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The Theory of a Simple Solar Referigerator

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ABSTRACT

This paper gives the basic thermodynamic theory of the ammonia-water liquid-vapour system and applies it to calculating the heat transfers that take place in the intermittent ammonia-water absorption refrigeration cycle. The cycle is adapted to operation with the ammonia-water solution in a flat-plate solar collector which serves both as the generator and as the absorber. The methods of calculating the heat transfer rates to and from the solar collector are treated in detail, and an outline of the method of modelling the system with a varying input of solar energy is given. Some experimental tests that verify the model are quoted.

INTRODUCTION

Broadly speaking, there are two ways of using solar energy to operate refrigerators. One way is to convert solar radiation into electricity by means of photovoltaic solar cells and use the electricity to drive a conventional compression refrigerator. The other way is to convert solar radiation into heat in a thermal solar collector and use the heat to drive an absorption refrigerator.

Photovoltaic converters and compression refrigerators are currently being developed for small-scale domestic use and for the preservation of vaccines in isolated areas. Absorption refrigeration systems with thermal solar collectors enjoy a wider range of possible applications, which include air conditioning, medium-scale cold storage, ice manufacturing, and the same small-scale uses as the photovoltaic systems.

The whole subject of solar powered refrigeration thus embraces a variety of solar and refrigeration technologies, all of which are active areas of research and development at the present time. However, none of the systems developed to date are economically viable because they are too expensive. Efforts are being made to improve efficiencies and bring down costs, but there is still little to choose between the various types, and it is difficult to see which, if any, will ultimately secure appreciable markets.

In order to throw some light on one aspect of this research we shall examine in this article the theory of one of the possible systems, namely an intermittent ammonia-water absorption refrigerator with a flat-plate solar collector. Although this system is not necessarily the best available, the method of analysis could be adapted to other configurations or systems using other working substances.

BASIC CONCEPTS

Fig. 1 shows the principle of operation. Two vessels are connected by an overhead pipe. Initially the left-hand vessel contains an ammonia-water solution, having about 45% of ammonia by weight, at ambient temperature 30°C (303 K). The right hand vessel is empty. First the left hand vessel is heated to about 80°C (353 K) while the right hand vessel is kept at 30°C. This causes the pressure to rise to about 12 atmospheres, and ammonia to boil from the solution in the left-hand vessel and condense in the right-hand vessel. To produce refrigeration the left-hand vessel is now cooled and held at ambient temperature. This reduces the pressure to about 3 atmospheres and causes the liquid ammonia in the right-hand vessel to boil at the refrigeration temperature of about −10°C (263 K). The ammonia vapour is reabsorbed into the solution in the left-hand vessel. When the liquid ammonia has been used up, the right-hand vessel returns to ambient temperature, and the initial state of the system is restored.

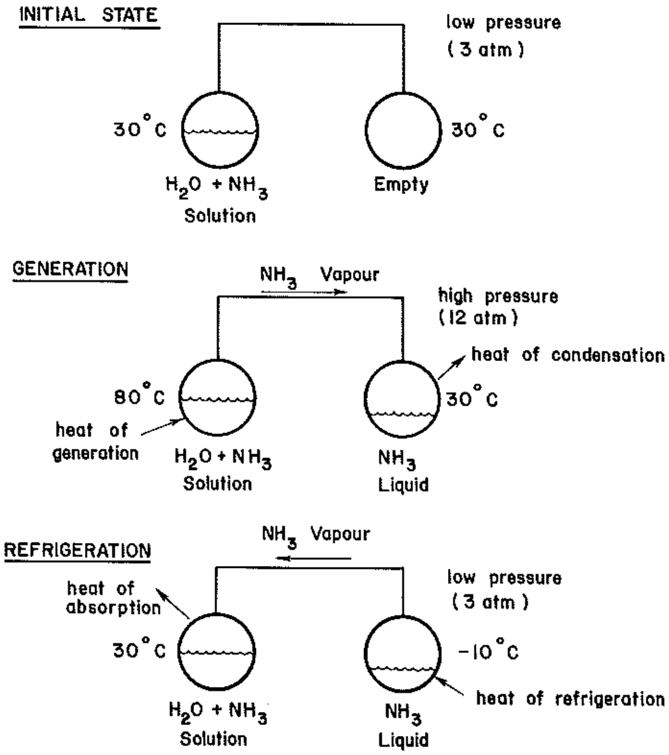

Fig. 1 Principle of ammonia-water absorption refrigeration.

During the generation process, heat is supplied at a high temperature to the left-hand vessel, and the heat of condensation of ammonia is rejected to the environment at ambient temperature from the right-hand vessel. During refrigeration, the right-hand vessel absorbs heat from its refrigeration load at a low temperature, and the heat of reabsorption of the ammonia into the solution in the left-hand vessel is rejected to the environment at ambient temperature.

Let Q_g denote the heat of generation absorbed by the left-hand vessel at the generating temperature T_g; let Q_f denote the heat of refrigeration absorbed by the right-hand vessel at the refrigeration temperature T_f; and let Q_a denote the total heat rejected to the environment at ambient temperature T_a, namely the heat of condensation of the ammonia in the right-hand vessel and the heat of reabsorption of the ammonia in the left-hand vessel.

In designing a refrigeration system we decide on the refrigeration temperature T_f and the heat of refrigeration Q_f it is required to absorb. The value of Q_f determines the amount of solution required. We are then interested in what temperature T_g must be reached in the generating process and how much heat Q_g is required. It is a general rule that the required rise in temperature $(T_g - T_a)$ above ambient temperature during generation is slightly greater than the required fall in temperature $(T_a - T_f)$ below ambient temperature during refrigeration. In the example above the fall in temperature during refrigeration is 40°C, and the required rise in temperature during generation is 50°C.

The amount of refrigeration obtained per unit of heat supplied during the generation process is indicated by the *cooling ratio* defined to be Q_f/Q_g. The maximum possible cooling ratio according to the laws of thermodynamics can be calculated as follows. Since in a complete cycle the system returns to its initial state, the total change in internal energy must be zero; and since there is no work done (no change in volume of the whole system), the only changes in internal energy are those due to the heat flows. Therefore $Q_g + Q_f - Q_a = 0$. Likewise, in a complete cycle the total change in entropy of the system must be zero, so ideally $(Q_g/T_g) + (Q_f/T_f) - (Q_a/T_a) = 0$, where the temperatures must be on the thermodynamic scale. Eliminating Q_a from the above two equations we have $Q_f/Q_g = T_f(T_g - T_a)/T_g\,(T_a - T_f)$. Substituting $T_f = 263$ K, $T_a = 303$ K, and $T_g = 353$ K we obtain $Q_f/Q_g = 0.93$. The cooling ratio in a real system is less than this because (a) some of the heat of generation is transferred at temperatures lower than T_g during the warming up process, (b) some of the heat of refrigeration is transferred at temperatures higher than T_f during the cooling down process, and (c) the temperatures at which heat is transferred to the environment are higher than T_a because of the temperature differences needed for adequate heat transfer rates. Furthermore, in the intermittent cycle, heat is wasted warming up the solution and its container during generation, and cooling the ammonia liquid and its container during refrigeration.

THERMODYNAMIC PROPERTIES OF AMMONIA-WATER SYSTEMS

We shall consider a thermodynamic system having two components, ammonia and water, and a fixed volume that is greater than the volume of the liquid so that two phases, liquid and vapour, are present.

According to the Gibbs phase rule the number of degrees of freedom f, that is to say the number of variables required to specify the thermodynamic state of the system, is given by

$$f = c - \phi + 2,$$

where c is the number of components and ϕ is the number of phases. Since we have $c = 2$ (ammonia and water) and $\phi = 2$ (liquid and vapour), it follows that $f = 2$. The two independent variables that we shall use to specify the thermodynamic state of the system are:

X_L, the mass fraction of ammonia in the liquid, and

T, the temperature.

All other properties of the system are functions of X_L and T, for example:

P,	the pressure,
X_V,	the mass fraction of ammonia in the vapour,
v,	the mass specific volume of the liquid,
h_L, h_V,	the mass specific enthalpies of the liquid and vapour, and
s_L, s_V,	the mass specific entropies of the liquid and vapour.

Some values of these properties over the region of interest are given in the abbreviated Tables 1 to 5. The enthalpies in Table 4, and the entropies in Table 5, are thermodynamic potentials, which means that only differences between them have physical significance. Their individual values cannot be used in isolation. The values quoted are based upon the arbitrary standard values $h_L = 0$ kJ/kg and $s_L = 0$ kJ/kg K for pure water ($X_L = 0$) and pure ammonia ($X_L = 1$) at 0°C. Care is needed when comparing values from these tables with values given in other publications because different authors base their calculations on different standard values. Table 6 shows some thermodynamic properties of the liquid and vapour of pure ammonia in equilibrium.

Table 1
Pressure P of ammonia-water solutions (kPa)

T	X_L:	0.2	0.3	0.4	0.5	0.6
20°C		32	73	147	270	419
40		77	159	302	523	780
60		163	313	561	926	1337
80		310	565	964	1527	2166
100°C		541	954	1558	2400	3429

Table 2
Mass fraction X_V of ammonia in vapour in equilibrium with ammonia-water solutions

T	X_L:	0.2	0.3	0.4	0.5	0.6
20°C		.944	.981	.994	.998	.999
40		.924	.971	.989	.996	.998
60		.900	.957	.982	.992	.996
80		.872	.941	.972	.986	.993
100°C		.842	.921	.959	.978	.988

Table 3
Mass specific volume v of ammonia-water solutions (L/kg)*

T	X_L:	0.2	0.3	0.4	0.5	0.6
20°C		1.08	1.12	1.15	1.21	1.26
40		1.09	1.14	1.18	1.23	1.30
60		1.11	1.16	1.21	1.27	1.35
80		1.14	1.18	1.25	1.31	1.40
100°C		1.15	1.20	1.29	1.37	1.47

*L is used as the unit symbol for litres in this paper.

Table 4
Mass specific enthalpy of ammonia-water solution h_L and vapour in equilibrium with solution h_V (kJ/kg)

T	X_L:	0.2	0.3	0.4	0.5	0.6
20°C	h_L	−65	−125	−160	−163	−145
	h_V	1407	1360	1341	1330	1318
40	h_L	+23	−36	−69	−70	−50
	h_V	1471	1411	1382	1363	1345
60	h_L	112	+57	+26	+27	+49
	h_V	1537	1463	1424	1396	1371
80	h_L	204	155	129	129	152
	h_V	1606	1517	1466	1430	1395
100°C	h_L	300	259	240	241	263
	h_V	1677	1573	1509	1459	1403

Table 5
Mass specific entropy of ammonia-water solution s_L and vapour in equilibrium with solution s_V (kJ/kg K)

T	X_L:	0.2	0.3	0.4	0.5	0.6
20°C	s_L	.23	.14	.07	.07	.09
	s_V	6.31	5.80	5.41	5.07	4.83
40	s_L	.51	.43	.38	.38	.41
	s_V	6.07	5.58	5.20	4.88	4.64
60	s_L	.79	.71	.67	.68	.71
	s_V	5.89	5.41	5.03	4.72	4.48
80	s_L	1.05	.99	.96	.96	1.00
	s_V	5.76	5.28	4.90	4.59	4.34
100°C	s_L	1.31	1.28	1.26	1.27	1.30
	s_V	5.65	5.18	4.80	4.48	4.19

A PRACTICAL SYSTEM

The arrangement shown in Fig. 1 would not work in practice because it has two defects. First, during the refrigeration process ammonia vapour reabsorbed into the solution in the left-hand vessel does so on the surface of the liquid and increases the ammonia concentration X_L there. Because the strong solution is less dense than the weaker solution below the surface, no

Table 6
Properties of saturated ammonia

T (°C)	P (kPa)	v (L/kg)	h_L (kJ/kg)	h_V (kJ/kg)	s_L (kJ/kg K)	s_V (kJ/kg K)
20	190	1.51	91	1239	0.35	4.91
0	429	1.56	0	1263	0	4.63
20	857	1.64	94	1281	0.33	4.38
40	1554	1.73	191	1292	0.49	4.16

mixing takes place and the equilibrium vapour pressure P at the surface increases. This raises the temperature of the ammonia in the right hand vessel, and eventually the process stops.

The second defect is that the vapour passing from the left-hand vessel to the right-hand vessel during generation is not pure ammonia, as desired, but contains 2-3% of water. During refrigeration the presence of this water in the right-hand vessel raises the evaporation temperature above the desired value. Moreover, since no water evaporates during this process more and more of it accumulates in the right-hand vessel as the cycle is repeated, making the system less and less effective.

An arrangement designed to overcome these defects, and to use solar energy from a flat-plate collector to supply the heat of generation, is sketched schematically in Fig. 2. The vessels, pipes and valves are all made of steel; copper and brass are attacked by the ammonia-water solution and cannot be used.

The ammonia-water solution is contained in the solar collector, and a tall air-cooled rectifier pipe serves to remove water from the generated vapour. The vapour is condensed in a condenser

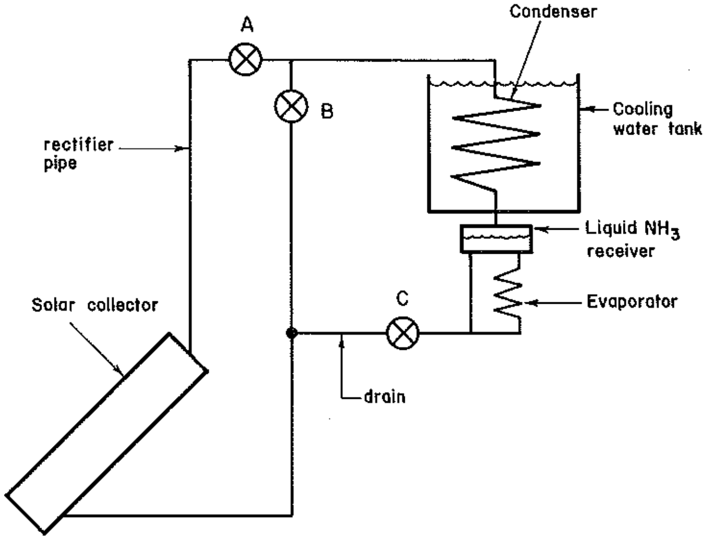

Fig. 2 Scheme for solar refrigerator.

cooled by a tank of water, and the liquid ammonia is collected in the receiver. An evaporator coil provides a heat transfer surface between the liquid ammonia and the refrigeration load, which may be a quantity of water for making ice. There is a drain, closed by a valve **C**, from the evaporator to the solar collector so that if unwanted water accumulates inside the evaporator it can be returned to the solar collector from time to time. During the refrigeration process the ammonia vapour is directed to the bottom of the solar collector to ensure good mixing and uniform reabsorption throughout the solution.

THE IDEAL CYCLE

The ideal cycle for operating the system shown in Fig. 2 consists of four processes, which are illustrated on the thermodynamic diagram in Fig. 3. Starting with the solution in the solar collector at ambient temperature and the receiver empty, the four processes are as follows:

(1) **a-b**: Heating of the solution in the collector with valves **A** and **B** closed, during which the concentration X_L remains constant and the pressure P increases.

(2) **b-c**: Heating of the solution in the collector with valve **A** open and **B** closed. During this process, which takes place at constant pressure, ammonia evaporates from the solution reducing its concentration, condenses at ambient temperature (point **e**), and accumulates in the receiver.

(3) **c-d**: Cooling of the solution in the collector with valves **A** and **B** closed, during which the concentration X_L remains constant and the pressure P decreases.

(4) **d-a**: Evaporation of ammonia from the evaporator (point **f**) with valve **A** closed and **B** open. During this process, which takes place at constant pressure, the ammonia vapour is reabsorbed into the solution, increasing its concentration.

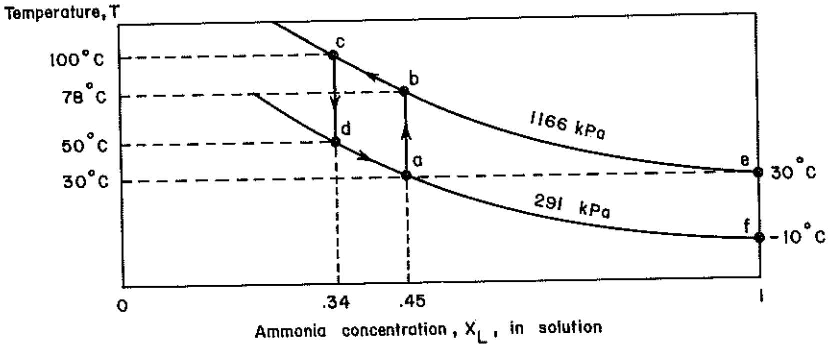

Fig. 3 Ideal thermodynamic cycle.

The heating processes (1) and (2) are carried out during the daytime under the sun. The cooling processes (3) and (4) are carried out at night with the back insulation removed from the solar collector to allow the heat to escape.

The method of determining the points on the cycle will now be explained. We shall suppose that the ambient temperature is 30°C and that the required refrigeration temperature is −10°C. The point f is fixed by the refrigeration temperature −10°C and the concentration X_L = 1. Here the vapour pressure of pure ammonia is 291 kPa. The point **a** is the intersection of the 291 kPa isobar and the 30°C isotherm. This gives the ammonia concentration $X_L = 0.45$ of the solution that must be charged into the system at the outset. The point **e** is fixed by the condensing temperature 30°C and the concentration $X_L = 1$. Here the vapour pressure of pure ammonia is 1166 kPa. The point **b** is now the intersection of the 1166 kPa isobar and the line $X_L = 0.45$, which occurs at temperature $T = 78°$C. The position of the point **c** on the 1166 kPa isobar depends on the amount of solar heating that occurs on a particular day, and on the amount of solution the solar collector contains per unit area of collecting surface. In a typical design on a sunny day a temperature of 100°C might be reached. This occurs at a concentration $X_L = 0.34$. Finally, the point **d** is the intersection of the line $X_L = 0.34$ and the 291 kPa isobar, which occurs at a temperature $T = 50°$C. The whole cycle is now established.

It is convenient to have expressions for the mass of liquid m_L in the solar collector, and the mass of ammonia vapour m_V that has been distilled from the liquid, in terms of the initial mass $m_L(a)$ and concentration $X_L(a)$ of the liquid at point **a** in the cycle and the liquid concentration X_L at any other point. A complete theory must take into account the small amount of water in the vapour. However, to simplify our calculations we shall assume that the vapour is pure ammonia. Then we have

$$m_L = m_L(a)\,(1 - X_L(a))/(1 - X_L), \tag{1}$$

and

$$m_V = m_L(a)\,(X_L(a) - X_L)/(1 - X_L). \tag{2}$$

HEAT TRANSFERS IN THE CYCLE

Let m_{sc} denote the mass of steel in the pipework and the black plate of the solar collector, and let C_s denote the specific heat capacity of steel. Then the differential of the heat entering the steel in any temperature change is $m_{sc}C_s\mathrm{d}T$.

We first calculate the heat of generation in the process **a-b**. Since this occurs at constant volume with no production of ammonia vapour, the differential of the heat supplied to the liquid is $m_L(a)\,T\,\mathrm{d}s_L$, where T must be expressed in kelvins. Therefore, the total heat supplied in the process **a-b** is

$$Q_{ab} = \int_a^b (m_{sc}\,C_s\,\mathrm{d}T + m_L\,(a)\,T\,\mathrm{d}s_L).$$

For our calculations the integral may be approximated by

$$Q_{ab} = m_{sc}\,C_s\,(T(b) - T(a)) + m_L(a)\tfrac{1}{2}(T(a) + T(b))\,(s_L(b) - s_L\,(a)). \tag{3}$$

The values of m_{sc} and $m_L(a)$ depend on the design of the system. In accordance with the designs tested in the Asian Institute of Technology we shall take $m_{sc} = 65$ kg and $m_L(a) = 12$ kg per square metre of collector area, and quote all the results below on this basis. We also have $C_s = 0.46$ kJ/kg K.

Thermodynamic tables (more detailed than those in this paper) give for $T(a) = 303$ K and $T(b) = 351$ K, the values $s_L(a) = 0.22$ kJ/kg K and $s_L(b) = 0.93$ kJ/kg K. Substituting in (3) we obtain

$$Q_{ab} = 4221 \text{ kJ}.$$

The process **b-c** takes place at constant pressure with valve **A** open, so the differential of the heat supplied to the ammonia-water system in the solar collector is the differential of the enthalpy of the liquid-vapour system with the term for the sensible heating of the vapour omitted. The total differential of the enthalpy of the liquid-vapour system is

$$\mathrm{d}\,(m_L h_L + m_V h_V) = \mathrm{d}(m_L h_L) + h_V \mathrm{d}m_V + m_V \mathrm{d}h_V.$$

Adding the term $m_{sc} C_s \mathrm{d}T$ for the heating of the steel pipes, omitting the term $m_V \mathrm{d}h_V$ for the heating of the vapour, and integrating from b to c we obtain

$$Q_{bc} = \int_b^c (m_{sc}\, C_s\, \mathrm{d}T + \mathrm{d}\,(m_L h_L) + h_V \mathrm{d}m_V),$$

which may be approximated by

$$Q_{bc} = m_{sc} C_s\, (T(c) - T(b)) + m_L(c)\, h_L(c) - m_L(b)\, h_L(b)$$
$$+ \tfrac{1}{2}\, (h_V(b) + h_V(c))\, (m_V(c) - m_V(b)). \qquad (4)$$

Here we have $T(c) = 373$ K, and from thermodynamic tables $h_L(\mathrm{b}) = 115$ kJ/kg, $h_V(\mathrm{b}) = 1444$ kJ/kg, $h_L(c) = 249$ kJ/kg, and $h_V(c) = 1544$ kJ/kg. Equations (1) and (2) with $m_L(a) = 12$ kg and $X_L = 0.34$ give $m_L(c) = 10$ kg and $m_V(c) = 2$ kg. Substituting in (4) we obtain

$$Q_{bc} = 4756 \text{ kJ}.$$

Summarizing, we find the total heat of generation is $Q_{ab} + Q_{bc} = 8977$ kJ, and this produces 2 kg of distilled ammonia per square metre of solar collector area.

By similar calculations we may find the amount of heat that must be released by the solar collector in the processes **c-d** and **d-a**. Thus, replacing a by c and b by d in equation (3) leads to

$$Q_{cd} = -4001 \text{ kJ},$$

and replacing b by d and c by a in equation (4) leads to

$$Q_{da} = -4743 \text{ kJ},$$

so the total amount of heat to be released is $Q_{cd} + Q_{da} = -8744$ kJ, the negative sign indicating that the heat is lost by the solution and the solar collector.

In designing the condenser (see Fig. 2) we must know how much heat is released to the tank of water by the condensing ammonia. Since the process takes place at constant pressure, this is simply the difference between the enthalpy of the ammonia vapour entering the condenser $m_A h_{AV}$ and the enthalpy of the ammonia liquid leaving the condenser $m_A h_{AL}$. Here we have $m_A = 2$ kg, and making the simplifying assumption that the ammonia is pure and that the whole process takes place at 30°C, we have from thermodynamic tables of pure ammonia $h_{AV} = 1288$ kJ/kg and $h_{AL} = 142$ kJ/kg. The total amount of heat released by 2 kg of ammonia condensing at

30°C is therefore about 2292 kJ per square metre of solar collector area. In designing the system one must ensure that there is sufficient water in the tank to prevent its temperature from being raised more than a few degrees by this heat.

We shall suppose that at the point **d** in the cycle, when the valve **B** is opened, the liquid ammonia and the receiver containing it fall rapidly to the temperature $-10°C$, which is represented by the transition from point **e** to point **f** in Fig. 3. This is a simplification of what actually occurs because the release of the heat of absorption from the solar collector takes time.

Let m_{sr} denote the mass of steel in the receiver and evaporator; then the differential of the heat flow from the receiver to the liquid ammonia is $m_{sr}C_s\,dT$. The differential of the heat flow into the ammonia is

$$dH_A - V_A dP = d(m_{AL}h_{AL} + m_{AV}h_{AV}) - (m_{AL}v_{AL} - m_{AV}v_{AV})dP,$$

where the subscripts A, L and V refer to ammonia, liquid and vapour respectively. Setting $dh_{AV} = -dm_{AL}$, omitting the terms containing dh_{AV} and v_{AV}, equating the result to the differential of the heat from the receiver, and integrating from state **e** to state **f**, we obtain

$$\int_e^f (m_{AL}dh_{AL} + (h_{AL} - h_{AV})dm_{AL} - m_{AL}v_{AL}dP) = \int_e^f m_{sr}C_s dT.$$

Finally, approximating the integrals in the same way as before we obtain

$$\begin{aligned}
&\tfrac{1}{2}(m_{AL}(e) + m_{AL}(f))(h_{AL}(f) - h_{AL}(e)) \\
&+ \tfrac{1}{2}(h_{AL}(e) - h_{AV}(e) + h_{AL}(f) - h_{AV}(f))(m_{AL}(f) - m_{AL}(e)) \\
&- \tfrac{1}{2}(m_{AL}(e) + m_{AL}(f))\,\tfrac{1}{2}(v_{AL}(e) + v_{AL}(f))(P(f) - P(e)) \\
&= m_{sr}C_s(T(f) - T(e)).
\end{aligned}$$

All the quantities in this equation are known from the system design and from thermodynamic tables except $m_{AL}(f)$, which may therefore be calculated. Taking $m_{sr} = 10$ kg and the remaining quantities from thermodynamic tables we obtain $m_{AL}(f) = 1.576$ kg per square metre of collector.

The useful heat of refrigeration Q_f is the heat required to vaporize the remaining liquid ammonia at constant temperature $-10°C$ and pressure 291 kPa, which is given by

$$Q_f = m_{AL}(f)(h_{AV}(f) - h_{AL}(f)).$$

Substituting the known values on the right hand side we obtain $Q_f = 2046$ kJ per square metre of collector. The cooling ratio for this cycle is defined to be $Q_f/(Q_{ab} + Q_{bc})$. Using the results calculated above we obtain a cooling ratio of 0.228. This is much less than the ideal ratio 0.93 mentioned earlier because heat is transferred to and from the working fluids at temperatures in between the principal operating temperatures, and because heat is needed to raise and lower the temperature of the steel pipework.

HEAT TRANSFER RATES IN THE SOLAR COLLECTOR

The rates at which the processes in the cycle take place are controlled by the energy flow rates to and from the solar collector. In discussing these rates we shall suppose that the solar flat-plate collector has one glass cover at a distance 50 mm from a selective black plate, and there is polyurethane foam insulation 100 mm thick at the back which can be removed at night to allow the system to cool quickly.

The transmissivity $\tau(\theta)$ of the glass cover, and the absorptivity $\alpha(\theta)$ of the black plate, for solar radiation both depend on the angle of incidence θ of the radiation. Typical values assumed for this calculation are given in Table 7.

Table 7
Transmissivity of the glass cover and absorptivity of the black plate for solar radiation

θ:	0°	60	70	80	90°
$\tau(\theta)$:	0.90	0.80	0.68	0.44	0
$\alpha(\theta)$:	0.92	0.85	0.74	0.50	0
$\tau(\theta)\,\alpha(\theta)$:	0.83	0.68	0.50	0.22	0

The total solar irradiance q_{in} incident on the glass cover is given by

$$q_{in} = I\cos\theta + D,$$

where I is the direct solar irradiance at angle θ, and D is the diffuse solar irradiance. The rate at which solar radiation is absorbed by the black plate per unit area is therefore

$$q_{abs} = \tau(\theta)\alpha(\theta)\, I\cos\theta + \overline{\tau\alpha}\, D, \tag{5}$$

where $\overline{\tau\alpha}$ is the mean value of $\tau(\theta)\alpha(\theta)$. The quantity $\overline{\tau\alpha}$ can be found for a collector that is nearly horizontal, as in the tropics, by means of integrals over the hemispherical sky thus:

$$\overline{\tau\alpha} = \left(\int_{\theta=0}^{\pi/2} \tau(\theta)\,\alpha(\theta)\sin\theta\cos\theta\, d\theta\right) \Big/ \left(\int_{\theta=0}^{\pi/2} \sin\theta\cos\theta\, d\theta\right).$$

Using the values of $\tau(\theta)\,\alpha(\theta)$ given in Table 7, we obtain $\overline{\tau\alpha} = 0.70$.

The annual mean values of q_{abs} calculated by the above methods for the climatic conditions in Thailand and various states of cloudiness are shown in Table 8. The parameter k is defined to be the ratio of the daily total global solar radiation to the corresponding value under a clear sky. The dependence of q_{abs} on k is not simple because the ratio of direct to diffuse solar radiation, and hence the effect of the angle of incidence θ, changes as k changes.

There are several processes by which a flat-plate solar collector loses heat, as illustrated in

Table 8
Mean values of radiation absorbed by a flate plate with one glass cover (W/m^2)

Time of day		k=1	k=0.75	k=0.5	k=0.25
6h	18h	6	6	5	3
7h	17h	101	88	64	34
8h	16h	307	232	156	78
9h	15h	524	374	242	119
10h	14h	699	487	310	151
11h	13h	809	558	354	172
12h		846	582	368	179

Fig. 4. They are as follows:

(1) Convection and radiation from the glass cover at temperature T_c to the surroundings at ambient temperature T_a, are given by

$$q_{up} = h_{ca}\,(T_c - T_a) + \epsilon_c \sigma T_c^{\,4} - \epsilon_c L, \tag{6}$$

where h_{ca} is the heat transfer coefficient for convection from the glass cover to the surrounding air, ϵ_c is the long wave emissivity of the glass cover, σ is the Stefan-Boltzmann constant 56.7 x 10^{-9} W/m^{2}K^4, and L is the downward longwave radiation from the sky.

The coefficient h_{ca} is difficult to estimate since it is partly due to free convection and partly due to forced convection by the wind. Watmuff, Charters and Procter recommend

$$h_{ca} = 2.8 + 3.0\,V \text{ W/m}^2\text{K},$$

where V is the wind speed in metres per second. For example, with a wind speed 3 m/s we have $h_{ca} = 11.8$ W/m^{2}K.

The glass cover behaves almost as a black-body for longwave radiation (at wavelengths around 10 μm); we shall take $\epsilon_c = 0.95$.

The downward longwave radiation L depends on the temperature and humidity of the air,

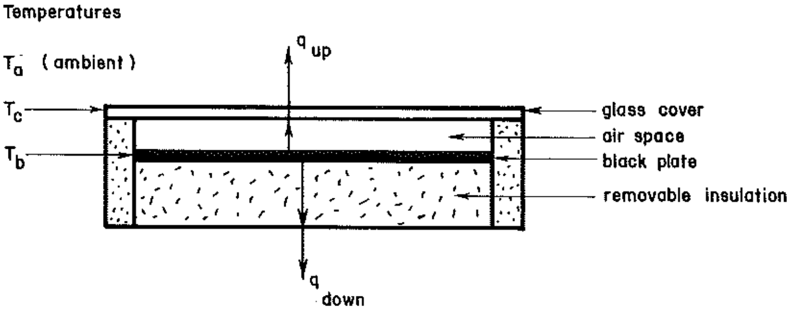

Fig. 4 Temperatures and heat losses in solar collector.

and on the presence of clouds in the sky. With ambient temperature T_a = 303 K (30°C) a typical value might be L = 420 W/m^2.

(2) Convection and radiation between the black plate at temperature T_b and the glass cover are given by

$$q_{up} = h_{bc}\,(T_b - T_c) + \epsilon_{bc}\,\sigma\,(T_b{}^4 - T_c{}^4), \tag{7}$$

where h_{bc} is the coefficient for free convection across the air space, and

$$\epsilon_{bc} = 1/(1/\epsilon_b + 1/\epsilon_c - 1),$$

where ϵ_b is the longwave emissivity of the black plate.

The coefficient h_{bc} is estimated from experimentally determined relations between the Nusselt number Nu and the Grashof number Gr for free convection between parallel planes. For $10^4 < Gr < 10^7$ Tabor recommends

$Nu = 0.152\,Gr^{0.281}$ for horizontal planes, and

$Nu = 0.093\,Gr^{0.310}$ for tilt angle 45°

The Grashof number is defined by

$Gr = g\,\beta\,(T_b - T_c)x^3/\nu^2$, where

g = acceleration of gravity = 9.8 m/s^2,

β = volumetric thermal expansivity = $1/T$,

x = plate spacing,

ν = kinematic viscosity of air.

As a typical example in our case with T_b = 80°C, T_c = 40°C, T = 60°C = 333 K, x = 50 mm = 0.05 m, $\nu = 0.194 \times 10^{-4}$ m^2/s we obtain $Gr = 3.91 \times 10^5$, which is within the range mentioned above. Now, assuming a tilt angle of 15°, we estimate by interpolation from Tabor's equations

$$Nu = 0.132\,Gr^{0.2907} = 5.57.$$

Since $Nu = h_{bc}x/k$, where k is the thermal conductivity of air 0.0275 W/m K, we have h_{bc} = 3.07 W/m^2 K.

Assuming the black plate to be selective we take its emissivity to be ϵ_b = 0.1. Then we find, from the equation above for the radiation exchange between the black plate and the glass cover, ϵ_{bc} = 0.0995.

(3) Conduction through the back insulation is

$$q_{down} = (k/x)\,(T_b - T_a), \tag{8}$$

where k is the thermal conductivity of the insulating material, and x is its thickness. We have in our example k = 0.025 W/m K for polyurethane foam, and x = 0.1 m. Equation (8) assumes that the thermal resistance between the outside surface of the back insulation and the surroundings is small compared with the resistance of the insulation itself.

(4) Heat losses through the edges of the collector and through the connecting pipes. These losses may not be neglegible, but in good designs they should be small and we shall not consider them here.

(5) Convection and radiation from the back of the black plate when the insulation is removed at night to cool the collector and release the heat of absorption. This is given by

$$q_{down} = h_{ba} (T_b - T_a), \tag{9}$$

where h_{ba} is an overall heat transfer coefficient. It is difficult to estimate h_{ba} theoretically. Experiments show that a value h_{ba} = 7.0 W/m^{2}K is appropriate. This is consistent with free convection from a heated plate facing downwards with a slight tilt and a breeze less than 1 m/s, plus the effects of radiation.

The total heat loss rate q_{loss} from the collector (when the back insulation is in position) is given by

$$q_{loss} = q_{up} + q_{down} \tag{10}$$

as a function of T_b with the help of equations (6), (7) and (8). To find this function first assume a value for q_{up} and solve equation (6) for T_c. Next solve equation (7) for T_b. Finally, q_{down} is found from equation (8), and q_{loss} is found from equation (10). Then repeat the calculation for different values of q_{up} to obtain q_{loss} as a function of T_b. The calculations must be done numerically.

Some results obtained for ambient temperature 30°C using this method give for a non-selective black plate and one glass cover:

$$q_{loss} = 12 + 6.13\,(T_b - 30) + 0.0222\,(T_b - 30)^2 \text{ W/m}^2, \tag{11}$$

and for a selective black plate and one glass cover:

$$q_{loss} = 3 + 3.4\,(T_b - 30) + 0.0064\,(T_b - 30)^2 \text{ W/m}^2, \tag{12}$$

where T_b is in degrees Celsius.

When the insulation is removed from the back of the solar collector equations (11) and (12) for the heat loss have to be modified by adding the term from equation (9). This gives for a non-selective black plate

$$q_{loss} = 12 + 13.13\,(T_b - 30) + 0.0222\,(T_b - 30)^2 \text{ W/m}^2, \tag{13}$$

and for a selective black plate

$$q_{loss} = 3 + 10.4\,(T_b - 30) + 0.0064\,(T_b - 30)^2 \text{ W/m}^2, \tag{14}$$

where T_b is in degrees Celsius.

The instantaneous efficiency η of the solar collector is defined by the equation

$$\eta = (q_{abs} - q_{loss})/q_{in}.$$

This depends on the intensities of direct and diffuse solar irradiance, I and D, on the angle of incidence θ, and on the temperature T_b of the black plate. Consider as an example $q_{in} = 800$ W/m^2 with $I = 650$ W/m^2, $D = 150$ W/m^2, $\theta = 0$ at midday, and the black plate at 80°C such as one might find during generation on a fair day. Then taking $\tau(0)\alpha|(0) = 0.83$ from Table 7, and $\overline{\tau\alpha} = 0.70$ we obtain from equation (5) $q_{abs} = 644.5$ W/m^2. For a non-selective black plate we obtain from equation (11) $q_{loss} = 374$ W/m^2, and hence $\eta = 0.34$. For a selective black plate we obtain from equation (12) $q_{loss} = 189$ W/m^2, and hence $\eta = 0.57$.

The overall performance of the system may be defined to be the ratio of the useful heat of refrigeration Q_f to the total solar radiation incident on the collector per day. This equals the cooling ratio multiplied by the average efficiency of the solar collector throughout the day, which may be about 0.4. Taking the cooling ratio 0.288 found earlier we obtain an overall performance of about 0.09. This is close to the overall performance of a large number of solar energy conversion devices that are limited by the second law of thermodynamics.

MODELLING OF THE SYSTEM THROUGH TIME

Although we have seen how to calculate the total quantities of heat in a particular cycle, and the heat transfer rates in the solar collector that control the processes in the cycle, we still require a method of calculating the response of the system to a given solar radiation input throughout the day. In order to do this we must write down the differential equations for the heat transfers that occur in each of the processes **a-b**, **b-c**, **c-d** and **d-a**, and integrate them with respect to time. The calculations involved are simple in principle, but detailed and numerous. They must therefore be performed on an automatic computer. Full details cannot be given in an article of this length; however, the basic ideas can be explained briefly.

In the process **a-b** the differential of the absorbed solar radiation minus the losses from the collector equals the differential of the heating of the collector and solution:

$$(q_{abs} - q_{loss})\mathrm{d}t = m_{sc}C_s\mathrm{d}T + m_L T\,\mathrm{d}s_L. \tag{15}$$

Here q_{abs} is given by equation (5) with I, D and θ specified as functions of time; while q_{loss} is given by equation (11) or (12). For simplicity we assume that the temperature T of the solution and the black plate temperature T_b are the same. In real designs these temperatures differ slightly, and the temperature of the solution is not uniform, but we shall ignore these difficulties. The thermodynamic properties of the solution are assumed to be known as functions of X_L and T. The right-hand side of (15) is then written

$$m_{sc}C_s\mathrm{d}T + m_L(X_L, T)\,T\,(\partial s_L(X_L, T)/\partial T)\mathrm{d}T,$$

and the resulting equation when integrated numerically step-by-step gives T as a function of time t. At each step the pressure $P(X_L, T)$ is found and the integration is continued until the condensing pressure 1166 kPa is reached. On a cloudy day the insolation may not be strong enough for this to occur, and then no ammonia is distilled.

In the process **b-c** the differential equation is

$$(q_{abs} - q_{loss})\mathrm{d}t = m_{sc}C_s\mathrm{d}T + \mathrm{d}(m_L h_L) + h_V\mathrm{d}m_V, \tag{16}$$

where m_L, h_L, m_V and h_V are known functions of X_L and T. It follows that the right-hand side can be written in terms of the differentials dX_L and dT. In order to follow the isobar we need $dP(X_L, T) = 0$; so the differentials dX_L and dT must satisfy the equation

$$(\partial P(X_L, T)/\partial X_L)dX_L + (\partial P(X_L, T)/\partial T)dT = 0. \tag{17}$$

Equations (16) and (17) are now solved simultaneously step-by-step to determine how far the process goes. On a good day the final temperature may be high and a large quantity of ammonia may be distilled. It is assumed that when the temperature stops rising, due to the reduction of the solar heating towards the end of the day, the valve **A** (Fig. 2) is closed and the insulation is removed from the back of the collector to start the process **c-d**.

At this point we have calculated the amount of ammonia distilled and available for refrigeration. For the calculation of the process **c-d** one uses equation (15) again with q_{abs} small at the end of the afternoon and zero after dark, and q_{loss} given by equation (13) or (14). The process **d-a** is controlled partly by what happens in the evaporator and how it is coupled to the refrigeration load. Further development of the model is needed to calculate this process.

The above method of modelling the system has been found to give results that compare fairly well with experimental tests in the Asian Institute of Technology. Table 9 gives some of the results obtained using 4 min time steps in the integrations and assuming uniform states of cloudiness occur throughout the day. The relationship between the amount of ammonia distilled and the total daily global solar radiation with uniformly cloudy skies is approximately linear. Fig. 5 shows the results of calculations for a non-selective black plate, and some results from the experimental tests. The variability of the experimental yields is caused by the manner in which the cloudiness varies throughout the day. If the weather is fine in the morning and cloudy in the afternoon, the yield of ammonia may be good due to the strong heating of the solar collector in the morning. On the other hand, when the morning is cloudy the collector will not become sufficiently hot by midday to generate ammonia, even if the weather is fine in the afternoon making the total solar radiation for the day large. Many of the observed yields shown in Fig. 5 are significantly higher than the calculation gives for uniform cloudiness throughout the day.

Table 9
Ammonia distilled per square metre of collector. H = daily global solar radiation.

Cloudiness, k: H(MJ/m^2):	1 26.5	0.75 19.8	0.5 13.2	0.25 6.6
Non-selective plate	2.16 kg	0.71 kg	0	0
Selective plate	3.36 kg	2.04 kg	0.49 kg	0

PRACTICAL TESTS

The practical experiments on ammonia-water absorption refrigerators powered by solar energy in the Asian Institute of Technology over the past few years have been aimed at developing an ice-maker suitable for small communities in remote areas. The first unit designed had a flat-

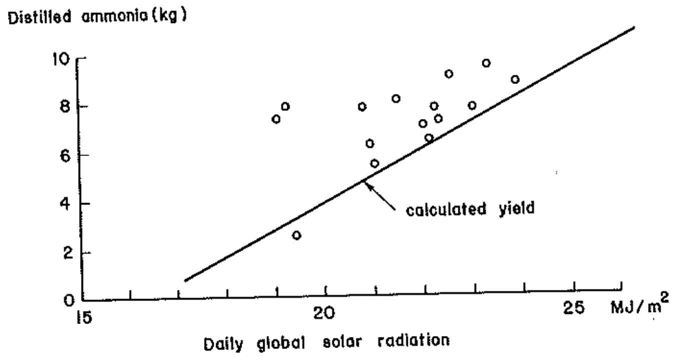

Fig. 5 Test results (small circles) and calculated yield.

plate collector of area 1.6 m^2. Next, a system with two collectors each 2.5 m^2 in area was made. This produced 20-25 kg of ice on a fine day, and was estimated to be capable of producing over 6000 kg of ice per year.

Both of these systems were hand operated, and would require the services of a skilled attendant. The hand valves were also liable to fail after a time. The need for operating valves was therefore eliminated by the use of a special liquid seal designed to control the vapour flows to and from the collector during generation and refrigeration in the manner described earlier.

The ultimate objective of the work is to develop a village-size refrigerator that can provide 0.5 m^3 of food storage space below 10°C, or make about 100 kg of ice per day under favorable conditions. Such a unit has recently been constructed on the AIT campus. It has twelve solar panels, each 2 m^2 in area, containing the ammonia-water solution. A liquid seal controls the flow of ammonia vapour, and a special device has been designed to distribute the vapour equally to the solar panels for reabsorption during refrigeration at night. The system has already been shown to be capable of making 60 kg of ice on a moderately good day.

BIBLIOGRAPHY

EXELL, R.H.B. & SOMMAI KORNSAKOO (1981), *Design and Testing of a Solar Powered Refrigerator*, AIT Research Report No. 126.

SCATCHARD, G., L.F. EPSTEIN, J. WARBURTON Jr. & P.J. CODY (1947), Thermodynamic Properties of Saturated Liquid and Vapour Ammonia-Water Mixtures, *Refrigeration Engineering*, May 1947, pp. 413-419, 446-452.

DUFFIE J.A. & W.A. BECKMAN (1974), *Solar Energy Thermal Processes*, John Wiley and Son Ltd. (1980), *Solar Engineering of Thermal Processes*, John Wiley and Son Ltd.

EXELL, R.H.B., CHEN YEA-GONG & S. KORNSAKOO (1983), Mathematical Simulation of a Solar Absorption Refrigerator, Solar World Congress, Perth 1983, International Solar Energy Society.