

SIMULATION OF AN AQUA-AMMONIA INTERMITTENT SOLAR REFRIGERATOR

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ABSTRACT

Performance of an aqua-ammonia absorption solar refrigerator operating intermittently was simulated using a simple thermodynamic model. Input variables were generation and evaporation temperatures, varying from 323 to 393 K and 263 to 298 K, respectively. Calculations were performed for coefficient of performance, solution concentrations, evaporator pressure and refrigerant flow rate for a refrigeration effect of 2880 kJ/day. The results are found to be satisfactory considering the highly non-ideal nature of the system.

Keywords: Solar, refrigerator, aqua-ammonia, simulation, intermittent-cycle.

INTRODUCTION

The use of heat in a refrigeration machine as a source of the energy necessary to transfer heat from a low temperature to a higher one has been in practice for a very long time in absorption refrigeration machines. The machine typically employed ammonia as refrigerant in an ammonia-water, usually called aqua-ammonia system and a hydrocarbon fuel being the source of heat. If solar energy could be harnessed as source of heat to operate an absorption refrigeration cycle it would result in saving in operating cost and would not affect the environment.

The possibility of converting solar energy into motive power, which in turn can be used to drive the compressor in conventional vapour compression refrigeration cycle has been investigated in the past. Conversion efficiencies in case of steam and vapour regeneration being relatively poor and photovoltaic an expensive option, they are not usually economical for

vapour compression refrigeration purposes [1]. The vapour absorption system employs a combination of refrigerant and absorbent satisfying certain desirable properties for better performance of the system. Several combinations such as ammonia-water, water-lithium bromide, ammonia-sodium thiocyanate have been investigated. Ammonia-water is of special interest due to special thermodynamic features like large heat of vaporization due to low molecular weights and high non-ideality. Disadvantages are toxic and flammable nature of ammonia.

The coefficient of performance (COP) of aqua-ammonia solar refrigerator cycles is very low (typically ranging from 0.3 – 0.6 as compared with 3-6 for vapour compression cycle) and even a minor improvement can make a major impact on its cost effectiveness. Many investigators have studied the effect of operating variables on COP in the past. Dixit *et. al.* [2] studied effect of absorber and condenser temperatures on COP. Intermittent nature of the cycle was not taken into consideration and state point thermodynamic property data were from tabulated sources. Venkatesh *et. al.* [3] investigated intermittent cycle using state point property data. The effect of generator temperature and solution concentration on COP was studied. In the present study the emphasis has been on the intermittent nature of the cycle and the use of polynomial equations for equilibrium property data[4]. Intermittent solar refrigeration cycle consists of two distinct sets of operations. The first set, which occurs during the day, consists of 'generation' and 'condensation', while the other set of operations which occurs during the night consists of 'evaporation' and 'absorption'. These two sets of intermittent operations constitute the refrigeration cycle.



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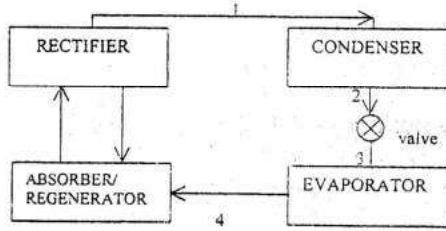


Figure 1: Intermittent Solar Refrigeration Cycle

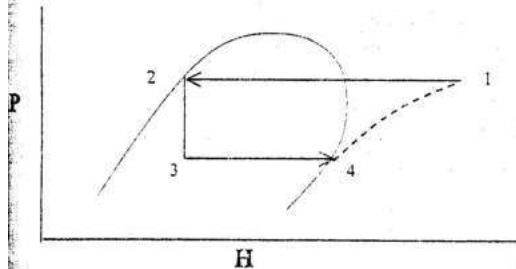


Figure 2: Pressure - Enthalpy Diagram

During 'generation' the refrigerant solution (ammonia in water) in the generator is heated by solar energy and the low boiling refrigerant (ammonia) distills off. This is a batch distillation process. The distillate refrigerant vapour thus generated is condensed in a condenser. For ammonia vapour, water at ambient temperature may be used for condensation. These generation and condensation operations are jointly termed as the 'charging mode' which take place during the day. At the end of this 'charging mode', the ammonia concentration in the regenerator corresponds to the equilibrium concentration corresponding to the generator temperature and pressure. During the night, as the ambient temperature drops considerably, the vapour pressure of ammonia in the generator is lower than the vapour pressure of ammonia in the condenser and there is a tendency for ammonia to evaporate from the condenser unit and be

absorbed in the generator unit. The latent heat of evaporation is supplied by the evaporator chamber thereby cooling the chamber. These operations of evaporation and absorption are jointly termed as the 'cooling mode'. The cooling ends when an equilibrium is established between the evaporator and absorber. At this stage the ammonia concentration in the absorber unit is greater than in the evaporator, and the cycle is complete. Figure 1 shows this cycle. Pressure-enthalpy (P-H) diagram of the cycle is shown in Figure 2.

THERMODYNAMIC MODEL OF THE CYCLE

As shown in Figure 1, solar energy heats the aqua-ammonia solution in the generator.

The vapour from generator is a binary mixture of ammonia and water. This is sent to a rectification unit (Rectifier) where a multistage vapour-liquid contacting lowers the temperature thereby condensing water vapour and distilling off pure ammonia. A weak solution of ammonia-water, residue from rectification, is sent back to the generator. Rectification is assumed to be an adiabatic constant pressure operation.

The following assumptions are made in the analysis:-

1. Steady-state operation
2. Entire system in a state of thermodynamic equilibrium
3. No pressure drop between regenerator, rectifier and condenser; and evaporator and absorber. Hence there would be only two pressure levels, a high pressure (P_{High}) in generator, rectifier and condenser; and a low pressure (P_{Low}) in the evaporator and absorber.
4. Constant enthalpy (Joule Thompson) expansion between state points 2 and 3.
5. Work of pumping and heat loss to the surroundings negligible.

$$P_{High} = P_1 = P_2, \text{ and } P_{Low} = P_3 = P_4$$

Mass and energy balance relationships employed are as following:-

For constant pressure processes, $\Delta H = Q$. Hence total enthalpy change in the evaporator, Q_e^1 -

$$Q_e^1 = Q_e \cdot t_e \quad (1)$$

where Q_e is the heat flow rate and t_e is the time period

$$\text{Also, } Q_e = \Delta H = m (h_4 - h_3) \quad (2)$$

Where m is the refrigerant flow rate and h_3 and h_4 are specific enthalpies at state points 3 and 4.

Overall mass balance around generator gives.

$$M_{ref} = M_1 - M_2 = m \cdot t_e \quad (3)$$

And ammonia balance,

$$M_{ref} = x_s \cdot M_1 - x_w \cdot M_2 \quad (4)$$

Total solar energy supplied to generator, Q_g^1 is given by,

$$Q_g^1 = (M_{ref} \cdot h_{g2} + M_2 \cdot h_{f2}) - M_1 \cdot h_{f1} \quad (5)$$

where M_{ref} and h_{g2} are the mass and enthalpy of ammonia distilled off. M_1 , x_s and h_{f1} are total mass, mass fraction and enthalpy of strong solution before generation. M_2 , x_w and h_{f2} are total mass, mass fraction and enthalpy of the weak solution after generation.

Coefficient of Performance (COP) is given by:

$$\begin{aligned} \text{COP} &= \frac{\text{Refrigeration effect}}{\text{Solar energy input at generator}} \\ &= \frac{Q_e^1}{Q_g^1} \quad (6) \end{aligned}$$

SIMULATION PROCEDURE

Performance of the cycle was simulated by mass and energy balance equations given above at various state points in the cycle. Other thermodynamic equations and correlations used are listed in the Appendix. The input variables were temperatures at the generator, evaporator, condenser and absorber. Rectifier temperature (T_1) was taken to be an average of generator and condenser temperatures [5]. The experimental data employed to test the simulation results were for a refrigeration effect of 2,880 kJ/day, hence Q_e^1 was assigned this value.

The charging mode time (t_c) was assumed to be 6 hours (9 am-3 pm) and cooling mode time to be 8 hours (10 pm - 6 am).

Condenser temperature (T_2) was 303 K and absorber temperature (T_3) was 298 K, being cooling water temperature. Generator temperature (T_5) was varied from 323 to 393 K with an increment of 10°. Evaporator temperature (T_4) was varied from 263 to 298 K with an increment of 50°.

State Point 1: State of the fluid-superheated ammonia vapors at (T_1 , P_1). Here $P_1 = P^{sat}(T_5)$ was evaluated by Reduced Kirchoff equation [6] and enthalpy (h_1) was calculated by using Redlich/Kwong equation of state [7]. Heat capacity of ammonia as function of temperature was taken from International Critical Tables [8].

Reference state was taken to be saturated vapour at 233.15K [9]. Enthalpy of vaporization of ammonia were calculated by using Watson relation [10].

State Point 4: State of fluid-saturated vapour. Saturation vapor pressure at (P_4 , T_4), and enthalpy (h_4) are calculated by Reduced Kirchoff equation and Redlich/Kwong equations as above.

State Point 2: State of fluid-saturated liquid ammonia. Effect of pressure on liquid enthalpy is negligible and hence enthalpy (h_2) was read from thermodynamic property table for saturated liquid ammonia at 303 K (T_2).

State Point 3: State of fluid-partial liquid. For throttle expansion, $h_3 = h_2$.

During Absorption Cycle: Absorber temperature was fixed at 298 K. From thermodynamic tables at 298 K, $P_{\text{NH}_3}^{\text{sat}} = 11.07 \text{ atm}$ and $P_{\text{H}_2\text{O}}^{\text{sat}} = 0.03 \text{ atm}$.

If we assume Raoult's law to be valid, the total pressure, P –

$$P = P_{\text{H}_2\text{O}}^{\text{sat}} + (P_{\text{NH}_3}^{\text{sat}} - P_{\text{H}_2\text{O}}^{\text{sat}}) \cdot x_{\text{NH}_3} \quad (7)$$

And equation (A9) gives $T = t(P, x_{\text{NH}_3})$. These two equations were solved simultaneously to give x_{NH_3} for strong solution. Again equation (A10) gives solution enthalpy,

$h_{f1} = h(T, x_{\text{NH}_3})$. Hence x_s and h_{f1} were thus evaluated.

During Generation: Temperature (T_5) was varied from 323 to 393 K.

$P_{\text{NH}_3}^{\text{sat}}$ and $P_{\text{H}_2\text{O}}^{\text{sat}}$ at T_5 are found by Reduced Kirchoff and Antoine equations [11].

Equation (A7) gives $T = t(P, x_{\text{NH}_3})$. This equation and equation (7) gave x_{NH_3} (x_w). Enthalpy of weak solution was then found by equation (A8).

M_{ref} , M_1 and M_2 were found by equations (1) to (4). Q_g was found by equation (5) and COP by equation (6).

SIMULATION RESULTS AND DISCUSSION

Computations were made for COP, concentrations of strong and weak solutions, pressure, total mass of solution and the refrigerant flow rates as functions of generator and evaporator temperatures. The results are plotted in Figures 3 to 7. Figure 3 is a plot of coefficient of performance (COP) versus rectification temperature for various evaporator temperatures. COP increases sharply as rectifier temperature increases at lower evaporation temperature. At higher evaporator temperatures the COP curve starts to flatten and the increase in COP is not much. Also for a given evaporator temperature, COP becomes almost constant at rectifier temperatures beyond 340 K. Figures 4 and 5 give the effect of temperature on solution

concentration for regeneration cycle (charging mode) and evaporation cycle (cooling mode). In the regeneration cycle the solution concentration drops steadily as rectifier temperature increases. A higher temperature means a higher vapour pressure and a subsequent lower solution concentration. In Figure 4, published experimental data [12] for a work done in Botswana for a refrigerator operating on the same system and capacity, have also been plotted. The experimental solution concentration remains almost constant for temperatures up to about 340 K and then drops. Considering the highly non-ideal nature of the aqua-ammonia system and the simplicity of the present model, the agreement between the predicted and experimental values is satisfactory. Figure 5 gives the solution concentration for evaporation cycle. Concentration increases as the evaporator temperature increases. Figure 6 gives the pressure during the cooling mode as a function of evaporator temperature. Figure 7 gives the refrigerant circulation rate (kg/s) as a function of evaporator temperature. The enthalpy of sat. vapour ammonia increases with temperature and for a given refrigeration effect the refrigerant flow rate must increase with evaporator temperature as given by equations (1) and (2). However, the increase in enthalpy of sat. vapour in the temperature range of 263 – 298 K is only about 2%, which gives almost constant flow rate. The predicted rate at 263 K is 0.095 kg/s and at 298 K is 0.096 kg/s. Hence the plot is almost horizontal.

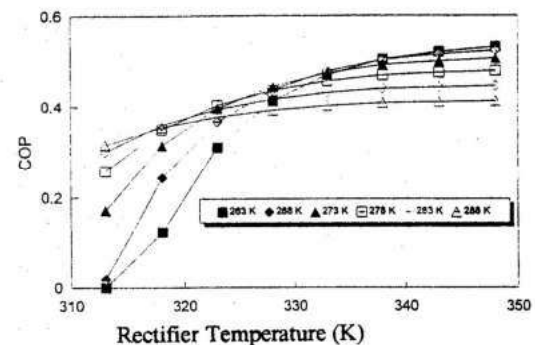


Figure 3: Rectifier Temperature versus COP (for varying evaporator temperature)

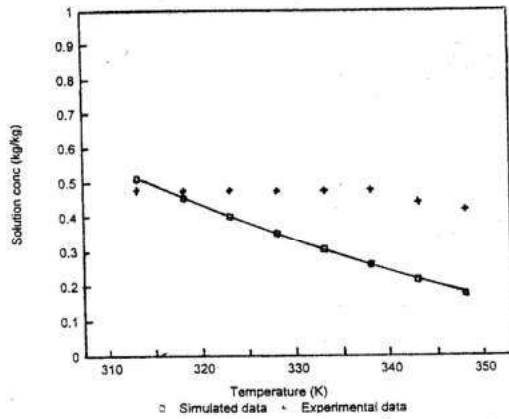


Figure 4: Rectifier Temperature vs. Solution Concentration (Regeneration Cycle)

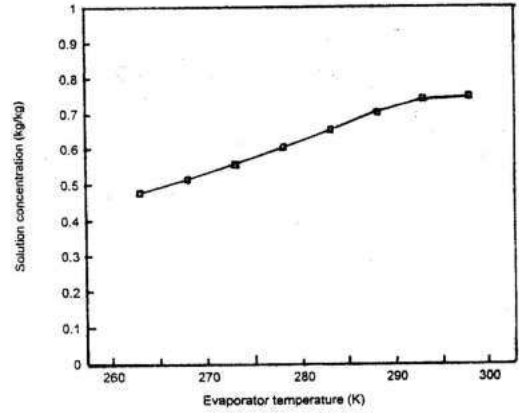


Figure 5: Evaporator Temperature vs. Solution Concentration (Regeneration Cycle)

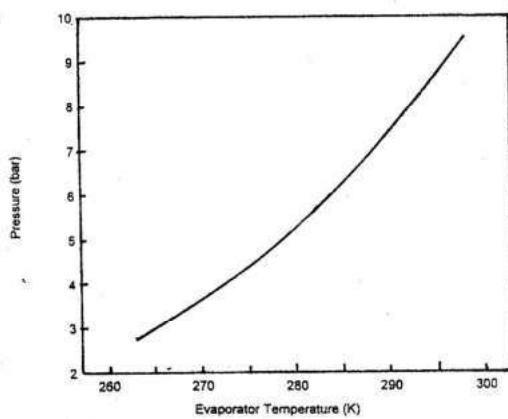


Figure 6: Evaporator Temperature vs. Pressure (Evaporation Cycle)

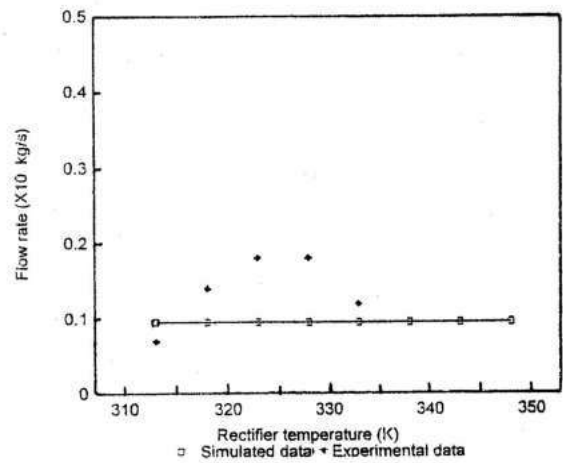


Figure 7: Rectifier Temperature vs. Refrigerant Flow Rate (Regeneration Cycle)

The experimental data points are from the same source mentioned above. The agreement between predicted and experimental data is satisfactory, although not very good due to the limitations of the model. Two of them are:-

- assumption of steady state and equilibrium conditions
- assumption of uniform day and night temperatures (and hence constant pressures)

Since temperature changes during the day and night, a state of thermodynamic equilibrium is never reached. However a better result could be obtained if hourly changes in temperatures are incorporated. Since many variables are in fact controlled by nature, a very good agreement between experimental and theoretical results may be difficult to achieve.

5. CONCLUSIONS

The simulation of an aqua-ammonia intermittent solar refrigerator was performed employing a highly simplistic thermodynamic model and the results have been found to be satisfactory. The comparison with the published experimental data, although very limited, shows that the predicted values compare well.

APPENDIX

Thermodynamic equations and property correlations used in the computations.

Vapour enthalpy

$$\frac{H^R}{RT} = Z - 1 - \left(\frac{3a}{2bRT^{1.5}} \right) \ln(1+h) \quad \dots A1$$

$$H = H_o^{ig} + C_{p,m}^{ig} (T - T_o) + H^R$$

Redlich/Kwong equation of state-

$$Z = \left(\frac{1}{1-h} \right) - \left(\frac{a}{bRT^{1.5}} \right) \left(\frac{h}{1+h} \right)$$

$$h = bP/ZRT \quad \dots A2$$

$$a = \frac{0.42748 R^2 T_c^{2.5}}{P_c}, \quad b = \frac{0.09664 RT_c}{P_c}$$

Vapour Pressure

$$h = T_{br} \left(\frac{\log P_c}{1 - T_{br}} \right) \quad \dots A3$$

$$\log P_r^* = h \left(1 - \frac{1}{T_r} \right)$$

Reduced Kirchoff Vapour Pressure Equation-

Antoine equation

$$\log P^* = A - B/(T + C) \quad \dots A4$$

Specific heat

For Ammonia (0-680°C)

$$C_p = 36.08 + 0.008(T-273.15) + 3.0 \times 10^{-3} (T-273.15)^2 \quad \dots A5$$

Enthalpy of Vaporization

Watson relation-

$$\frac{\Delta H_{v2}}{\Delta H_{v1}} \left(\frac{1 - T_{r2}}{1 - T_{r1}} \right)^n \quad \dots A6$$

$$n = 0.740T_{br} - 0.116$$

Property equations for aqua-ammonia mixtures

For high pressure range ($350 \geq P \geq 250$ psia)

$$T = ((((-240.11 * XL + 346.31) * XL - 27.120) * XL + 166.94) * XL - 535.76) * XL + (0.038849 - 0.18053E-03 * P) * XL * P + 305.04 + (0.44631 - 0.24284E-03 * P) * P \quad \dots A7$$

$$h_r = (((561.86 * XL - 1929.6) * XL + 2343.3) * XL - 828.41) * XL - 103.48) * XL - 76.824 + 1.12703 * T \quad \dots A8$$

For low pressure range ($80 \geq P \geq 50$ psia)

$$T = ((((-692.82 * XL + 1673.3) * XL - 1424.98) * XL + 787.79) * XL - 584.78) * XL + (-0.34428 + 0.00011334 * P * XL + 203.80 + (1.8362 - 0.0060111 * P) * P) \dots A9$$

$$h_f = ((((-656.458 * XL + 1358.93) * XL - 498.318) * XL - 182.534) * XL - 57.1775 + 1.09174 * T) \dots A10$$

NOTATIONS

P = total pressure
 H = enthalpy
 Q = heat
 t = time
 M = total mass
 h = specific enthalpy
 h_f = saturated liq. solution enthalpy
 h_g = saturated vap. enthalpy
 m = refrigerant mass flow rate
 C_p = specific heat
 x, XL = mass fraction of ammonia in solution
 Z = compressibility factor

Superscripts

ig = ideal gas
 r = reduced properties
 t = total
 *,sat = saturated

Subscripts

e = evaporation
 g = generation
 o = reference state
 c = critical parameters
 b = boiling point
 fl = solution before generation
 f2 = solution after generation
 ref = refrigerant
 ḡ = saturated vapour
 s = strong solution
 w = weak solution

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