from the absorber to the generator. Smaller values of this ratio are sought in the optimum design of absorption cooling cycles.

In the comparative study of working fluid combinations (COP), upper bound of  $\dot{M}_i/\dot{M}_p$  and  $\dot{W}_p/\dot{Q}_E$  should be used as the criteria. For the computations performed in this report two different cooling temperatures, one for refrigeration (36°F or 2.2°C) and another for air-conditioning ( $T = 45^\circ\text{F}$  or  $7.2^\circ\text{C}$ ), are employed. The solar collector temperature,  $T_s$ , is varied from 75 to 100°C which is the expected range of temperature attained through flat-plate solar collectors. For each  $T_s$  and  $T$  the environmental temperature,  $T_o$ , is varied from 21 to 38°C (70–100°F).

On Figs. 2–7 the COP of the cycle as defined by eqn (16), its upper limit as given by eqn (27), and its lower limit as given by eqn (26) for the three working fluid combinations chosen are reported versus solar collector temperature  $T_s$  at different environmental temperatures  $T_o$ . The dashed areas on Figs. 2–7 indicate the expected ranges of the actual COP of the cycle according to inequality (28). These dashed areas are bounded by the  $(COP)_{\text{cycle}}$  as defined by eqn (16) from the above and  $LL$  as defined by eqn (26) from the below. Figures 2–4 are

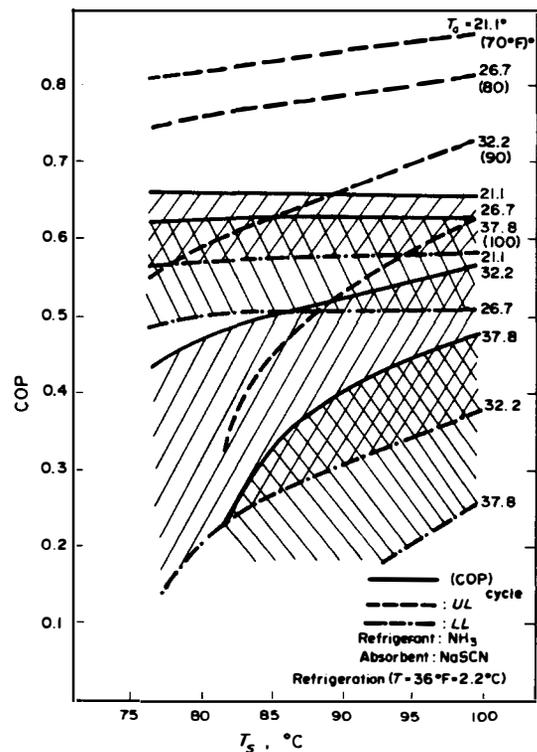


Fig. 2. COP of the  $\text{NH}_3 + \text{NaSCN}$  absorption cycle, its upper-limit (UL) and lower-limit (LL) vs solar collector temperature ( $T_s$ ) for refrigeration at different environmental temperatures ( $T_o$ ). The dashed areas are the expected ranges of the actual COP of the cycle.

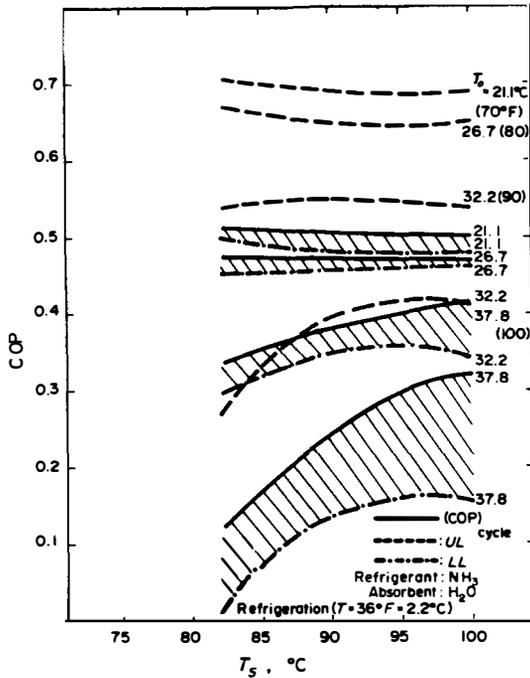


Fig. 3. COP of the  $\text{NH}_3 + \text{H}_2\text{O}$  cycle and its upper and lower limits vs solar collector temperature for refrigeration at different environmental temperatures. The dashed areas are the expected ranges of the actual COP of the cycle.

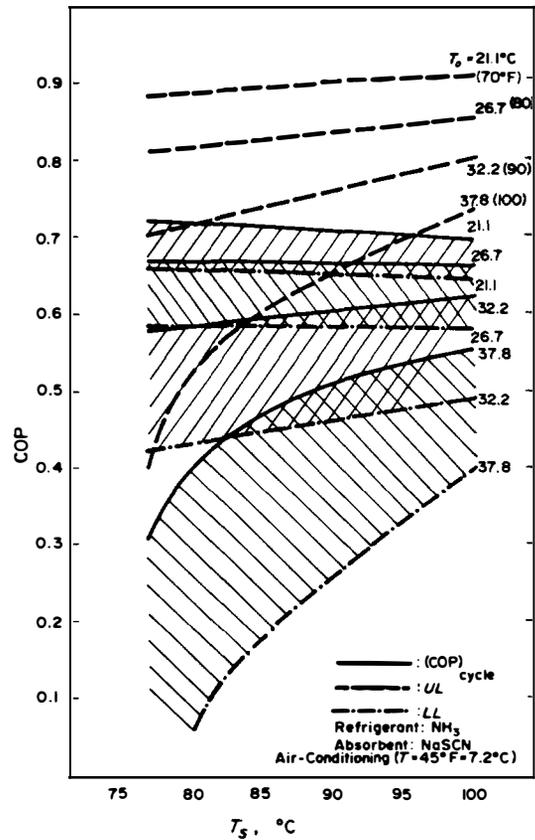


Fig. 5. COP of the  $\text{NH}_3 + \text{NaSCN}$  cycle and its upper and lower limits vs solar collector temperature for air-conditioning at different environmental temperatures. The dashed areas are the expected ranges of the actual COP of the cycle.

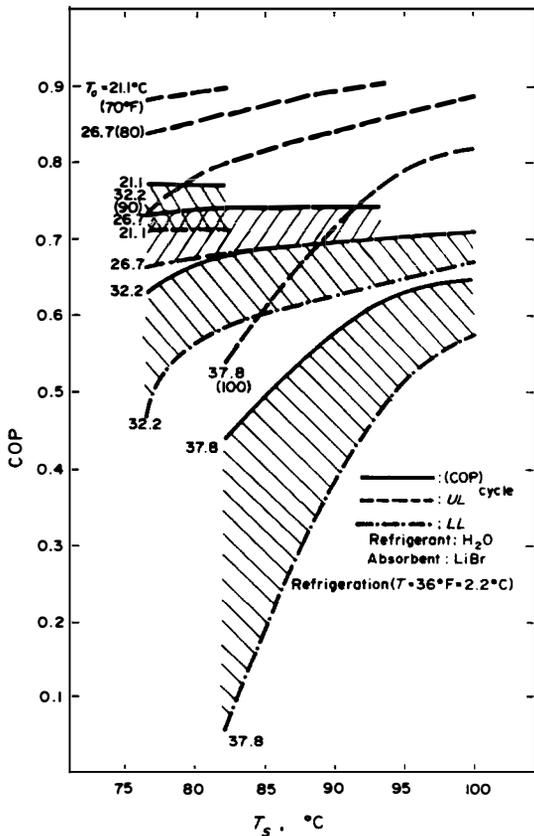


Fig. 4. COP of the  $\text{H}_2\text{O} + \text{LiBr}$  cycle and its upper and lower limits vs solar collector temperature for refrigeration at different environmental temperatures. The dashed areas are the expected ranges of the actual COP of the cycle.

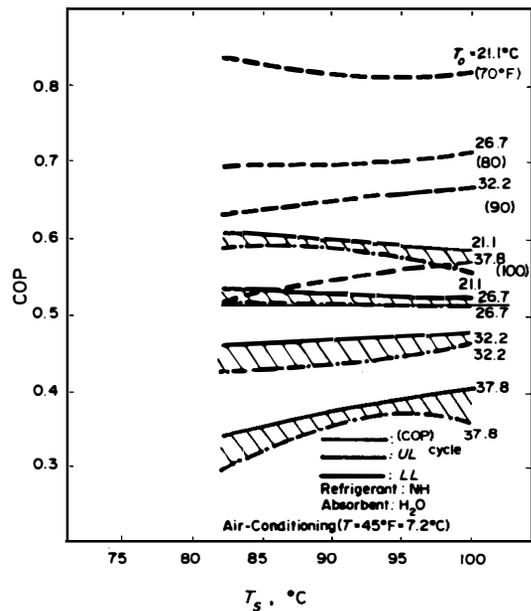


Fig. 6. COP of the  $\text{NH}_3 + \text{H}_2\text{O}$  cycle and its upper and lower limits vs solar collector temperature for air-conditioning at different environmental temperatures. The dashed areas are the expected ranges of the actual COP of the cycle.

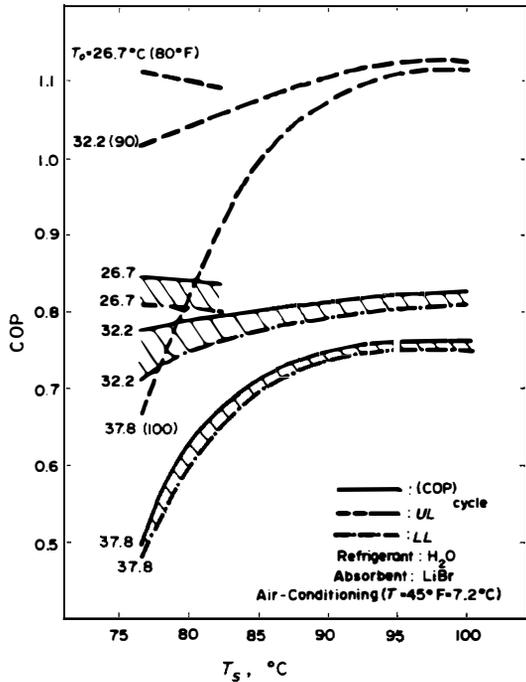


Fig. 7. COP of the  $\text{H}_2\text{O} + \text{LiBr}$  absorption cycle and its upper and lower limits vs solar collector temperature for air-conditioning at different environmental temperatures. The dashed areas are the expected ranges of the actual COP of the cycle.

for refrigeration case ( $T = 36^\circ\text{F}$ ) while Figs. 5-7 are for the case of air-conditioning ( $T = 45^\circ\text{F}$ ). Comparison of Figs. 2-4 indicate that water-LiBr system is a better working fluid combination for refrigeration at higher environmental temperatures, followed by  $\text{NH}_3\text{-NaSCN}$  and  $\text{NH}_3\text{-H}_2\text{O}$  systems. However at lower environmental temperatures  $\text{NH}_3\text{-NaSCN}$  seems to be preferable and that  $\text{H}_2\text{O-LiBr}$  system does not operate at all at lower environmental temperatures. Figures 2-4 also indicate that the actual COP of the cycle seems to be less

sensitive to collector temperature,  $T_s$ , for  $\text{NH}_3\text{-H}_2\text{O}$  system, while it seems to be most sensitive to  $T_s$  for  $\text{H}_2\text{O-LiBr}$  system. The same trends are observed for the case of air-conditioning as it is demonstrated by Figs. 5-7. Overall the COP is higher in air-conditioning than in refrigeration for all the working fluid combinations as it is expected.

On Figs. 8-10 the upper limit of the ratio  $\dot{M}_i/\dot{M}_p$  of the cycle as defined by eqn (13) for different working fluid combinations are reported vs solar collector temperature,  $T_s$ , at different environmental temperatures,  $T_o$ , for the two cases of refrigeration and air-conditioning. According to these figures  $\text{NH}_3\text{-H}_2\text{O}$  system is preferred over the other two systems. Also, comparison of Figs. 8 and

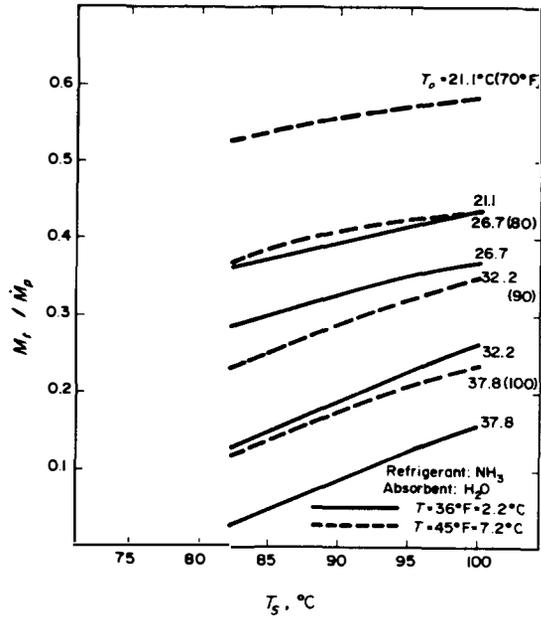


Fig. 9. Upper bounds of  $\dot{M}_i/\dot{M}_p$  of the  $\text{NH}_3 + \text{H}_2\text{O}$  cycle vs solar collector temperature at different environmental temperatures for refrigeration and air-conditioning.

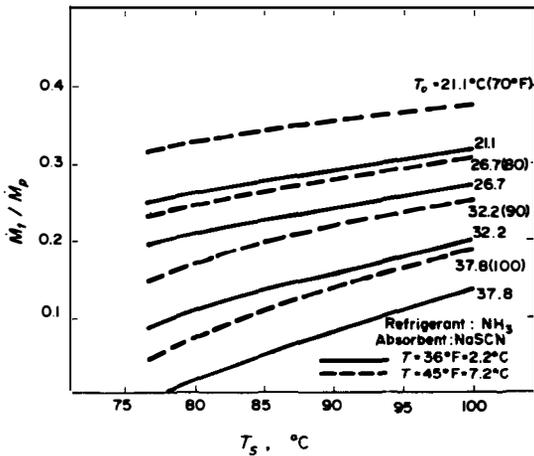


Fig. 8. Upper bounds of  $\dot{M}_i/\dot{M}_p$  of the  $\text{NH}_3 + \text{NaSCN}$  cycle vs solar collector temperature at different environmental temperatures for refrigeration and air-conditioning.

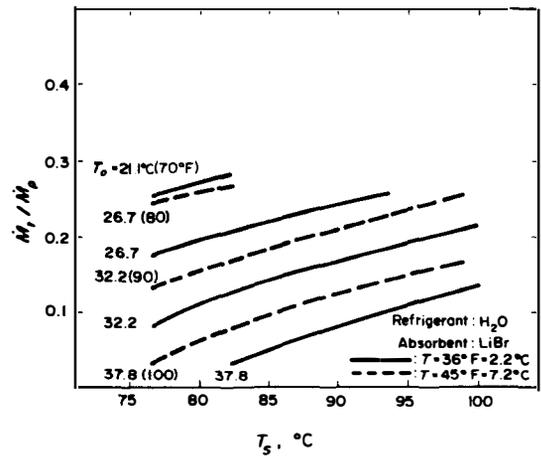


Fig. 10. Upper bounds of  $\dot{M}_i/\dot{M}_p$  of the  $\text{H}_2\text{O} + \text{LiBr}$  cycle vs solar collector temperature at different environmental temperatures for refrigeration and air-conditioning.

10 indicate that the upper limits of  $\dot{M}_i/\dot{M}_p$  values for H<sub>2</sub>O–NaSCN system are slightly better than those for H<sub>2</sub>O–LiBr system especially at lower environmental temperatures.

On Figs. 11–13 the ratio of pump work per ton of cooling,  $\dot{W}_p/\dot{Q}_E$ , for the cycle as given by eqn (29) for different working fluid combinations are reported vs solar collector temperature  $T_s$ , at different environmental temperatures  $T_o$  for the two cases of refrigeration and air-conditioning. According to these figures H<sub>2</sub>O–LiBr combination is preferred over the other two combinations. Also, comparison of Figs. 11 and 12 indicate that  $\dot{W}_p/\dot{Q}_E$  values for NH<sub>3</sub>–NaSCN system are slightly better than for NH<sub>3</sub>–H<sub>2</sub>O system.

From the above comparisons it may be concluded that H<sub>2</sub>O–LiBr combination is preferred for localities with high environmental temperatures. Because of low cost and excellent performance of this working fluid combination it is a favorable candidate for use in solar cooling cycles in hot climates. The NH<sub>3</sub>–H<sub>2</sub>O combination, however, is more versatile due to its applicability in all kinds of environmental temperatures. As a result the NH<sub>3</sub>–NaSCN combination should be categorized in between the other two working fluid combinations.

The method presented in this report can be utilized for comparative study of different refrigerant–absorbent combination which may be chosen. However, for this purpose thermodynamic properties (specifically enthalpy and entropy values) of the refrigerant, absorbent and their liquid mixture should be available.

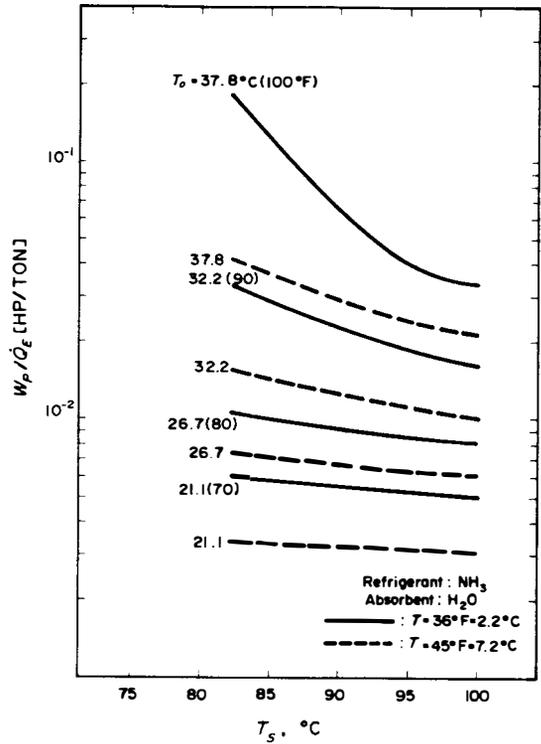


Fig. 12.  $\dot{W}_p/\dot{Q}_E$  of the NH<sub>3</sub> + H<sub>2</sub>O cycle vs solar collector temperature at different environmental temperatures for refrigeration and air-conditioning.

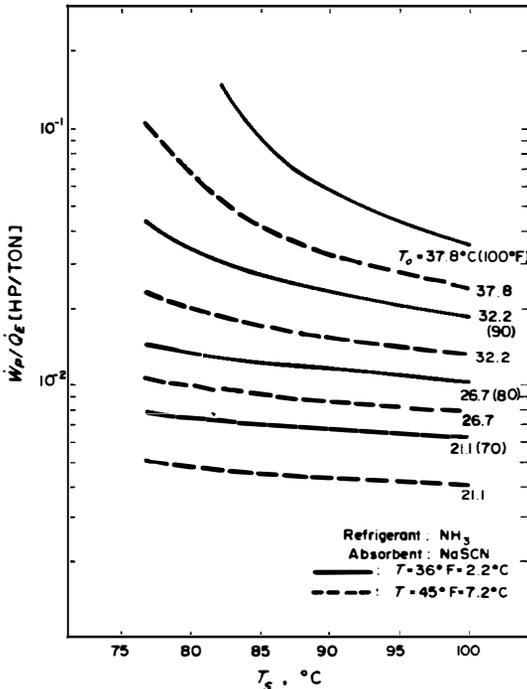


Fig. 11.  $\dot{W}_p/\dot{Q}_E$  of the NH<sub>3</sub> + NaSCN cycle vs solar collector temperature at different environmental temperatures for refrigeration and air-conditioning.

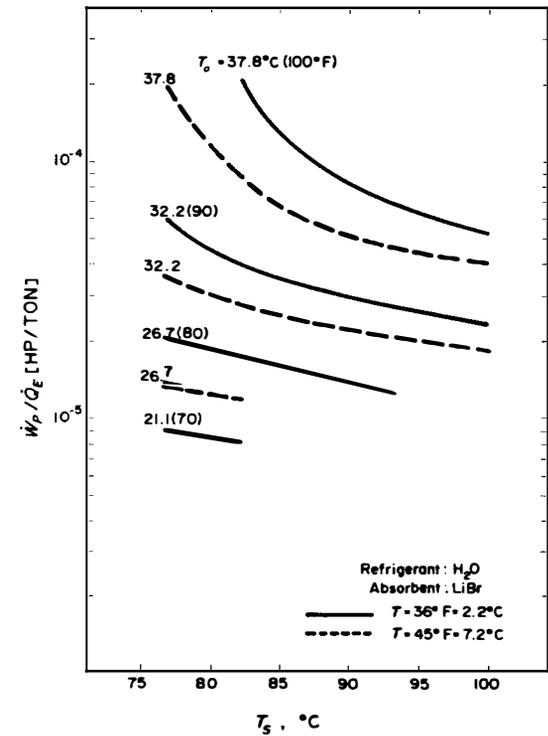


Fig. 13.  $\dot{W}_p/\dot{Q}_E$  of the H<sub>2</sub>O + LiBr cycle vs solar collector temperature at different environmental temperatures for refrigeration and air-conditioning.

*Acknowledgements*—The authors appreciate the support of this research by the U.S. National Science Foundation, grant GK 43139.

#### REFERENCES

1. *ASHRAE Handbook and Product Directory*. Published by ASHRAE Inc., New York (1977).
2. R. W. Allen and D. K. Anand, Parametric study of a dynamic solar powered absorption cycle. *Proc. ISES*, Winnipeg, Vol. 3, p. 27 (1976).
3. D. K. Anand, Solar air conditioning system performance predictions using load, storage and stochastic weather models for different regions, *Paper No. 1048-ISES-ISEC'77 Proc.*, New Delhi, India (1978).
4. P. J. Wilbur and T. R. Mancini, A comparison of solar absorption air conditioning systems. *Solar Energy* 18, 569 (1976).
5. S. L. Grassie and N. R. Sheridan, Modeling of a solar-operated absorption air conditioner system with refrigerant storage. *Solar Energy* 19, 691 (1977).
6. K. G. Denbigh, *Thermodynamics of Steady State*. Wiley, New York (1965).
7. R. K. Swartman *et al.*, Comparison of ammonia-water and ammonia-sodium thiocyanate as the refrigerant-absorbent in a solar refrigeration system. *Solar Energy* 17, 123 (1975).
8. W. F. Stoecker and L. D. Reed, Effect of operating temperatures on the COP of aqua-ammonia refrigerating systems. *ASHRAE Trans.* 77(1), 163 (1971).
9. E. G. Ainbinder *et al.*, Operation of a lithium bromide absorption refrigeration machine. *Otkrytiya, Izobret., Prom. Obratzy, Tovarnye Znaki* 51(45), 164 (1947).
10. E. H. Perry, Theoretical performance of the lithium bromide-water intermittent absorption refrigeration cycle. *Solar Energy* 17, 321 (1975).
11. G. C. Blytas and F. Daniels, Concentrated solutions of NaSCN in liquid ammonia: solubility, density, vapor pressure, viscosity, thermal conductance, heat of solution and heat capacity. *J. Am. Chem. Soc.* 84(7), 1075 (1962).