COBEM 99
XV Congresso Brasileiro de Engenharia Mecânica
15th Brazilian Congress of Mechanical Engineering
22 - 26 de Novembro de 1999 / November 22 - 26, 1999 Águas de Lindóia, São Paulo,

THERMODYNAMIC EVALUATION OF A SCALE ICEMAKER WATER-AMMONIA ABSORPTION SYSTEM BASED ON EXPERIMENTAL DATA.

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

Abstract: *Because of the recent energy crisis, worldwide efforts are concentrated on the energy conservation, recovery and utilization. Absorption heat pumps play a crucial role in all theses concerns. On the other hand, the importance of the cooling technology for the tropical agricultural developing countries have necessary that a lot of efforts have to be derived for the technological developments that still must be done and to preparing the human source for its application at these countries.*

The present work is a thermal evaluation based on experimental data of an icemaker water-ammonia absorption system. This evaluation employs both First and Second Law of Thermodynamic concept and revealed where the principal sources of irreversibilities are, and how far the actual system is from the ideal cycle.

Key words: *Absorption refrigeration, icemaker, energetic and exergetic evaluation.*

1. INTRODUCTION

Because of the recent energy crisis, worldwide efforts are concentrated on the energy conservation, recovery and utilization (Shah et al, 1998). Absorption heat pumps play a crucial role in all theses concerns. On the other hand, the importance of the cooling technology for the tropical agricultural developing countries have necessary that a lot of efforts have to be derived for the technological developments that still must be done on these systems.

This paper presents a study made on an actual water-ammonia absorption system for making two tons of scale ice per day. This experimental plant is located at Hospital of Clinics of the State University at Campinas (UNICAMP). Although it was initially projected for work on ships using Diesel engine exhaust gas, it is driven by a little part of the vapor produced for the use of the hospital.

For this thermal evaluation both First and Second Law of Thermodynamic concepts were used employing a set of experimental data that was completed by means of a steady-state simulation computer program (Silvero et al, 1999). This evaluation has revealed where the principal sources of irreversibilities are, and how far the actual system is from the ideal cycle.

2. THERMODYNAC EVALUATION

Figure 1 shows the simple basic cycle of the actual icemaker based on water-ammonia absorption refrigeration system. For the sake of simplicity was not represented the auxiliary equipment which is necessary for guarantee an adequate performance coefficient. The evaporator is configured to produce ice over its wall and the condenser, absorbent and week solution cooler are evaporative exchanger all located inside a unique evaporative cooling tower.

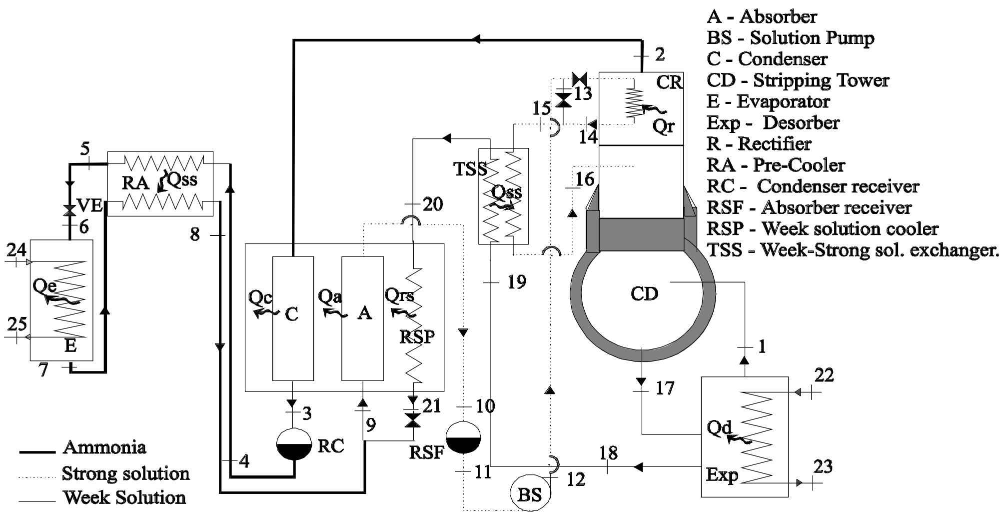

Fig.1. Schematic description of the absorption system.

2.1 Experimental data system

It was installed a data acquisition system by which values of pressure, temperature and flux and density of the working fluid are known at several points. Figure 2 shows a schematic representation of the experimental sensors system employed for the data acquisition.

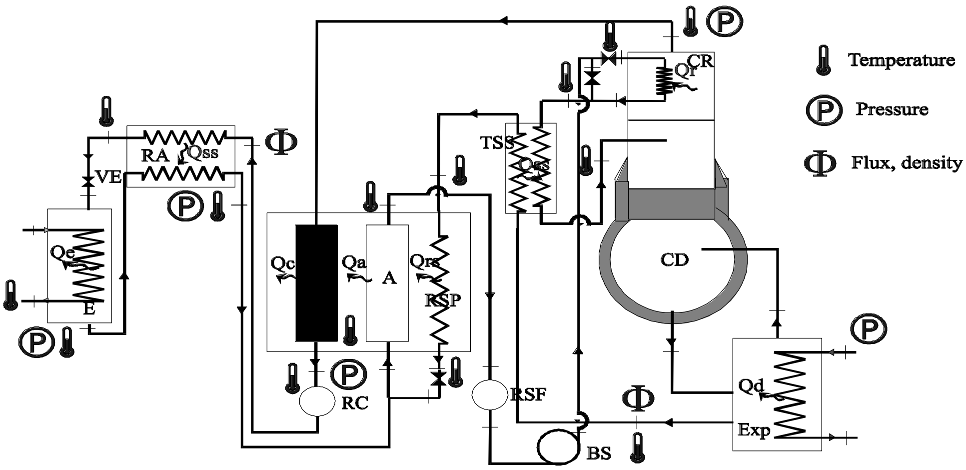

Fig.2. Schematic description of the experimental sensor system.

After being detected by the sensors all the data are collected transform in digital signals and processed by means a computer program. Using this data set, the thermodynamic properties of the fluid at each point are determined through the AQUAM software created by Jordan (1992). The another interest points that cannot be obtained by direct experiments are estimated via steady-state simulation computer program (Silverio et al, 1999).

2.2 Evaluation methodology

The method to be employed for thermodynamic evaluation consists in determining the internal operational variables under steady-state operation at all points of interest throughout the cycle. The solution of energy and material balance for each element system combined with the set of experimental data and a computer steady-state performance simulation program (Silverio et al, 1999) to obtain the unknowns experimental variables, permit the evaluation based on both First and Second Laws of thermodynamics.

Performance Coefficient First Law Analysis. The Carnot cycle has been adopted by present regulations in the refrigeration field as the ideal cycle absorption systems. The Carnot cycle is characterized by a complete reversibility of every process taking place in the cycle. Therefore in the absorption refrigeration cycle the following conditions must be fulfilled:

i) Isentropic expansion of the refrigerant solution from the condenser to the evaporator and also to the generator to the absorber. This expansion is supposed to be done in an expansion engine from that is obtained the power needed to the pump solution. Therefore no work or mechanical energy external to the system is required.

ii) Complete rectification of the refrigerant vapor is obtained in the desorber thus only pure refrigerant is flowing in the operational circuit. This is equivalent to consider that in the binary mixture has no vapor pressure and there is not necessity of the rectification heat.

iii) The working fluid is a mixture of ideal substance. Thus the specific heat of the refrigerant, either as a vapor or as a liquid, is nil trough the limiting curves. Both the evaporation heat and the heat of solution are constant depending only of the concentration and not of the temperature.

iv) The ideal Carnot cycle of an absorption refrigeration system is considered a combination of a heat engine and a refrigerating machine.

The performance coefficient of any refrigerating system is measured by the Coefficient of Performance (COP) which is the ratio of the total cooling effect to the energy input to the system.

From fig. 1 the energy balance of the whole system is expressed by:

$$Q_D + Q_E + \sum W = Q_A + Q_C \tag{1}$$

where Q_D is the desorption heat, Q_E the evaporation heat, W power input to the solution pump, Q_A the absorption heat and Q_C is the condensation heat. Refrigeration output = Q_E and energy input = Q_D+W_B. Thus for the actual system:

$$COP = \frac{Q_E}{Q_D + W_B} \tag{2}$$

For the Carnot cycle of this system it is necessary to take into account the set of condition from (i) to (iv) already established. Thus from (i) the ideal cycle $W_B = 0$:

$$COP = \frac{Q_E}{Q_D} \tag{3}$$

From the second Law is found the relation:

$$\frac{Q_D}{T_D} + \frac{Q_E}{T_E} = \frac{Q_A}{T_A} + \frac{Q_C}{T_C} \tag{4}$$

Substituting equations (2-72) e (2-76) into equation (2-75), and considering condition (iii) and (iv) $Q_C = Q_D$ the Carnot coefficient of Performance has the following expression:

$$COP_C^i = \frac{\dfrac{1}{T_A} - \dfrac{1}{T_D}}{\dfrac{1}{T_E} - \dfrac{1}{T_C}} \tag{5}$$

The above equation is the ideal internal performance of the absorption refrigeration system under study, where T_A, T_D, T_E, T_C are the absolute temperatures of absorption, desorption, evaporation and condensation respectively.

The external performance takes into consideration the changes of the state of the surrounding which include that part taking heat from the system and that part that transfer heat to the system. The expression for the external coefficient of performance is:

$$COP_C^e = \frac{\dfrac{1}{T_b} - \dfrac{1}{T_v}}{\dfrac{1}{T_g} - \dfrac{1}{T_b}} \tag{6}$$

Where T_b, T_v, T_g, are the absolute wet-bulb, heating vapor and ice temperatures respectively, as the system produce ice directly on the wall of evaporator, is driven by steam and the condenser and absorber are evaporative exchangers.

The internal relative coefficient of performance is expressed by:

$$\eta_r^i = \frac{COP}{COP_C^i} \tag{7}$$

and the external:

$$\eta_r^e = \frac{COP}{COP_C^e} \tag{8}$$

Exergetic method of evaluation. Second Law analysis. The efficiency of the system installation can be evaluated through the use of the availability or capacity of work (exergy) of the system a concept based upon the Second Law of thermodynamic. The exergetic method to determine the loss of work capacity or availability is an important qualitative analysis tool, which gives a clear picture of the main sources ofirreversibilities of the cycle.

In the exergetic method each element of the installation is considered as a thermodynamically independent system. The performance of each element is evaluated by comparing the available work of the fluids entering this element with the lost of this capacity due to the irreversibilities of the process that take place in the element under analysis.

For a quantitative evaluation of the thermodynamic perfection of any apparatus the concept of "exergy efficiency" has been suggested (Baehr, 1969), which is defined, when no work is done as:

$$\xi exg = \frac{\text{exergy outlet}}{\text{exergy inlet}} \quad (9)$$

and for a cycle

$$\xi c = \frac{\text{exergy of the cold produced}}{\text{exergy of the heat input}} \quad (10)$$

System component balance. For the evaluation of the system through the above performance analysis methods is essential the component balancing process in order to detail any technical problem in that component.

The capacity of each of the components can be defined in terms of certain variables. Some of these variables are internal to the system such as refrigerant temperatures in the evaporator and condenser. Others are external such as wet-bulb temperature.

Determination of the performance of the components operating together as a system involves the elimination of the internal variables of the system by the process of simultaneously solving a set of equations each of them expressing the performance of the component. This set of equations is solving with the help of experimental data take in the actual system. The component equations of the system are:

- Conservation of total mass:

$$\sum_i m_i = 0 \quad (11)$$

where m is the mass flux.

- Conservation of mass for each material species (refrigerant/absorbent)

$$\sum_i m_i X_i = 0 \quad (12)$$

where X is the solution concentration

- Energy balance:

$$\sum_i m_i h_i + Q_{unid} = \sum_o m_o h_o + W_{unid} \quad (13)$$

where h is the enthalpy of the fluid, Q_{unid} is the heat transferred at the unit and W_{unid} is the external mechanical work input to the component.

- Exergy balance:

$$\sum_i m_i e_i + \sum \left(\frac{T_f - T_0}{T_f} \right) Q_{unid} = \sum m_o e_o + W + I \tag{14}$$

where e is de exergy of the fluid, T_f the fluid temperature inside the component, T_0 the surrounding temperature and I the irreversibility generated.

- Irreversibility by entropy generation:

$$I = T_0 \left[\sum_i m_i s_i - \sum_o m_o s_o - \sum \frac{Q_{unid}}{T_f} \right] \tag{15}$$

2.3 Discussion of results

Table 1 shows the values of all the different experimental data points acquired on the absorption machine. Can be noted several nil points meaning that there is not sensor for those points for that are necessity a value estimated by means of the simulation process.

Table.1. Experimental data points

Point	Pressure (MPa)	Temp. (°C)	Conc.	Mass flux (Kg/h)
1	1,407	111,01	0,9212	
2	1,407	40,05	0,99971	73,86
3	1,297	28	0,99971	73,86
4	1,297	28	0,99971	73,86
5	1,297	13	0,99971	73,86
6	0,301	-9,0	0,99971	73,86
7	0,301	-8,5	0,99971	73,86
8	0,271	20	0,99971	73,86
9	0,271		0,38	727,82
10	0,271	39,5	0,38	727,82
11	0,271	39,5	0,38	727,82
12	1,407	39,5	0,38	727,82
13	1,407	39,5	0,38	727,82
14	1,407	50,02	0,38	727,82
15	1,407	50,02	0,38	727,82
16	1,407	92,03	0,38	727,82
17	1,407	-	0,38	727,82
18	1.407	111,01	0,31	653,95
19	1.407	105	0,31	653,95
20	1,407	55	0,31	653,95
21	1,407	41,5	0,31	653,95
22	0,252	127,7	-	96,06
23	0,252	127,7	-	96,06
24	0,101	22,5	-	172,8
25	0,101	0,00	-	172,8
26	0,101	22,5	-	180

Table 2. shows the total enthalpy, entropy and exergy of each experimental points. These values were calculated by means of AQUAM software due to Jordan (1992).

Table 2. Total properties

Point	H (kW)	S (KW/K)	Ex (kW)
1	-	-	-
2	30,3995	0,1015	7,4305
3	6,4024	0,0239	6,5495
4	6,4024	0,0239	6,5495
5	4,9380	0,0189	6,5686
6	4,9380	0,0208	6,3887
7	29,1333	0,1110	3,3178
8	30,8099	0,1171	3,1842
9	-	-	-
10	2,8951	0,1382	-10,3755
11	2,8951	0,1382	-10,3755
12	3,8190	0,1388	-9,6274
13	3,8190	0,1388	-9,6274
14	13,4283	0,1691	-9,0331
15	13,4283	0,1691	-9,0331
16	51,9502	0,2813	-3,9680
17	-	-	-
18	63,8382	0,2906	-2,1361
19	58,4413	0,2764	-3,3139
20	18,2234	0,1624	-9,5433
21	7,6625	0,1296	-10,3061
22	72,5066	0,1881	16,5402
23	14,5819	0,0430	1,6282
24	4,5366	0,0159	0,0021
25	-15,9979	-0,0586	1,6928
26	4,7257	0,0166	0,0022

Table 3 shows the energetic balance of the system. The little percent representing the electrical power consumption with respect to the cooling effect (less than 0.16%) can be seen on it. Data from both the generator group and the evaporator shows that the actual system coefficient of performance is COP$\cong$ 42 %.

Table 3. Energetic System Balance

Component	Energy input		Energy output	
	kW	%	kW	%
Generator group	57,92	67,70	6,42	7,63
Solution Pump	0,61	0,71		
Auxiliaries	2,65	3,09		
Condenser			23,99	28,53
Evaporator	24,16	28,24		
Absorber			35,57	42,3
Expansion valve.				
Solutions heat exchanger			1,6959	2,02
Ammonia sub-cooler	0,2122	0,25		
Week solution cooler			10,56	12,55
Others losses			5,286	6,28
Total	**85,5**	**100**	**84,08**	**100**

In table 4 the balance of exergy of the system can be observed. It reveals that the major irreversibilities are located at generator group. Szargut (1988) and Jordan (1985) obtained similar results. It is an expected result because de irreversible separation process that takes place in this component. As in the already mentioned paper the evaporator and the absorber are the others component with high degree of irreversibility. However, contrary to the others works the absorber in this system do not occupy third position, but the second with 20 % of de total irreversibilities.

Table 4. Exergetic System Balance

Component	Input Exergy		Irreversibilities			Util Exergy		Exergetic Efficiency
	(kW)	**(%)**	**Var. of Exergy (kW)**	**Entropy gener. (kW)**	**(%)**	**(kW)**	**(%)**	**(%)**
Generator group	15	85,2	5,15	3,96	31,77			66,29
Solution Pump	0,75	4,26	0,14		0,86			
Auxiliaries	1,85	10,51	0,278		1,71			
Condenser			0,881	0,882	5,43			
Evaporator			1,439	1,05	8,87	1,69	10,42	54,02
Absorber			3,25	4,00	20			
Expansion valve.			0,21	0,56	1,29			
Solutions heat exchanger			1,159	2.17	7,15			81,00
Ammonia sub-cooler			0,1136	0,32	0,7			14,29
Week solution cooler			0,7628	0,7856	4,7			
Others losses			1,15		7,0			
Total	**17,6**	**100**	**14,52**	**13,82**	**89,58**	**1,69**	**10,42**	

From equation (5) to (8) and from de above tables the following performance coefficients can be estimated:

COP = 0,42;

$COP_C^i = 1{,}31;$

$COP_C^e = 2{,}7;$

$\eta_r^i = 0.320$

$\eta_r^e = 0.16$

$\xi c = 0{,}28$

2.4 Conclusions

There are limited data available from absorption refrigeration systems employed to produce ice and less information about the system characteristics for scale icemaker absorption system, in which the growing ice is directly on the evaporator wall and use reverse cycle method for defrost provoking the transient operation of the system. This fact is a handicap against a proper comparison with systems of the same general arrangements as the one presented in this study.

The actual Coefficient of Performance attained (0,42) is slow when compared with typical values for simples water-ammonia absorption system already published in the specialized literature (≈ 0,55).

The relative internal efficiency is in the same order of magnitude when compared, on similar basis, with results quoted by several specialized publications. This relative poor approach to the theoretical or ideal equivalent system is typical on these installations.

The low relative external efficiency indicates that the actual COP is very low and may be due to a high temperature. Observing equations (5) and (6) and revising table 1 is noted the low ideal coefficient that can be expected with the high absorption temperature.

The cycle analysis from the Second Law standpoint shows by the overall balance of exergy losses that the major source of irreversibilities in the system are found in the generator (31 %) and in the absorber (20 %).

The exergetic efficiency of the whole system ($\xi c = 0,28$) shows, as it is expected, a substantial reduced value compared with the conventional COP. in use.

Acknowledgment

The authors would like express their grateful to Foundation of Support to Research of São Paulo State (FAPESP) for its partial support to this work.

REFERENCES

Baehr, H. D., Hofman, E. and Plank, R., 1969, Handbuch der Kaltetecnich, Vol. III, Mac Graw Hill.

Condes, F., 1978, “Sistema de absorción com dos etapas de evaporación”. Doctoral Thesis. Unpublished. ISPJAE, Cuba.

Jordan, D. P., 1985, Exergetic evaluation of aqua-ammonia absorption refrigeration system. Texas Tech University.

Jordan, D. P., 1992, AQUAM, Texas Tech University.

Szargut, J., 1988, Exergy Analysis of Thermal, Chemical and Metallurgical Processes, Hemisphere Publishing Corporation.

Shah, R. K., Subbarao, E. C. and Mashelkar, R. A., 1988, Heat Transfer Equipment Design, Hemisphere Publishing Corporation.

Silverio, R. J, Pratts, R. L., Figueiredo, J. R, and Pereira, T. V., Numerical simulation and analysis of a scale ice maker water-ammonia absorption system. In preparing.