

Performance of a Solar LiBr - Water Absorption Refrigerating Systems

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Abstract - Up to now, the production of the cold is undertaken mainly by the classical way of compression refrigerator cycle. The cycle need an electrical power or mechanics expense for the training of the compressor and the C.F.C for their functioning. Most of the C.F.C impoverish the layer of ozone and participate with a considerable manner, to the increase of the sere effect, then to climatic changes. To contribute to the rational utilisation of energy and the protection of the environment, the research undertaken in the area of the cold has been mainly oriented to a new process adaptation in many fields like the food product conservation and climatisation. This can be obtained by the utilisation of process and sources of not conventional energy like the combined absorption process and solar energy. Besides the environmental aspect, these refrigerator systems are of great interest in the far rural zones which are not served by the electrical network, particularly for food product conservation and the air conditioning of tourist infrastructures and hotels in distant rural zones. Absorption cycles are autonomous systems which can provide a solution to this problem. After modelisation of the absorption refrigerator machine functioning with LiBr-H₂O, by determining energy and mass statements of each cycle element supposed in the permanent regime. We have determined the COP of the solar single-stage absorption refrigerator as a function of the temperature in the different components of the cycle. Therefore, we have changed the dimension of the cycle by introducing double-line heat exchanger (between generator-condenser and between evaporator-generator). In this study, we present the variation of the COP and the heat-transfer rate with different temperature of the cycle, the obtained values are compared with an improved configuration using a double- line heat exchanger.

Keywords: Refrigeration cycle - Absorption refrigeration - Solar energy - COP.

1. INTRODUCTION

Solar cooling is a very attractive topic for researches of the solar thermal system field. The reason is that the cooling load usually reaches its peak value when solar energy is mostly available. Most of the research works in this field are carried out employing either a vapour compression cooling cycle or an absorption cooling cycle. The absorption cooling cycle is usually a preferable alternative since it uses thermal energy collected from the sun without the need to convert this energy into mechanical energy as required by the vapour compression cycle. In addition the absorption cycle uses thermal energy at a lower temperature than that dictated by the vapour compression cycle.

Most of the absorption cooling cycles use either LiBr-H₂O or NH₃-H₂O solutions. In solar applications the LiBr-H₂O system is superior to the NH₃-H₂O system for several reason. Among these reasons are that the LiBr-H₂O system is simpler in design and operation, and cheaper in cost as compared to the NH₃-H₂O system. Also, the LiBr-H₂O system can operate at a low generator temperature and this range of temperature the system has a COP (Coefficient of Performance) higher than that of the NH₃-H₂O system. The LiBr-H₂O can be operated by the simple flat plate collectors while the NH₃-H₂O systems operates with evacuated tubes collectors or solar concentrators. The NH₃-H₂O system is restricted in building applications because of the hazards associated with the use NH₃ whereas LiBr-H₂O systems are safe.

A part from the many advantages of the LiBr-H₂O system mentioned above, the system ails to function if the ambient (heat sink) temperature is increased. At high ambient temperature the system may either suffer from crystallisation at the absorber inlet for the strong solution, or from having water vapour in the suction line of the solution circulating.

Considerable research has been carried out to develop an efficient and a economic coupling between solar energy collection and the absorption unit [1]. The performance of absorption cycles has been studied extensively in the recent past using computer simulation [2-4].

The aim of the present study is to perform a direct analysis of the solar LiBr-H₂O absorption refrigerating cycle. The variation of the different heat and flow of the poor solution with generator temperature is obtained. The COPs (Solar Coefficient of Performance) of the cycle is obtained as a function of the different temperature of the cycle and concentration, more we have studied the effect of using a heat exchanger to recover part of the condenser load to heat LiBr-H₂O solution going to the generator.

2. DESCRIPTION OF THE SOLAR SINGLE-STAGE ABSORPTION CYCLE

A schematic representation of the solar single-stage absorption cycle is shown in Fig. 1. The cycle consists of generator, absorber, a condenser, an evaporator, two heat exchanger, circulating pumps and solar collector. The cycle working between two pressure levels : low pressure at the evaporator-absorber and high pressure at condenser-generator. The emitted vapours in the generator are constituted by pure water when in LiBr salt remain in the solution.

In the evaporator, heat input of the cold bath to T_e is used in the boiling the liquid water. The produced vapours are absorbed in the absorber by the rich concentration X_r LiBr solution (kg of LiBr/kg of the solution). This latter becomes poor $X_p \ll 1$ LiBr solution before reaching the generator. Once in the generator the solution receives the heat which allows to take the water to its boiling point. The solution enriched in LiBr returns to the absorber via a detender. The vapours of refrigerant fluid (H_2O) to high pressure are sent in the condenser while the liquid leaving it returns to the evaporator. The presentation of the cycle in P-T-X diagram is given in the figure 2, where numbers refer to corresponding state points in the figure 1. The super stars of the numbers in the figure 2 indicate the corresponding equilibrium temperatures for a given pressure and concentration. The continuous lines in the figures 1 and 2 refer to pure H_2O (i.e. refrigerant).

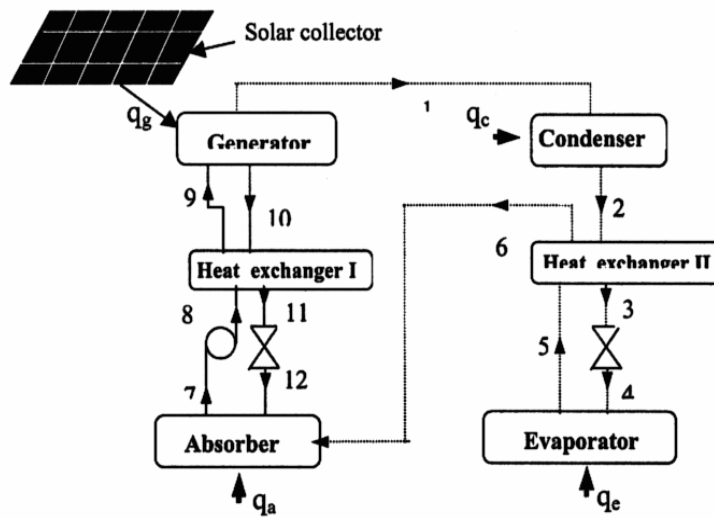


Fig. 1: Schematic of the single-stage absorption cycle

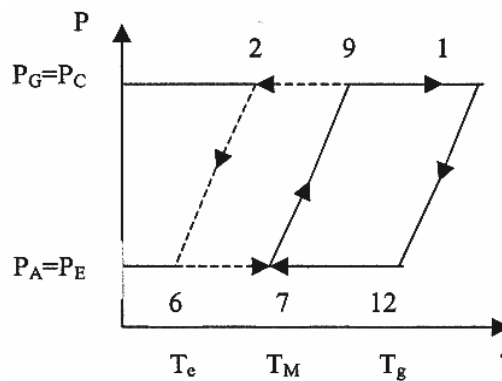


Fig. 2: The presentation of the single-stage absorption cycle on the P-T-X diagram

3. SIMULATION OF THE SOLAR SINGLE-STAGE ABSORPTION CYCLE

The simulation is based on mass and energy balances for each element of the cycle, supposed in permanent regime. The equilibrium pressure of the vapour phase or liquid P_s (kPa) is determined according to the water saturation temperature T_s ($^{\circ}C$) by using the equation proposed by ASHRAE [5].

$$\ln P = 7.05 - \frac{1596.49}{(T_s + 273.15)} - \frac{104095.5}{(T_s + 273.15)^2} \quad (1)$$

The vapour emitted at the generator level, which circulates through the condenser and the evaporator, is formed by pure water. Then Y (kg of LiBr/kg of the solution H₂O-LiBr) is equal to zero. Hence, we only need to calculate the title X (kg of LiBr/kg of the liquid solution H₂O-LiBr) of the H₂O-LiBr liquid solution circulating between the generator and the absorber. The title is given by the following equation [6]:

$$T = T_s \times \sum_{n=0}^3 A_n X_n + \sum_{n=0}^3 B_n X_n \quad (2)$$

where T (°C) is the equilibrium temperature of H₂O-LiBr solution at a given concentration X and at a pressure P corresponding to T_s (°C). The A_n and B_n coefficient are given as follows:

i- For $0 \leq X \leq 0.45$; $0 \leq T_s \leq 20$ °C and $0 \leq T \leq 144$ °C

N	A _n	B _n
0	1.00	0.0
1	-0.4427	34.108
2	2.64478	-193.02
3	-2.7419	414.90

ii- For $0.45 \leq X \leq 0.7$; $0 \leq T_s \leq 120$ °C and $5 \leq T \leq 174$ °C

N	A _n	B _n
0	-2.00755	124.94
1	16.976	-771.649
2	-31.33362	1522.858
3	19.7668	-795.09

The specific enthalpy of the liquid water h_l (kJ/kg) and the vapour water h_v (kJ/kg) are determined as follow [7]:

$$h_l = 4.199 T \quad (3)$$

$$h_v = 1.73 T + 2501.6 \quad (4)$$

In the case of the solution, the specific enthalpy h (kJ/kg) can be calculated according to the solution temperature T (°C) and the mass fraction X of the solution H₂O-LiBr by [8] :

$$h(T, X) = \sum_{n=0}^4 C_n X_n + T \times \sum_{n=0}^4 D_n X_n + T_2 \times \sum_{n=0}^4 E_n X_n \quad (5)$$

The various calorific powers are as follows :

$$q_g = h_1 - h_9 + (h_{10} - h_9) \times \frac{X_9}{X_{10} - X_9} \quad (6)$$

$$\frac{q_q}{q_{ml}} = h_{12} - h_6 + (h_7 - h_{12}) \times \frac{X_{12}}{X_{12} - X_7} \quad (7)$$

$$\frac{q_c}{q_{ml}} = h_2 - h_1 \quad (8)$$

$$q_e = q_{ml} \times (h_5 - h_4) \quad (9)$$

Neglecting the pumping work, the COP (Coefficient of Performance) of the cycle is estimated as follows :

$$\text{COP} = \frac{q_e}{q_g} \quad (10)$$

The efficiency of a collector depends on its temperature. The higher the temperature of a given collector, the greater its heat loss. In designing a solar absorption refrigerator, the collector efficiency (η_s), which is defined as the ratio of the output (q_g) to the incident solar power (I_s), is very important :

$$\eta_s = \frac{q_g}{q_i} \quad (11)$$

Taking into account the temperatures reached by solar collectors, we have selected the flat plate solar collectors. The efficiency of which is given by Hottel - Whilier [9]:

$$\eta_s = F_b \left(\eta_0 - \frac{U(T_g - T_c)}{I_s} \right) \quad (12)$$

where :

$$F_b = F_r (1 - F_r \beta)^{-1} \quad , \quad F_r = \frac{1 - \exp(-\beta)}{\beta} \quad , \quad \beta = \frac{U}{mC_p}$$

with the following characteristics: $\eta_0 = 0.7$ is the optical efficiency, $U = 2 \text{ W/m}^2\text{K}$ is the coefficient of thermal losses, $m = 50 \text{ kg/hm}^2$ is the flow rate by surface unit, $I_s = 800 \text{ W/m}^2$ is the global irradiation of an inclined plane, $C_p = 4186 \text{ J/kgK}$ is the specific heat of working fluid (water), T_g is the temperature of heat source and T_c is the condenser temperature.

The overall COPs of the solar absorption refrigerator is equal to the product of the efficiency of the solar collector (η_s) and the COP of the absorption refrigerator given by equation [10].

$$\text{COPs} = \eta_s \text{ COP} \quad (13)$$

4. THE DOUBLE - LINE HEAT EXCHANGER

Heat exchangers, usually known as economisers, are used in the absorption cycle to recover the heat from the strong solution leaving the generator and thus improving the COP of the absorption cycle. The heat carried out by the refrigerant leaving the generator is not recovered by this means. As the generator temperature is increased the amount of heat carried by the refrigerant, and available for recovery, is also increased. It is now suggested to recover this heat in the heat exchanger to preheat the weak solution going to the generator. This is done as shown in the figure 3 where H.E. no.2 is integrated with the condenser to reduce the cost. The extra cost involved in including the refrigerant line to the heat exchanger is compensated by the reduction in the cost of the condenser since its thermal load is reduced.

The previous analysis for the cycle with single-line heat exchanger (Fig. 1) is also applied for the cycle with double- line heat exchanger except the following changes :

$$q_c = q_{m1} \times (h_{1a} - h_2) \quad (14)$$

$$T_{1a} = T_{11} \quad (15)$$

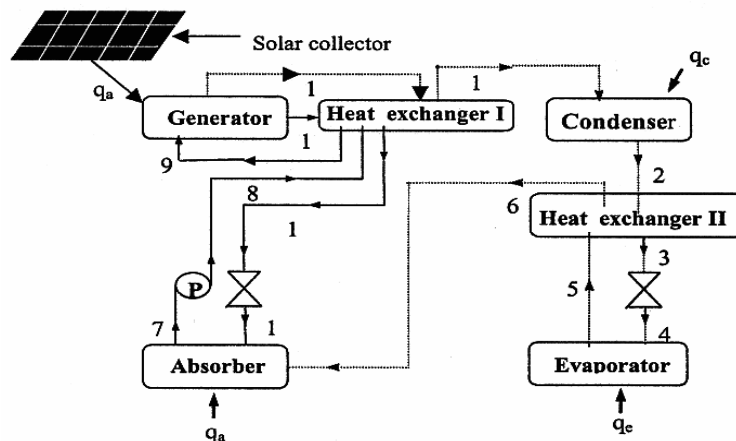


Fig. 3: Schematic of the single-stage absorption cycle using double line heat exchanger

5. RESULTS AND DISCUSSION

The temperature T_m of the average source has a considerable influence on the threshold of cycle functioning (T_g). The figure 4 illustrates the evolution of the COPs according to T_g for different values of T_m and for $T_e = 10^\circ\text{C}$. One notices that the functioning temperature limits T_g , that corresponds to a null COPs, grow when T_m increases. Contrarily to the $\text{NH}_3\text{-H}_2\text{O}$ absorption refrigerating cycle [10], the COPs depends slightly on the generator temperature T_g due to the fact that the refrigerant is pure. One observes a diminution of the COPs according to T_m . More this figure represents flows of heat exchanged by the different elements of the absorption cycle for $T_e = 10^\circ\text{C}$, $T_m = 20^\circ\text{C}$ and $q_{ml} = 1 \text{ kg/s}$. One observes that the refrigerator effect q_e remains practically constant when the temperature T_g increases.

In presenting the results, one temperature is changed while the others are held constant. To study the effect of using a double-line heat exchanger on the performance of the solar single-stage cycle, the COPs of both cycle are calculated at various values of the parameters design. The results are plotted in Figs 5 and 6. The use of a double line heat exchanger improves the COP by about 4 % as compared to its value using a single-line heat exchanger, for both cycles. The figure 7 depict the variation of q_g and q_c with T_g . The at the generator and the condenser are decreased in comparison with its value using heat-transfer a single-line heat exchanger.

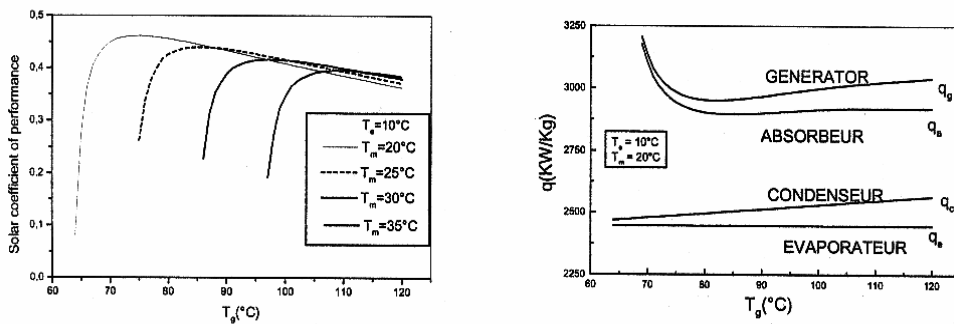


Fig. 4: Variation of the COP and exchanged heat of the different organs of the solar single-stage absorption cycle with T_g (single-line H.E.)

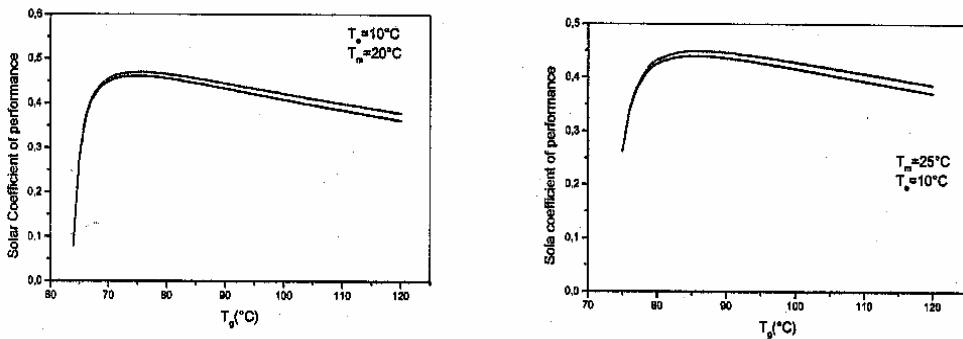


Fig. 5: Variation of the COP of the solar single-stage absorption cycle with T_g (---- double-line H.E., — single-line H.E.)

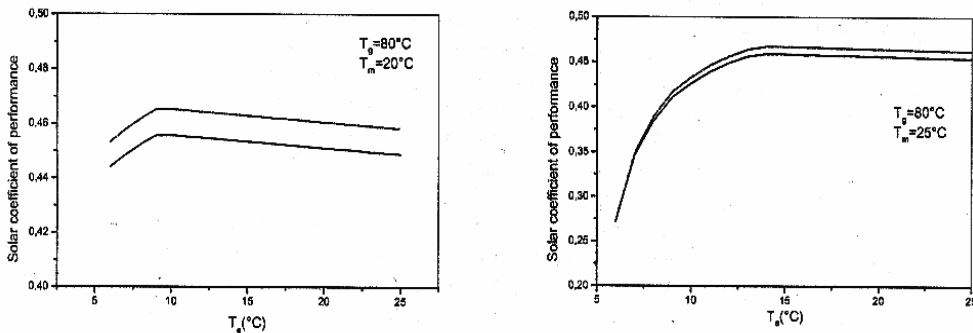


Fig. 6: Variation of the COP of the solar single-stage absorption cycle with T_e (---- double-line H.E., — single-line H.E.)

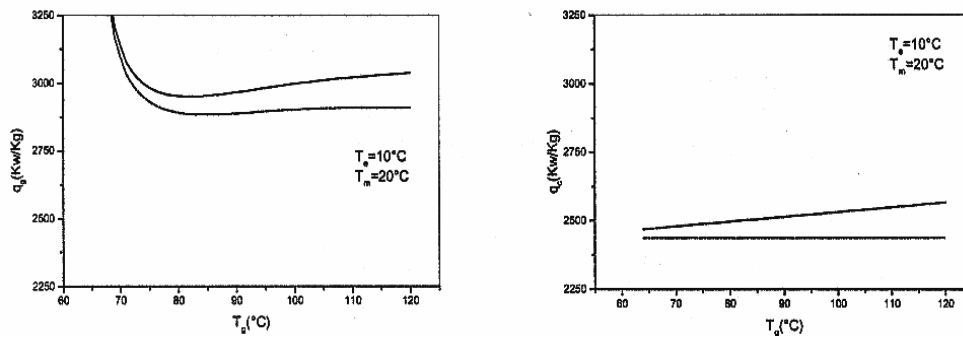


Fig. 7: Variation of the q_g and q_c of the solar single-stage absorption cycle with T_g (---- double-line H.E., — single-line H.E.)

6. CONCLUSION

In this paper, we have developed a method of calculation that is based on simple analytic data which relate the thermodynamic variable of the H_2O -LiBr fluid couple. Thus, we were able to describe perfectly the thermodynamic mixture state of H_2O -LiBr fluid, in different steps of the absorption cycle without having to use data or diagram tables. The simulation results have revealed that there is a temperature limit "underhand/to the over" of which the cycle can not working. Moreover, the temperature of the cold source T_m has an influence on the threshold of functioning of the cycle. The COP reaches maximal values for $T_m = 20^\circ\text{C}$ or 25°C and $T_e = 10^\circ\text{C}$. The use of a double line heat exchanger to recover a part of the condenser load improves the COP of both cycles by about 4 %

Actually we develop new approaches based an thermodynamic irreversible model [11]. The most important point in the study is the application non-equilibrium phenomenological theory of mass and heat transfer, with special emphasis few case including physical binary gas liquid interaction in non-ideal mixtures [12, 13].

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