23rd ABCM International Congress of Mechanical Engineering
December 6-11, 2015, Rio de Janeiro, RJ, Brazil

THERMODYNAMIC ANALYSIS AND DIMENSIONING OF A DOUBLE EFFECT LIBR-H_2O ABSORPTION REFRIGERATION SYSTEM

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Abstract. *In this paper, it is intended to carry out a thermodynamic analysis and dimensioning of a double effect LiBr-H_2O absorption refrigeration system with capacity of 15 TR, having as power supply solar energy collected by a radiant heat concentration reflector in a steam tube responsible for the transformation of radiation into thermal energy. With modeling and deduction of the thermodynamic equations for each cycle component (generators, condenser, evaporator, absorber, heat exchanger), it was generated an algorithm in the EES (Engineering Equation Solver) platform for the automation of calculus routines, allowing simulations of various sceneries by altering the process variables. The obtained results will be analyzed by tables and graphics in EES, identifying the best working conditions of the system, and then in a future an economic viability analysis can be implemented at zones with high incidence of solar radiation, such as Brazil's North and North-East regions, thus contributing to the crescent utilization of renewable energies and the country's sustainable development.*

Keywords: *Thermodynamics, Solar Energy, Refrigeration, Absorption*

1. INTRODUCTION

Brazil is currently passing through a number of challenges regarding its energetic matrix, where it is sought, on one side, diversification and growth of electric energy production to ensure its economic growth and, on the other side, the need of implementing long run sustainable actions, such as the utilization of solar and wind energies. Another pondering action is to take the initiative not only in the energy production, but also in the various consumer equipments, having as challenge finding alternative ways of more efficient, cleaner solutions, able to reuse the energy rejected by other industrial processes. In this context, the absorption refrigeration systems are inserted as a technically and economically viable strategies, since their initial cost would be diminished as the time pass, becoming more advantageous than the present steam compression refrigeration systems.

The present work shows the dimensioning of an absorption refrigeration system that uses the pair lithium bromide and water (LiBr-H_2O), with a capacity of 15 TR. The system is a simple-effect type and has as main input data the heat supplied to the generator through the water heated by a solar concentrator. The cycle thermodynamic analysis is based on the mass and energy conservation law, with definition of properties such as temperature, pressure, enthalpy, entropy and solution concentration in each point. With these properties for input and output, the heat transferred in each component can be determined.

As second analysis is the heat exchanger dimensioning, where finding the heat transfer area is based on the logarithmic mean temperature method, but in order to find the overall heat transfer coefficient as function of the component's geometry and the flow type, it was used adequate correlations. The applied correlations were from Dittus *et al* (1930) *apud* Incropera *et al* (2013). These correlations were used for determining the internal and external convection heat transfer coefficients. After defining the thermodynamic processes and the equations for the heat exchanger dimensioning, it was developed a computational code by using the EES (Engineering Equation Solver) platform, allowing the automate the calculus routine, allowing the simulation of various sceneries through the alteration of the system variables, obtaining the heat transfer area and consequently the number of tubes of each component.

2. DIMENSIONING SYSTEM

The simple effect LiBr-H_2O absorption refrigeration system consists in the absorption of the refrigerant in the low pressure line, and its dissolution in the high pressure line. The absorption cycle uses the following basic components: evaporator, absorber, recirculation pump, generators, condenser, throttling valves and expansion device (MOREIRA, 2004).

The absorbent fluid lithium bromide captures in the absorber the water steam refrigerant from the evaporator. The resulting solution is then pumped to the generator, where steam is split from the solution through transfer heat from the heated water by the solar concentrator. The compression of the liquid solution, which has less specific volume than steam, requires less power for a same pressure gain.

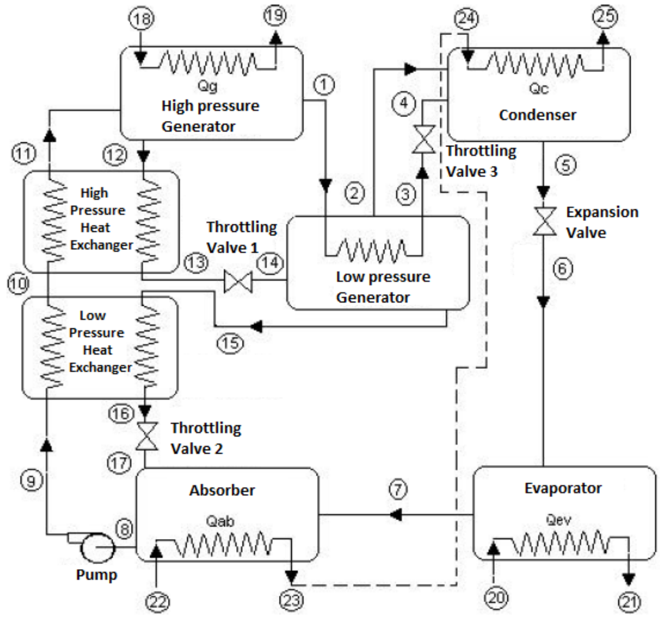

Figure 1. Absorption refrigeration of double effect system

The first law of thermodynamics applied to control volume allows, through the energy and mass balance, to find the rejected or absorbed heat rate in each component of the absorption refrigeration system.

$$\dot{Q}_{vc} + \sum_{in} \dot{m}.\left[h + \frac{V^2}{2} + g.z \right] = \frac{dE_{vc}}{dt} + \sum_{out} \dot{m}.\left[h + \frac{V^2}{2} + g.z \right] + \dot{W}_{vc} \tag{1}$$

where: $\dot{Q}_{vc}$ is the heat transfer rate for a control volume, $\dot{m}$ is the mass flow rate, h is the enthalpy, V is velocity, g is acceleration of gravity, Z is height and $\dot{W}_{vc}$ is the work rate for a control volume.

Because the system operates in steady state, $dE_{vc}/dt = 0$ and, because there is no work in the shaft, $\dot{W}_{vc} = 0$. Also, kinetic and potential energies changes are neglected. Thus, the first equation becomes:

$$\dot{Q}_{vc} + \sum_{in} \dot{m}.h - \sum_{out} \dot{m}.\mathrm{h} = 0 \tag{2}$$

After calculating the rejected or absorbed heat in each component, it is done the dimensioning of the heat exchangers by using the Logarithmic Mean Temperature Difference (LMTD) method, taking as reference the relation between the overall heat transfer rate (Q) and quantities such as the in and out temperatures, the overall heat transfer coefficient (U) and the heat transfer surface area (A) (*apud* Incropera *et al*, 2013).

After knowing the in and out temperatures of the hot and cold fluids, the LMDT can be determined applying the Equation (3).

$$LMTD = \frac{\Delta T_{\max} - \Delta T_{\min}}{\ln\left(\dfrac{\Delta T_{\max} - \Delta T_{\min}}{\Delta T_{\min}} \right)} \tag{3}$$

It must be determined the overall heat transfer coefficient (U) which is calculated as a function of the heat exchanger's internal and external convective coefficients.

$$\frac{1}{U.A}=\frac{1}{A_{ext}.h_{\text{ext}}}+\frac{1}{A_{int}.h_{\text{int}}}+\frac{R'_{ext}}{A_{ext}}+\frac{R'_{int}}{A_{int}}+\frac{\ln\left(D_{ext}/d_{\text{int}}\right)}{2\pi.k.l} \tag{4}$$

where: U is the overall heat transfer coefficient, A is the heat transfer surface area, h is the convective heat transfer coefficient, and int and ext stand for internal and external, respectively. All units are SI units.

By knowing LMTD and U, now it is just apply these values in the Equation 5 for obtaining the heat exchanger needed surface area.

$$\dot{Q}=U.A.LMTD \tag{5}$$

Equations from (1) to (5) are used for all absorption refrigeration cycle components. However, for calculating the internal and external convection coefficients it is necessary to be done a detailed analysis about the geometry and the flow type in each one of the heat exchangers.

2.1 High Pressure Generator

Component responsible for the heat transfer of the water heated by the solar concentrator to the LiBr-H_2O solution, since the heated water becomes steam, which is directed to the condenser. The steam generator consists of a cylindrical shell through which the solution circulates and a set of tubes disposed a triangular way through which heating water circulates.

MOREIRA (2004), consolidated the equations and the sequencing of its application to find the convection heat transfer coefficients, as shown in the following equations:

$$T_w=\frac{T_{18}+T_{19}}{2} \tag{6}$$

where: T_w is the surface temperature, calculated by an arithmetic mean of the inlet and outlet temperatures of the hot fluid.

$$T_{sol}=\frac{\dot{m}_1.T_1+\dot{m}_{11}.T_{11}+\dot{m}_{12}.T_{12}}{\dot{m}_1+\dot{m}_{11}+\dot{m}_{12}} \tag{7}$$

where: T_{sol} is the solution temperature, calculated by a weighted mean of the inlet and outlet temperatures of the hot fluid.

$$T_\infty=\frac{T_{sol}+T_{19}}{2} \tag{8}$$

where: T_∞ is the free-stream temperature, calculated by an arithmetic mean of the solution temperature and the hot fluid outlet temperature.

$$T_f=\frac{T_{sol}+T_w}{2} \tag{9}$$

where: T_f is the film-temperature, calculated by an arithmetic mean of the solution temperature and the surface temperature.

$$Ja=Cp_v.\frac{T_w-T_{sat}}{h_{lv}} \tag{10}$$

where: *Ja* is the Jacob Number, the ratio between sensible and latent energies absorbed in the liquid-vapor phase change. It is used to calculate the corrected enthalpy.

$$h'_{lv}=h_{lv}\left(1+0,68.Ja\right) \tag{11}$$

where: h'_{lv} is the liquid-vapor change phase corrected enthalpy.

For determining the convection heat transfer external coefficient in a horizontal cylinder, it is used the Equation (12).

$$Nu_{ext} = 0,62\left[\frac{g\left(\rho_l-\rho_v\right).h'_{lv}.D_{ext}^3}{v_v.k_v\left(T_w-T_\infty\right)}\right]^{3/4} = \frac{h_{ext}.D_{ext}}{k_v} \tag{12}$$

where: g is acceleration of gravity (m/s²), ρ_l is the solution specific mass (kg/m³), ρ_v is the vapor specific mass (kg/m³), h'_{lv} h is the corrected phase change enthalpy (kJ/kg) and k_v is the vapor thermal conductivity (kW/mK).

For determining the internal convection heat transfer coefficient in a set of tubes, the equations (13) and (14) are used.

$$Re_{\text{int}} = \frac{4.\dot{m}_{18}}{\pi.d_{\text{int}}.\mu.N_{tube}} \tag{13}$$

$$Nu = 0,023.Re_{\text{int}}^{4/5}.Pr_{water}^n = \frac{h_{\text{int}}.D_{\text{int}}}{k_{water}} \tag{14}$$

where: $n = 0,3$ for cooling of the fluid.

2.2 Low Pressure Generator

This heat exchanger has the same function of the high pressure generator. The vapor generated in the high pressure generator enters in the low pressure generator, passing inside the tubes. The solution that is outside the tubes is heated for the separation of the water vapor from absorber solution.

The same ebullition phenomenon that is observed in the high generator pressure happened in the low pressure generator. Then, the Eq. (12) was used for the determination of the outside convection heat transfer coefficient.

Most condensation processes encountered in refrigeration and air-conditioning applications involve condensation on the *inner surfaces* of horizontal or vertical tubes. Heat transfer analysis of condensation inside tubes is complicated by the fact that it is strongly influenced by the vapor velocity and the rate of liquid accumulation on the walls of the tubes. This case the liquid-vapor change phase corrected enthalpy can be determined applying the Equation (15).

$$h'_{lv} = h_{lv} + \frac{3}{8}Cp_l\left(T_{sat}-T_s\right) \tag{15}$$

2.3 Condenser

Component responsible for water vapor condensation, vapor from the generator flow will circulate in the cylindrical shell, and in the set of tubes it will be the cooling water.

The necessary equations are the same as the previously mentioned, from Eq. (6) to Eq. (11), but changing only the reference points and it is used the Eq. (15) in place of the Eq. (12) because it is a peculiar laminar condensation and the external convection coefficient will be found by the Dhir and Lienhard (1971) equation (*apud* INCROPERA & WITT, 2013).

$$h_{ext} = 0,729\left[\frac{g.\rho_l\left(\rho_l-\rho_v\right).h'_{lv}.k_l^3}{N_{tubes}.\mu_l.D_{\text{ext}}v_v.k_v\left(T_\infty-T_w\right)}\right]^{1/4} \tag{15}$$

where: $n = 0,4$ in the Eq. (14) in the condenser for heating of the internal fluid.

2.4 Evaporator

Component responsible for cooling the water from the environment to be refrigerated, the refrigeration liquid (system water) from the condenser is transformed in vapor when it exchanges heat with the water environment, whose temperature is lowered when it passes through the evaporator. The evaporator consists of a cylindrical shell through which the refrigerant liquid/vapor circulates, and a set of tubes disposed a triangular way through which the cooling water circulates.

The necessary equations are the same as the previously mentioned, from Eq. (6) to Eq. (11), but changing only the reference points and it is used Eq. (16) in place of Eq. (12), as a recommendation of Bourouni (1998).

$$\frac{\bar{h}.D_{ext}}{k_v} = 0.33\left(\frac{cp.\mu_v}{k_v}\right)^{1/3}.\left(\frac{D_{ext}.\rho_v.V_{\max}}{\mu_v}\right) \tag{16}$$

$$V_{\max} = \frac{S_T}{\left(S_T - D\right)}.V \tag{17}$$

$$V_{\max} = \frac{S_T}{2\left(S_D - D\right)}.V \tag{18}$$

where: $V_{\max}$ is the fluid maximum speed through the set of tubes, S_T is the vertical spacing among the centers of the tubes, S_D is the horizontal spacing among the centers of the tubes in the triangular configuration, D is the external diameter of the tubes.

2.5 Absober

In this component it is processed the absorption of the water vapor that comes from the evaporator by the LiBr solution and water present in the absorber's shell. Absorber is continuously refrigerated by the water that runs within it. The absorber consists of a double-pass shell through which the solution flows and a set of tubes disposed a triangular way through which the cooling water circulates.

The necessary equations are the same as the previously mentioned, from Eq. (6) to Eq. (11), but changing only the reference points and it is used Eq. (19) in place of Eq. (12), as a recommendation of Deng and Ma (1999), who experimentally determined the external convective coefficient, by deducing the correlation showed by the following equation:

$$Nu_{ext} = \frac{h_{ext}.D_{ext}}{k_{fluid}} = 10^{(0,75-0,087.C_w)}.Re_{ext}^{0,8}.Pr^{1,1} \tag{19}$$

$$C_w = 100\% - C_S \tag{20}$$

$$Re_{ext} = \frac{\rho.V_{\max}.D_{ext}}{\mu} \tag{21}$$

where: C_w is the water percentage in the LiBr solution, C_S is the LiBr concentration in the solution, n = 0,4 when using Eq. (14) for the absorber, since there is an internal fluid heating, $V_{\max}$ is determined by Eq. (18).

2.6 Intermediary Exchanger

In order to exchange heat towards the back flow of the rich solution from the generator to the poor solution that is headed to the generator, it settled the intermediate heat exchanger that will preheat the LiBr-H_2O solution. The intermediate heat exchanger differs from other system components because it has configuration of two concentric tubes with countercurrent flows. In the inner tube the hot solution flows while in the outer tube it is the cold solution that flows.

The necessary equations are the same as the previously mentioned, from Eq. (6) to Eq. (11), but changing only the reference points and it is used Eq. (21) in place of Eq. (13) to calculate Reynolds Number in the annular space. Eq. (14) is valid to calculating the convective coefficient in the two heat exchanger tubes.

$$Re_{ext} = \frac{4.\dot{m}_2}{\pi\left(D_{ext} - d_{\text{int}}\right)} \tag{22}$$

3. RESULTS AND DISCUSSION

Tab. (1) shows the result of the thermodynamic analysis based on mass and energy balance, as the second law of thermodynamics which through the development of computational algorithm in EES platform it was possible to facilitate the calculations, thus allowing the verification of various scenarios. In this table it can be observed the properties point to point, and they were used to determine the required heat transfer rate in each component of the absorption refrigeration system according to Tab. (2), resulting in the final calculation for dimensioning the heat exchangers Tab. (3).

Table 1. Parameter for the Double-effect absorption refrigeration system design

Point	P(kPa)	T (ºC)	h (kJ/kg)	m (kg/s)	X (solution concentration)
1	77,52	80,59	2747	0,00679	0
2	6,28	34,27	2651	0,00707	0
3	6,28	70,86	388	0,00679	0
4	6,28	90,64	388	0,00679	0
5	6,28	54,06	154,9	0,1386	0
6	0,87	54,06	154,9	0,1386	0
7	0,87	80,59	2510	0,1386	0
8	0,87	37	70,25	0,0944	0,549
9	77,52	5	70,3	0,0944	0,549
10	77,52	5	113	0,0944	0,549
11	77,52	12	191,4	0,0944	0,549
12	77,52	90,64	302,1	0,0876	0,57
13	77,52	86,54	226,8	0,0876	0,57
14	6,28	52,06	226,8	0,0876	0,57
15	6,28	64,54	213,1	0,0805	0,08058
16	6,28	54	196,4	0,0805	0,08058
17	0,87	48,68	196,4	0,0805	0,08058
18	-	95	290,6	0,1201	-
19	-	85	183,3	0,1201	-
20	-	12	50,24	1,559	-
21	-	7	29,31	1,559	-
22	-	29,5	123,5	2,745	-
23	-	32,5	135,9	2,745	-
24	-	32,5	135,9	2,745	-
25	-	35	146,5	2,745	-

Table 2. Heat transfer rate in the components of the system.

Component	**Heat Transfer Rate (kW)**
High Pressure Generator	66,45
Low Pressure Generator	44,55
Condenser	55,91
Evaporator	52,76
Absorber	66,21
Intermediary Heat Exchanger (high pressure)	9,11
Intermediary Heat Exchanger (low pressure)	8,55

Table 3. Dimensions of the heat exchangers.

Component	**Length (m)**	**Diameter (m)**	**Diameter (inches)**	**Heat transfer area (m²)**	**Number of tubes**
High Pressure Generator	4	0,0254	1	9,53	30
Low Pressure Generator	4	0,0254	1	6,35	20
Condenser	4	0,0254	1	5,88	19
Evaporator	4	0,0254	1	5,60	18
Absorber	4	0,0254	1	10,44	33
Internal tube of the Intermediary heat exchanger (high and low pressure)	*3,60	0,1016	4	1,72	1
External tube of the Intermediary heat exchanger (high and low pressure)	*3,60	0,2032	8	2,28	1

* The real length is 2.70 m. However, because of concentric tubes heat exchanger construction details, it was adopted the length of the biggest tube.

4. CONCLUSIONS

Thermodynamic modeling using the EES (Engineering Equation Solver) system allows simulating a variety of scenarios, by changing the inputs, thus facilitating the necessary calculations and allowing an ideal solution to be found. When dimensioning the heat exchanger, it was adopted iron pipes with thermal conductivity k = 75 (kW/mK) whose dimensions are within the industrial standards, resulting in a lower cost in comparison to other materials such as copper and aluminum that, if used, they would significantly reduce the area required for exchange due to their better thermal conductivity than iron. This would significantly decrease the amount of tubes. However, it would impact a lot in the manufacturing cost of the heat exchangers.

5. REFERENCES

Çengel, Y. A.; Boles, M. A. *Thermodynamics*. 7th ed. McGraw-Hill, 2013. 740p.

Dittus, F. W. and Boelter, L. M. K., 1930 – University of California, Berkeley, Publications on Engineering, vol.2, p. 443.

Deng, S. M. and Ma, W. B., 1999 – "*Experimental Studies on the Characteristics of an Absorber Using LiBr/H_2O Solution as Working Fluid*", International Journal of Refrigeration, pp 293- 301.

Incropera, F. P.; Dewitt, D. P.; Bergman, T. L. and Lavine, A. S., 2011 – "*Fundamentals of Heat and Mass Transfer*", John Wiley. 7th ed.

Moreira, H. L, 2004 - "*Thermoeconomic Analysis of an Absorption Refrigeration System with the Water-Lithium Bromide Pair*", Doctorate degree thesis, CPGEM/CT/UFPB.

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