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NUMERICAL SIMULATION AND ANALYSIS OF A SCALE ICEMAKER WATER-AMMONIA ABSORPTION SYSTEM.

Rodolfo J. Rodríguez Silverio.
Rafael L. Pratts Milanés
Prof. Dr. José Ricardo Figueiredo.
State University of Campinas, Department of Energy.
Cx. P. 6122 – 13083-970 – Campinas, SP, Brasil

Abstract: *Absorption heat pumps have received growing attention in the past two decades. The increasing cost of electricity has made this heat-powered cycle attractive for both residential and industrial applications.*

Further development of this technology demands reliable system simulations. Several computer models have been developed that have proven to be very valuable tools for design optimization. However, must of them do not take into account the heat and mass transfer characteristics.

The objective of this work is to create a mathematical model covering thermodynamics, heat and mass transfer and fluid mechanics phenomena, involving both constructive and external parameters, in order to simulate numerically the steady state performance of one absorption refrigeration system. This model was validated against experimental data with good agreement.

Key Words: *absorption refrigeration, ice making, mathematical model, simulation.*

1. INTRODUCTION

The studies conducted in recent years on absorption heat pumps have created a need for reliable and effective system simulations. Several computers models have been developed that have proven to be very valuable tools for design optimization (Vliet et al, 1982; Linden and Klein, 1985; Grossman et al, 1987). However, must of them do not consider the heat and mass transfer characteristics of the heat exchangers.

On the other hand the absorption refrigeration literature describes several systems developed in the last decade, but only limited information is given on their heat and mass transfer characteristics, with test data confined to a narrow range around the design point (Grossman, 1995).

The present work searches for a mathematical model taking into account the thermodynamics, heat and mass transfer and fluid mechanics phenomena, involving both constructive and external parameters, in order to simulate numerically the steady-state performance of one absorption refrigeration system.

The experimental system is for scales ice production and is located at Hospital of Clinics of the State University at Campinas (UNICAMP). Although it was initially projected for work on fishing boats using Diesel engine exhaust gas, it is driven by a small part of the steam produced for the use of the hospital.

The heat and mass transfer coefficients for the simulation processes were estimated from a set of processed experimental data (Pratts et al, 1999). The performance results obtained by means of the simulation algorithm were compared with the experimental counterpart and showed a good agreement.

2. SIMULATION PROCESS.

The simulation process was based on the simple basic cycle shown in figure 1. For the sake of simplicity the auxiliary equipment that is necessary for improving the performance coefficient is not represented. The evaporator is configured to produce ice over its walls and the condenser, absorbent and week solution cooler are evaporative exchangers located inside a unique evaporative cooling tower. A total description of the system can be revised in Pratts et al (1999).

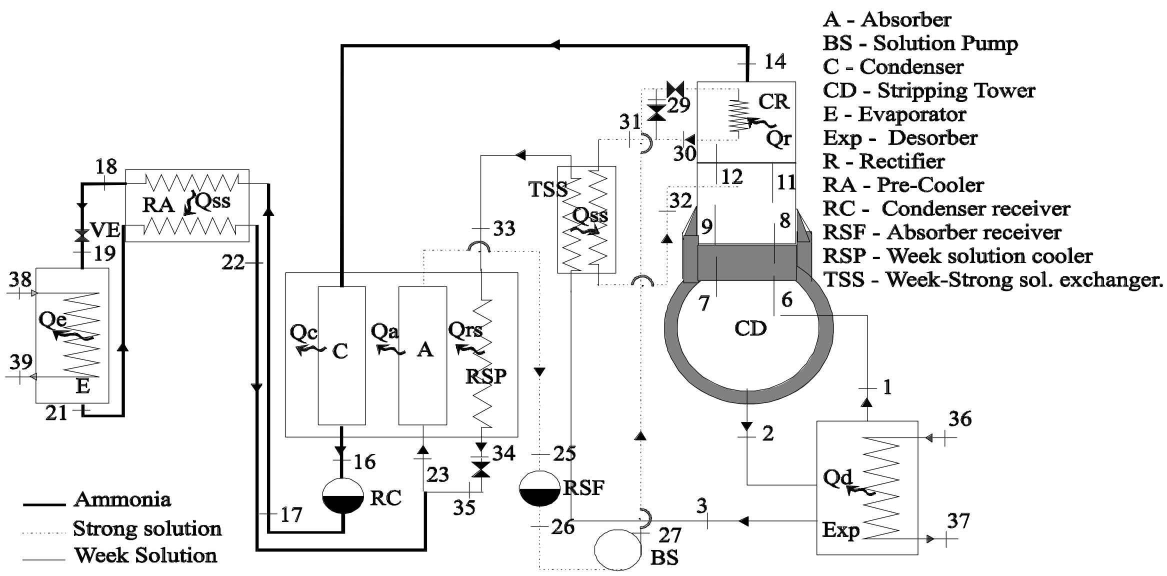

Fig.1 Schematic description of the absorption system.

2.1. Mathematical model.

Due to the ice production characteristics, it is formed directly on the evaporator walls, and because the reverse defrost cycle, both the evaporating temperature and evaporator’s overall heat transfer coefficient vary with the time and make the system operation cyclically transient all the time.

In order to know the optimal cycle period in function of the external parameters and for comparing the simulate with the actual system performance coefficient both steady state and transient model were established. However for the sake of simplicity only the steady-state operation is presented here.

Steady-state operation model. To set up the steady-state mathematical model each component is treated as a control volume, with its own inputs and outputs, then the following physical law equations are applied:

- Conservation of total mass:

$$\sum_{i} m_i = 0 \tag{1}$$

where m is the mass flux in kg/s.

- Conservation of mass for each material species:

$$\sum_{i} m_i X_i = 0 \tag{2}$$

where X is the solution concentration

- Energy balance in heat exchangers:

$$\sum_{i} m_i h_i = Q_{unid} \tag{3}$$

where h is the enthalpy of the fluid in kJ/kg and Q is the heat rate in kW.

- Heat transfer, expressed in one of the following forms:

$$Q_{unid} - (UA)_{unid} \Delta T_{unid} = 0 \tag{4}$$

$$EFE_{unid} - \overline{EFE}_{unid} = 0 \tag{5}$$

$$(NTU)^{R}_{unid} - (NTU)^{A}_{unid} = 0 \tag{6}$$

where U is the overall heat transfer coefficient in kW/m^2K, A is the surface heat transfer in m^2, ΔT is the true mean temperature difference, EFE and $\overline{EFE}$ are both the design and calculated heat transfer effectiveness of the heat exchangers and $(NTU)^{R}_{unid}$ and $(NTU)^{A}_{unid}$ are both the required and the available number of transfer units for the particular evaporative heat exchanger, as defined inWebb (1984).

- Thermodynamic state equation for the refrigerant-absorbent pair:

$$f(P_i, T_i, X_i) = 0 \tag{7}$$

This complex function can be reviewed inZigler and Trepp (1984)

- Mass transfer, expressed in terms of temperature deviation from equilibrium (DEV):

$$T_i = T_{i\,E}(P_i, X_i) + DEV \tag{8}$$

where T is the actual temperature of the fluid and T_E is the equilibrium temperature.

- Pressure drop in pipes and exchangers:

$$P_i = P_{i-1} - \Delta P(Re, \varepsilon, \frac{L}{Di}) \tag{9}$$

where Re is the Reynolds number of the flux, ε is the relative roughness of the conduct surface and L and Di are the length and internal diameter of the conduct respectively.

- Solution pump work:

$$W_B = \frac{(P_o - P_i)m}{\rho\eta_B} \tag{10}$$

where P_o and P_i are the output and input pressures respectively, ρ is the fluid density and η_B is the actual efficiency of the pump.

- Coefficient of Performance:

$$COP = \frac{Q_E}{Q_D + Q_B} \tag{11}$$

where Q_E and Q_D are the evaporation and desorption heat transfer rates respectively.

- Internal and external ideal coefficient of performance:

$$COP_C^i = \frac{\frac{1}{T_A} - \frac{1}{T_D}}{\frac{1}{T_E} - \frac{1}{T_C}} \tag{12}$$

where T_A, T_D, T_E, T_C are the absolute temperatures of absorption, desorption, evaporation and condensation respectively.

$$COP_C^e = \frac{\frac{1}{T_b} - \frac{1}{T_v}}{\frac{1}{T_g} - \frac{1}{T_b}} \tag{13}$$

where T_b, T_v, T_g, are the absolute wet-bulb, heating vapor and ice temperatures respectively. As the system produces ice directly on the wall of evaporator, it is driven by steam. The condenser and absorber are evaporative exchangers.

- Internal and external relative efficiencies:

$$\eta_r^i = \frac{COP}{COP_C^i} \qquad (14)$$

$$\eta_r^e = \frac{COP}{COP_C^e} \qquad (15)$$

2.2 Solution Algorithm

When the 10 first equations of the steady-state model are applied to each component, a set of nonlinear simultaneous equations is formed for the entire system. This is solved by the Newton-Ramphson method for nonlinear equation system, which expressed the equations in the form of residuals that must be reduced to zero or, in practice, to a value below a given tolerance:

$$\begin{aligned} &F_1(X_1,X_2....X_n) = \delta_1, \\ &F_2(X_1,X_2....X_n) = \delta_2, \\ &\quad \vdots \\ &F_i(X_1,X_2....X_n) = \delta_i, \\ &\quad \vdots \\ &F_n(X_1,X_2....X_n) = \delta_n \end{aligned}$$

with $|\delta_i| \to 0$ at the solution

By its nonlinear nature the system requires many iterative loops for solving the equations of state, the combined heat and mass transfer in the evaporative exchanger, etc. On the other hand there are many unknowns that may require a complex initial guesses. For these reasons an algorithm was created that permits to select m unknowns to work as implicit variables while the others act as dependent explicit variables. With this approach the nonlinear algorithm of solution reduces to an n-m equations system. This algorithm can be see inFigueiredo (1980).

2.3 Discussion of results

Table 6.1 shows the comparison between some of the experimental and simulated parameters of the installation in steady-state operation. These heat fluxes were calculated assuming average 23°C wet-bulb temperature. A large error can be seen in the estimation of solution pump power consumption due to the pump volumetric efficiency, which was not considered. All the others parameters present errors lower than 10 %.

Figure 2 gives a broad view of the external heat transfer rates when the wet bulb temperature vary from 20 to 30°C. In general all the rates diminished with the increase of the wet-bulb temperature, but the whole effect can be seen in figure 4 where the large decrease of the Coefficient of Performance for the entire unit is presented.

Table 6.1. Comparison between experimental and simulated parameters

Parameter	Experimental	Simulated	error [%]
Q_E[KW]	21.43	22.66	5.73
Q_C[KW]	23.90	22.02	7.86
Q_A[KW]	35.57	34.79	2.19
Q_D[KW]	51.42	49.58	3.55
Q_R[KW]	15.76	16.18	2.66
Q_B[KW]	0.57	0.43	24.3
Q_{SR}[KW]	1.67	1.71	2.39
Q_{RSP}[KW]	17.71	16.06	9.31
Q_{TSS}[KW]	40.218	44.14	9.75
COP[%]	41.94	45.42	9.87

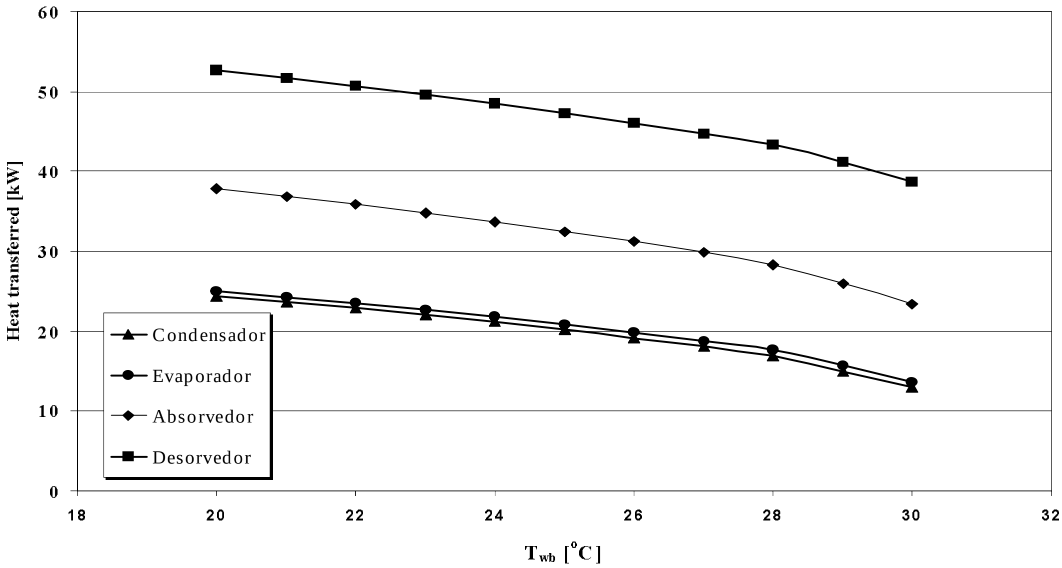

Fig. 2. External heat transfer rates variation.

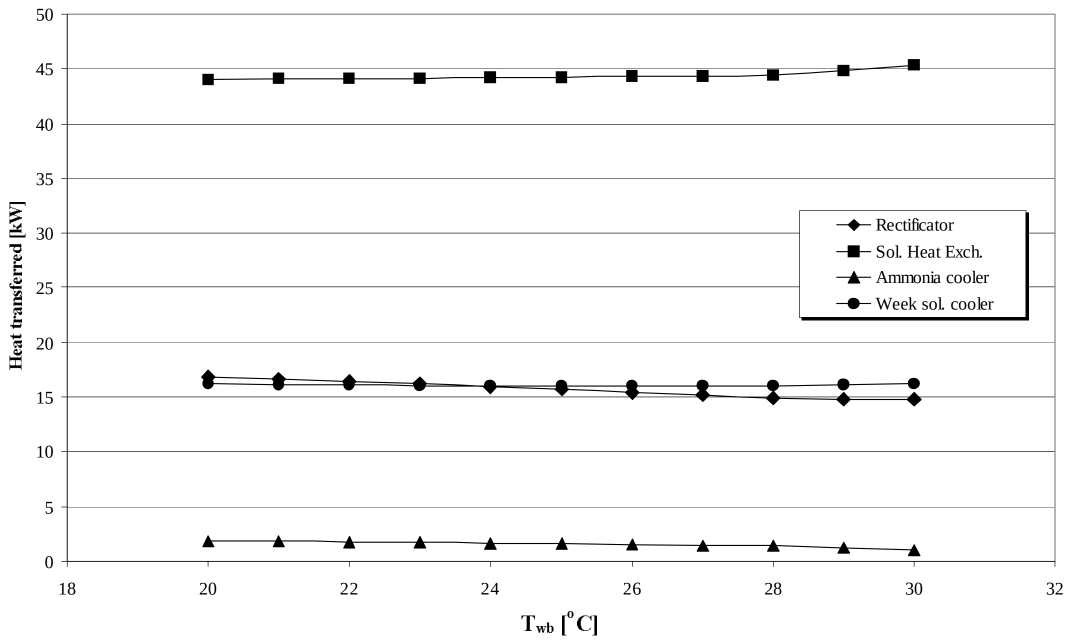

Fig. 3. Internal heat transfer rates variation.

In figure 3. are shown the internal heat transfer rates variation with respect to wet-bulb temperature. It can be seen that these rates are almost insensible to the external wet-bulb temperature. Indeed the week solution evaporative cooler changes heat directly with the surrounding air, but it is presented in fig. 3 to compare with the rectifier. The heat transfer rate is almost the same for both exchangers showing that there is not heat recovery from the rectification process.

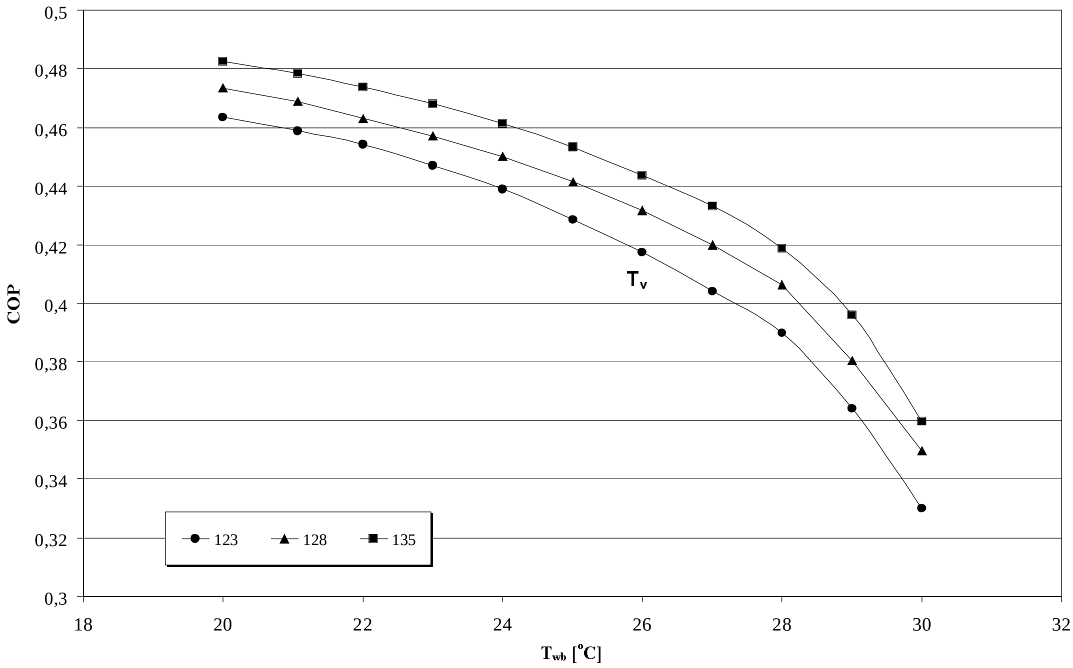

Fig. 4. Variation of the Coefficient of Performance.

Figure 4 shows the variation undergone by the Coefficient of Performance of the system with respect to the wet bulb temperature for different heating vapor temperatures, while figure 5 shows the variation of opposite effect.

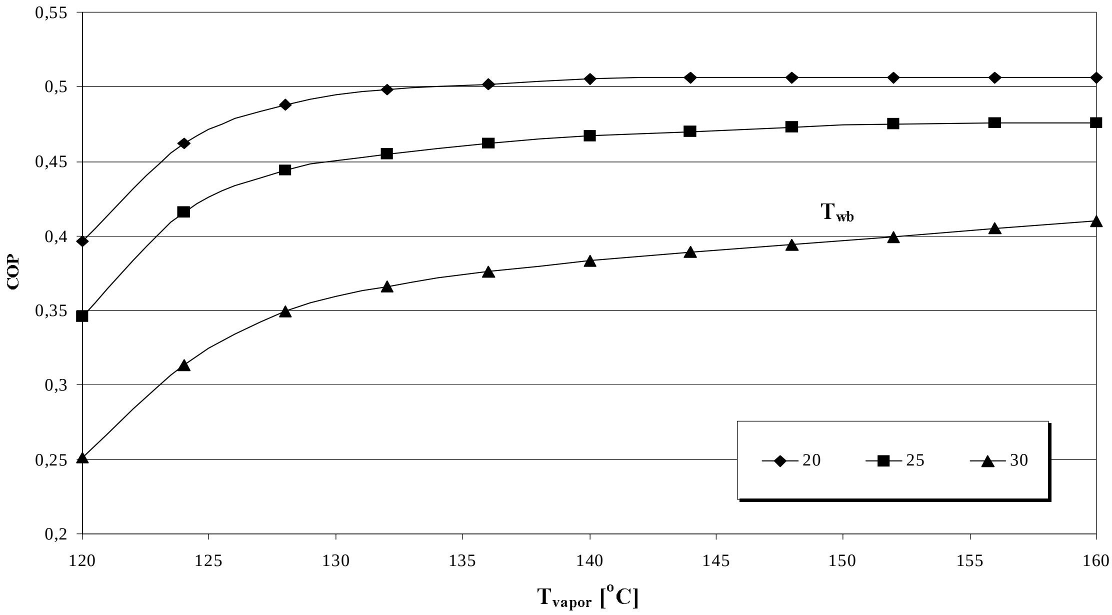

Fig. 5. Variation of the Coefficient of Performance.

In the above figures it is shown how is the variation of the COP with respect to the more relevant external parameters: the wet-bulb temperature and the heating vapor temperature. Observing these figures it can be concluded that for the present system the wet-bulb temperature have a more significant role on its Coefficient of Performance than the heating vapor temperature.

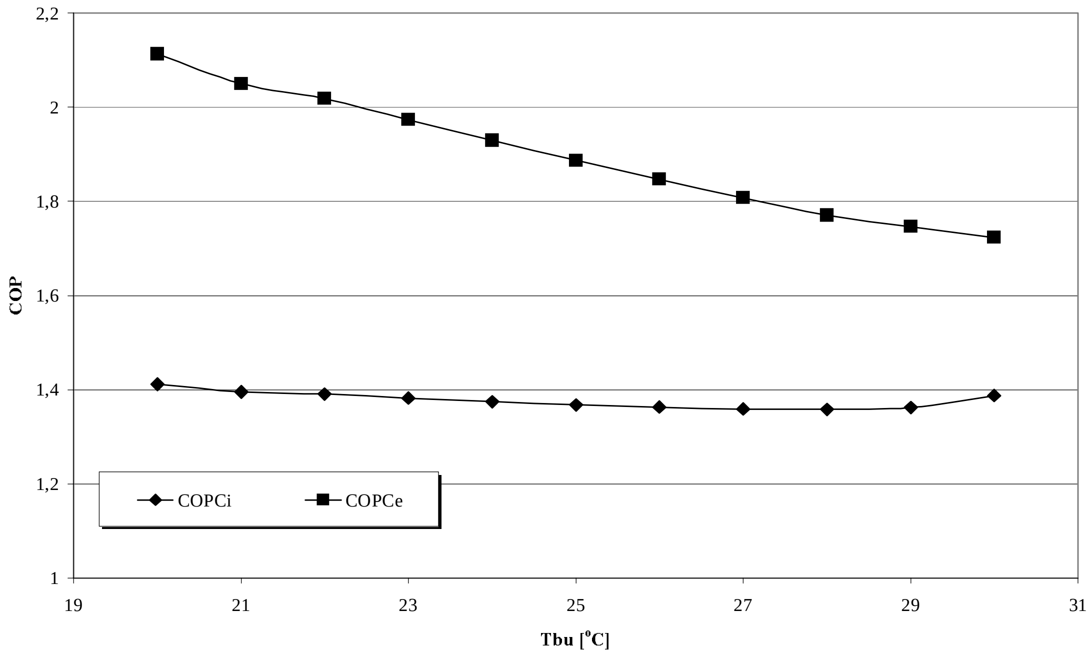

Fig.6. Variation of the Ideal Coefficients of Performance.

Figure 6 presents the variation of the ideal Carnot cycles with regard to both internal and external temperatures. In Pratts (1999) similar results were obtained for the same analysis but using average experimental values.

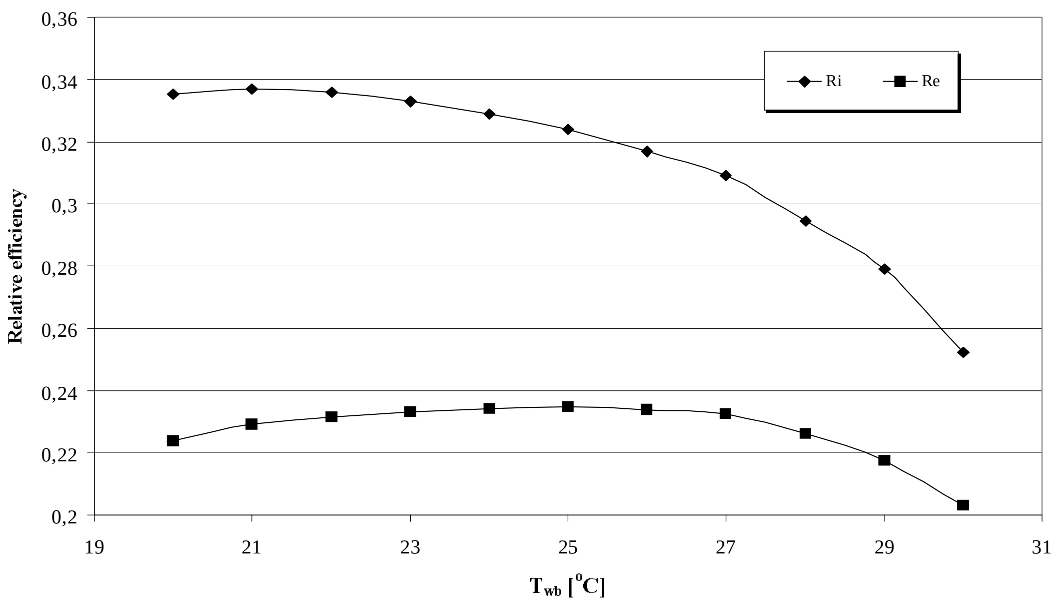

Fig.7. Variation of the relative efficiencies.

The relative efficiencies as defined in equation 14 are shown in figure 7. These indicate that the Coefficient of Performance of the system is of low order of magnitude when compared with others installation reported by several specialized publications, but the values of the relative internal efficiency are of the same order of those already reported, appointing to inadequate internal temperatures.

Analyzing the actual absorption temperatures in the cycle, that are of the same order of the simulated (Pratts et al., 1999), it can be concluded that the design of the system must be improved by increasing the heat and mass transfer area of the absorber.

This result is confirmed by figure 8, which shows the variation of the Coefficient of Performance with respect to several (UA) values. The system COP is significant damage with decrease of (UA) values below design point, while the improve is small when (UA) values increase above the design point $(UA)_D$. However an increase on the (UA) value for the absorber carry to a considerable improvement of the system COP. Although the four equipments are represented together each curve assume that the (AU) value of one equipment varies keeping the others unchanged.

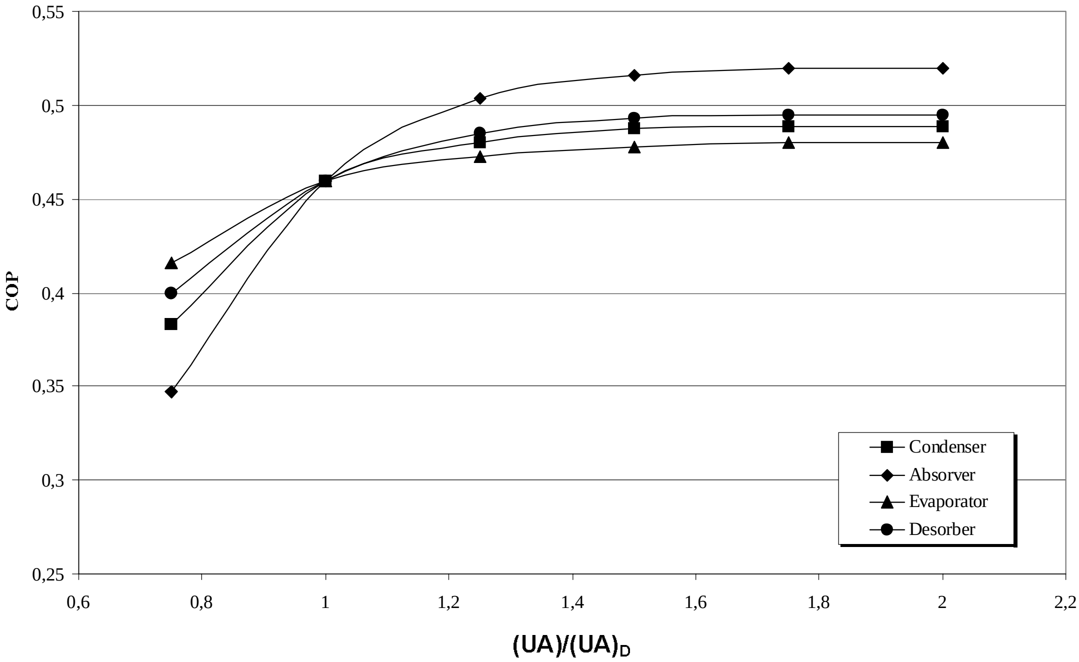

Fig. 8. Effect of the (UA) values on the Coefficient of Performance.

2.4 Conclusion

The steady state operation of a icemaker based on water-ammonia absorption refrigeration system are simulated. It was demonstrated that system COP is low due principally to the high temperature of the rich solution leaving the absorber, but also was showed that an increase in the absorbent heat transfer surface will significantly improve the COP.

The analysis presented in this work was based on the external Coefficient of Performance, which takes into consideration all the irreversibilities in the transfer of heat that occurs in the whole system. In order to know the degree of efficiency of each system component it is necessary an exergetic evaluation. The exergetic method to determine the loss of work capacity or availability is a very important qualitative analysis, which gives a clear pictures of the main sources of irreversibilities of the cycle.

On the other hand Bejan et al. (1995) are demonstrated through thermodynamic optimization (or entropy generation minimization) that there exist an optimal allocation of a heat-exchanger inventory for which the COP is maximum.

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