THEORETICAL – EXPERIMENTAL STUDY SYSTEM FOR ABSORPTION REFRIGERATION DOUBLE EFFECT IN SERIES USING PAR WATER / LITHIUM BROMIDE.

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Abstract: This paper describes a theoretical-experimental study of an absorption refrigeration machine of double effect in series using the pair water lithium bromide using the waste heat from the exhaust fumes of an internal combustion engine as an energy source for the steam generator steam high. This study aims to describe the constructive aspects of a pilot unit with cooling capacity of around 0.5 TR, emphasizing the discussions regarding accuracy, errors, problems found, identified problems and solutions found during the process of construction, testing, operation, data acquisition and evaluation of the pilot plant. It also made a comparison of experimental data with the results of a thermodynamic model based on the concepts of energy and exergy.

Keywords: Absorption refrigeration; double effect series, construction of the pilot plant; theoretical-experimental study.

1. INTRODUCTION

Something that has piqued the interest of many researchers is the cogeneration process that seeks to make the most of the waste heat from industrial equipment that are released directly into the environment such as the exhaust fumes of internal combustion engines, gas fireplaces and boilers ovens, among others.

This research is being developed at the Federal University of Paraíba by postgraduate students of mechanical engineering in thermo-fluid area, with respect to the cogeneration process and construction, analysis and design of absorption refrigeration machines steam.

The present study aims at theoretical- experimental study on an absorption refrigeration machine of double effect in the series that uses water lithium bromide pair, basing contributions to the development and optimization of components and the machine itself, giving thus continuation the research initiated by Varani (2001) who made energetic and exergetic analysis of systems of simple effect using Fortran platform for simulation, Moreira (2004) did thermoeconomic analysis systems for single and double acting in series, Santos (2005) did exergy analysis in systems using multiple effects of ESS platform, among others.

2. METHODOLOGY

Thermodynamic analysis: a thermodynamic analysis will be developed for each component of the unit to be built as well as for the refrigeration cycle vapor absorption effect in double series. Making the use of the first law of thermodynamics we obtain the fluxes of heat and mass, coefficient of performance and effectiveness of each component of systems for pre- conditions down to the steam generator high. From the second law of the terrmodynamics to evaluate the irreversibility, the ration efficiency and the degree of thermodynamic perfection.

Analysis of heat transfer and design: through the analysis of heat transfer, there will be sizing each component in order to determine if data regarding to the area of heat exchange and length and diameter of pipes required for the exchange of heat required to each component based on thermodynamic analysis.

Construction of the pilot plant: In this stage, the engine cooling vapor absorption of double effect in the series will be built. Therefore, it is necessary to monitor the proceedings involving the manufacture of machinery, testing of sealing of heat exchangers, check points for the installation of the measuring apparatus, instrumentation, machine, test machine operation and adjustments.

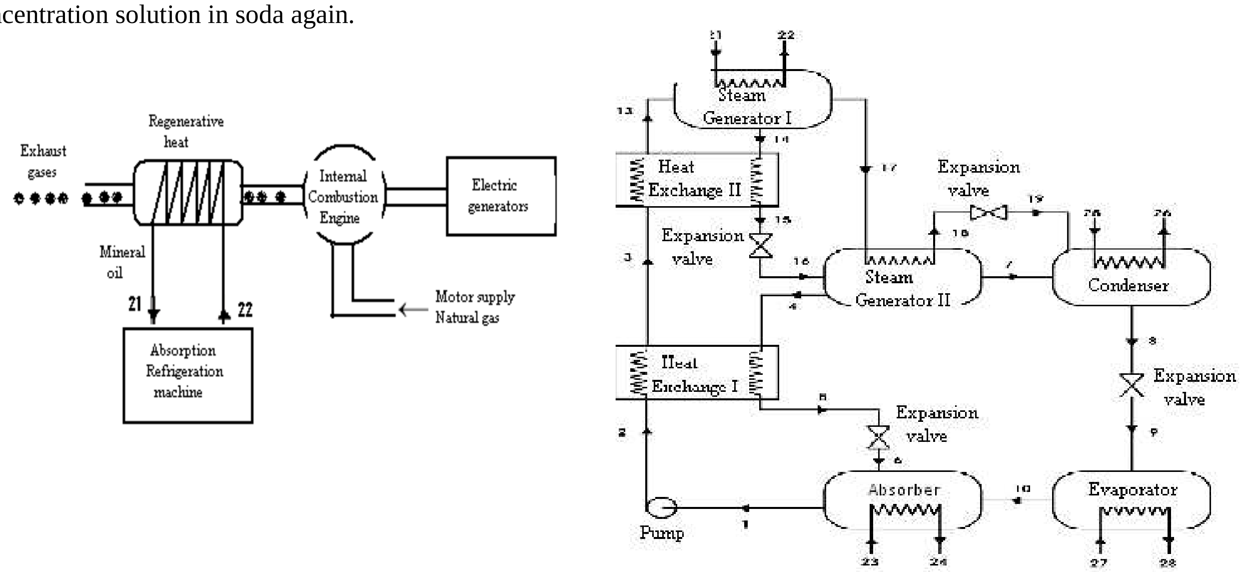
Instrumentation and acquisition of the data acquisition: it will be selected and installed pressure transducers, thermocouples, flow meter, variable frequency drive and digital indicator appropriate to need.

With the machine running and manipulated appropriately, it will be made to the appropriate values for the readings to be measured at various times.

Theoretical-experimental study: having the data obtained experimentally, it will be a theoretical support based on laws of thermodynamics and the equations of mass and heat transfer in order to evaluate the performance of various components and to reveal new paths to be followed in order to improve the pilot plant.

3. OPERATING PRINCIPLE

Figures1a and 1b illustrate, respectively, the cogeneration system and the machine cooled by absorption of double effect in the series that will be used in our comparative study. The machine is coupled to a heat regenerator, which in turn will receive the exhaust fumes of an internal combustion engine at high temperature and mineral oil at a lower temperature. The gases provide heat to the mineral oil and this, in turn, will be sent to the steam generator I (points 21 and 22) which will transfer heat to the solution water / lithium bromide. Before reaching the generator, the solution is preheated by passing twice heat exchangers I and II. Upon reaching the steam generator I, part of the water - which in this case is the refreshment and which is contained in the solution - is vaporized by receiving heat from the oil (points 21 and 22) and forwards it to the steam generator II (point 17). The solution, whose concentration is in average level refreshment, goes to the heat exchanger II (point 14) transferring heat to the solution that will go to the steam generator I. Soon after, will undergo an expansion device (points 15 and 16) to reduce their level pressure and enters the steam generator II. Mean while, the water vapor produced in the steam generator I is sent to the steam generator II (point 17) in an independent circuit. As the water vapor is at a temperature higher than the solution, that solution will yield the heat causing another amount of water still contained in the solution is vaporized. The vapor from the generator I, after the solution heat transfer, will undergo an expansion device (points 18 and 19) so that its pressure is reduced also the average level and then enters the condenser along with the vapor from the generator II. The solution is left in the steam generator II(solution with low concentration of refreshment), goes to the heat exchanger where I will preheat the refreshment-rich solution that will go to the steam generator and I then suffer a reduction of at low pressure while passing through an expansion device (point 5 and 6) reaching the absorber. The coolant, in turn, will condense in the condenser (19 points, 7 and 8), pass through the expansion device (points 8 and 9) also reducing their pressure to the low level and reach the evaporator. It's just that the refrigerant in the evaporator will absorb heat from the environment to be refreshed (points 27 and 28), that because the environment is hotter than the coolant. It then continues to the absorber, where they will be absorbed by the solution with low concentration of refrigerant is now becoming a high concentration solution in soda again.



a) Diagram of the cogeneration system b) Diagram of the absorption machine of double effect in series

Figure 1: Diagrams: a) cogeneration system; b) absorption machine of double effect in series

4. ANALYSIS OF HEAT TRANSFER AND DESIGN

The analysis of heat transfer takes into account the constructive aspects of each component of the unit. To make the design of various heat exchangers, we used the method of the Logarithmic Mean Temperature Difference - DTML. In this method, the rate of heat transfer between the fluids that circulate through the control volume is given by:

$$Q = A_t . U_m . \Delta T_{ml} \quad (1)$$

For the sizing of heat exchangers, used the following considerations: heat loss to the neighborhood negligible; change in kinetic and potential energy negligible; constant properties; factor negligible deposition in the tube; fully developed flow conditions; fluid properties obtained from the UES; all work in counter flow heat exchangers.

If it is necessary to make any alteration, addition or removal of the considerations mentioned in any particular case will be clarified in sizing the appropriate time. Moreover, the equations of heat transfer were taken from (Incropera et al, 2008), except those reported during the restriction equations.

4.1 SIZING OF CONDENSER

4.1.1 Internal flow

The fluid flowing in the inner tubes is water whose properties were obtained from the ESS based on the arithmetic average temperature compared to temperatures of incoming and outgoing water in the pipes.

- **Calculation of the de Reynolds number**

The Reynolds number is calculated for the fluid flow in a single tube and is given by:

$$\mathrm{Re}_{(\mathrm{int})} = \frac{4.\left(\dfrac{\dot{m}_{(H_2O)}}{N_T}\right)}{\pi . d_{(\mathrm{int})} . \mu_{(H_2O)}} \quad (2)$$

- **Calculation of the Nusselt number**

The Nusselt number in turbulent flow inside the flat tube can be determined from the correlations for the convection flow in circular tube. Re found to be valid using the equation:

$$Nu_{(\text{int})} = 0{,}023.\,\text{Re}_{(\text{int})}^{0,8}.pr^{0,3} \tag{3}$$

Prandlt being the number given by:

$$pr = \frac{Cp_{(H_2O)}.\mu_{(H_2O)}}{k_{(H_2O)}} \tag{4}$$

- **Calculating the convection coefficient**

Known Nusselt number, determine the internal convection coefficient from the equation:

$$h_{(\text{int})} = \frac{Nu_{(\text{int})}.k_{(H_2O)}}{d_{(\text{int})}} \tag{5}$$

4.1.2 External flow

The fluid that passes through the inner tubes and the shell of the condenser is water vapor which must be condensed.

- **Calculating the convection coefficient**

Whereas the process of condensation occurs in film form, the convection coefficient can be calculated from the equation:

$$h_{(ext)} = 0{,}729\left[\frac{g.\rho_l.(\rho_l-\rho_v).h_{lv}^{'}.k_l^3}{\mu_l.(T_{sat}-T_{\sup}).N_{T,F}.D_T}\right]^{\frac{1}{4}} \tag{6a}$$

being that (h'lv) the corrected enthalpy of phase change which is determined by the number of Jacob which is the ratio between the energy absorbed and sensitive latent energy absorbed in the change of liquid-vapor phase, ie

$$Ja = Cp_v\,\frac{(T_{\sup}-T_{sat})}{h_{lv}} \tag{6b}$$

$$h_{lv}^{'} = h_{lv}(1+0{,}68.Ja) \tag{6c}$$

The properties of saturated liquid contained in Eq(6a) were obtained taking into account the temperature of the film which can be determined by the equation

$$T_{(filme)} = \frac{T_{(\sup)}+T_{(\inf)}}{2} \tag{7}$$

The saturation temperature T_{sat} is equal to the temperature of condensate leaving the condenser, or T_8. However, as the condenser has two steam inlet piping is necessary that surface temperatures T_{sup} in the infinite T_{inf} are determined as follows:

Was first estimated for an average temperature steam into the condenser building on conservation of energy and considering the specific heat constant, resulting,

$$T_{Sat(7,19)} = \frac{\dot{m}_7\,T_7+\dot{m}_{19}\,T_{19}}{\dot{m}_7+\dot{m}_{19}} \tag{8}$$

Then estimated the temperature at a point distant from the surface of tubes called here the temperature at infinity, given by:

$$T_{(\inf)} = \frac{T_{(7,19)}+T_8}{2} \text{ Temperature in the infinite} \tag{9}$$

The surface temperature of the tubes was taken as the arithmetic mean of inlet and outlet temperatures of cooling water inside the tubes, given by;

$$T_{(\sup)} = \frac{T_{25}+T_{26}}{2} \text{ Temperature in the surface} \tag{10}$$

- **Calculation of the global coefficient of heat transfer**

To calculate the overall coefficient was done using the equation:

$$U_m = \frac{1}{h_{(\text{int}.)}.A_{(\text{int},lat)}} + \frac{\ln\left(D_h/d_{(\text{int})}\right)}{2.\pi.L.K_{aço}} + \frac{1}{h_{(ext.)}.A_{(ext,lat).N_T}} \tag{11}$$

- **Calculation the ΔT_{ml}**

The calculation of logarithmic mean of temperature differences is based on the heat exchangers in counter flow. Thus, the equations are used:

$$\Delta T_{ml} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} = \frac{T_{26} - T_{25}}{\ln\left(\frac{T_{26} - T_8}{T_{25} - T_8}\right)} \quad (12)$$

$$\Delta T_1 = T_{q,e} - T_{fr,s} \quad \text{e} \quad \Delta T_2 = T_{q,s} - T_{fr,e} \quad (13)$$

- **Calculating the total area of heat transfer**

The total area of heat exchange is calculated by the equation:

$$A_t = \pi . D_T . L_T . N_T \quad (14)$$

Substituting Eq (14) in Eq (1), is there relationship between tube length and number of pipes that is given by:

$$L_T . N_T = \frac{Q}{\pi . D_T . U_m . \Delta T_{ml}} \quad (15)$$

4.2 SIZING OF EVAPORATOR

Evaporator by circulating two fluid flows, and the water circulates inside the tubes which should be cooled, while the region between the hull and tubes circulate the coolant, in this case is also water. The detail is that the refrigerant will enter the evaporator in the liquid phase and out as well occurring saturated vapor phase change. For the sizing of the evaporator, we adopted the same considerations mentioned above for the condenser, save correction into consideration with respect to condensation on film, since it is now one of the boiling process on film.

4.2.1 Internal flow

Inside the tubes contained water circulates in the evaporator which should be chilled. To obtain the properties of water was determined using the arithmetic average of the temperatures of inlet and outlet water in the evaporator, since the water temperature varies during its passage through the tubes.

- **Calculation of the Reynolds number**

The Reynolds number is given by equation (2).

- **Calculation of the Nusselt number**

For Re found to be valid using the equation:

$$Nu = 0{,}023 . \mathrm{Re}_{(int)}^{0,8} . \mathrm{Pr}^{0,4} \quad (16)$$

Prandlt being the number given by equation (4).

- **Calculation of the convection coefficient**

Known Nusselt number, one can determine the internal convection coefficient by Eq. (5):

4.2.2 External flow

The fluid that passes through the inner tubes and the shell of the evaporator is also water, however it is intended that it leaves the evaporator as saturated vapor.

- **Calculating the convection coefficient**

The convection coefficient can be calculated from Eq.(6a) so the properties of saturated liquid were obtained taking into account the temperature of the film which can be determined by application of Eq.(7), resulting in:

$$T_{sup} = \frac{T_{27} + T_{28}}{2} \qquad e \qquad T_{sat} = T_{10} \quad (17)$$

- **Calculation of the global coefficient of heat transfer**

To calculate the overall coefficient was made use of Eq. (11):

- **Calculation of the ΔT_{ml}**

The calculation of logarithmic mean of temperature differences is based on the heat exchangers in counter flow. We use the Eq (12) and (13), resulting in:

$$\Delta T_{ml} = \frac{(T_{28} + T_9) - (T_{27} + T_{10})}{\ln\left(\frac{T_{28} - T_{10}}{T_{27} - T_9}\right)} \quad (18)$$

- **Calculating the total area of heat transfer**

The total area of heat exchange tube and the length and number of tubes is determined with Eq. (14) and (15).

4.3 SIZING OF THE HEAT EXCHANGERS INTERMEDIARY

The fluids that will circulate the heat exchangers is the intermediate solution of water / lithium bromide in different concentrations which will exchange heat with each other without phase change. The solution with weak passes through the annular region on both exchangers, while the solution with strong and medium passes through the inner tubes in the exchanger of low and high, respectively. The values of the properties of the solution water /lithium bromide were obtained from the UES.

4.3.1 Internal flow

- **Calculation of the Reynolds number**

The Reynolds number for a single tube is determined by the equation:

$$Re_{(int)} = \frac{4.\left(\dot{m}_{(LiBr)} / N_T \right)}{\pi . d_{(int)} . \mu_{(LiBr)}} \quad (19)$$

Found for Reynolds, Nusselt becomes equal to Nu= 4.36.

- **Cálculo de coeficiente de convecção**

Knowing the Nusselt number, the internal convection coefficient can be determined by:

$$Nu = \frac{h_{(int)} . d_{(int)}}{k_{(LiBr)}} = 4{,}36 \quad (20)$$

4.3.2 Anular flow

- **Calculation of the Reynolds number**

$$Re_{(an)} = \frac{4.\dot{m}_{(LiBr)}}{\pi . D_h . \mu_{(LiBr)}} \quad (21)$$

Found for Reynolds, Nusselt Nu= 4.36 is taken.

- **Calculating the convection coefficient**

Knowing the value of the Nusselt number, the external convection coefficient is determined from the equation:

$$Nu = \frac{h_{(ext)} . D_h}{k_{(LiBr)}} \quad (22)$$

- **Calculation of the global coefficient**

The calculation of the overall coefficient is given by equation (11).

- **Calculation of the ΔT_{ml}**

Exchangers as intermediaries also operate incounter current, the mean log of temperature differences is determined by Eq (12) and (13), resulting in:

$$\Delta T_{ml} = \frac{(T_{15} + T_{13}) - (T_3 + T_{14})}{\ln\left(\frac{T_{15} - T_3}{T_{14} - T_{13}} \right)} \quad \text{T.C high} \quad (23a)$$

$$\Delta T_{ml} = \frac{(T_5 + T_3) - (T_2 + T_4)}{\ln\left(\frac{T_5 - T_2}{T_4 - T_3} \right)} \quad \text{T.C low} \quad (23b)$$

- **Calculating the total area of heat transfer**

The heat transfer $\dot{Q}$ was determined by:

$$\dot{Q} = \dot{m}_3 \left(h_{13} - h_3 \right) \quad \text{exchanger high} \quad (24a)$$

$$\dot{Q} = \dot{m}_3 \left(h_{13} - h_2 \right) \quad \text{exchanger low} \quad (24b)$$

that applied in eq. (15) allows to know the area of heat exchange.

4.4 SIZING DO ABSORBING

The absorber is the device responsible for promoting the absorption of water vapor coming from the evaporator with a solution of lithium bromide concentration from the strong low pressure steam generator, causing the concentration of the solution returns to low level and thereby complete the absorption refrigeration cycle. In this process of absorption occurs releasing heat, a fact that will hinder the absorption of steam if not removed from the control volume. Thus, it does need a cooling system which in this case is accomplished by the passage of water through the tubes that make up the absorber. The heat must be removed is determined by the equation:

$$\dot{Q}_{abs} = \dot{m}_{23}\, cp_{(H2O)} \left(T_{24} - T_{23} \right) = UA_t \Delta T_{ml} \quad (25)$$

4.4.1 Internal Flow

The Reynolds number for a tube is given by equation (2).

- **Calculation of the Nusselt number:**

For Re found to be valid the use of Eq (16) and (4):

- **Calculation of the convection coefficient**

The internal convection coefficient is given by equation (5).

4.4.2 External Flow

The refreshment coming from the evaporator, preferably in the vapor phase, must be absorbed by the strong solution concentration coming from the steam generator low. In this case the steam will condense forming a weak solution concentration. Both streams will pass through the tubes and the shell of the absorber.

- **Calculation of the convection coefficient**

To determine the external convection coefficient makes use of Equation (6a). To determine the temperature of the film, since it has one inlet and one strong solution flowing water vapor, it is calculated as follows: First is determined to an average temperature steam into the solution concentration and strong like a single entry. This was done through the conservation of energy which resulted in the equation:

$$T_{(6,10)} = \frac{\dot{m}_6 .T_6 + \dot{m}_{10} .T_{10}}{\dot{m}_1} \qquad (26)$$

Then , we calculate the temperature at infinity, given by:

$$T_{(\inf)} = \frac{T_{(6,10)} + T_1}{2} \quad \text{temperature in the infinite} \qquad (27)$$

The surface temperature of the tubes was taken as the arithmetic mean of inlet and outlet temperatures of cooling water that circulates through the tubes, given by:

$$T_{(\sup)} = \frac{T_{23} + T_{24}}{2} \quad \text{temperature in the surface} \qquad (28)$$

The temperature of film used to determine the properties of the solution and vapor entering the absorber is given by equation (7), while the average concentration is calculated by the equation:

$$x_m = \frac{\dot{m}_6 \ x_6 + \dot{m}_{10} \ x_{10}}{\dot{m}1} \qquad (29)$$

- **Calculation of the global coefficient of heat transfer**

To calculate the overall coefficient is made use of Equation (11).

- **Cálculation of the ΔT_{ml}**

Applying the equation (12) and (13) is the logarithmic mean temperature difference of which is given by:

$$\Delta T_{ml} = \frac{(T_1 - T_{24}) - (T_{(6,10)} - T_{23})}{\ln\left(\frac{(T_1 - T_{24})}{(T_{(6,10)} - T_{23})}\right)} \qquad (30)$$

- **Calculating the total area of heat transfer**

The total area of heat exchange tubes corresponds to the surface and the relationship between tube length and number of tubes is determined by combining Equation (14), (15) and (25).

4.5 SIZING OF STEAM GENERATOR OF HIGH

It is also considered that the process of boiling water is carried out in movie form. The heat supplied to the operation of the steam generator will come from the high mineral oil will circulate through the tubes contained in the generator. Since there is no change in the oil phase, the heat that this will transfer to the solution of water / lithium bromide is estimated by using the thermodynamic equation,

$$\dot{Q}_{gI} = \dot{Q}_{óleo} = \dot{m}_{óleo} \ cp_{m,óleo} \Delta T_{óleo} = UA_t \Delta T_{ml} \qquad (31)$$

To calculate the average specific heat, we use data provided by the oil manufacturer. Determine the equation of the line through the chart that relates the specific heat versus temperature and, therefore, the average specific heat is calculated as the arithmetic average of specific heats oil in the input and output of the generator which was calculated by,

$$cp_{m,óleo} = 0{,}001675(T_{21} + T_{22}) + 1{,}84175 \qquad (32)$$

How is neglecting heat losses to the environment because the generator is insulated, it is considered that all the heat supplied by the oil will be absorbed by the solution of water / lithium bromide concentration weak passing through the tubes and the hull of the generator.

4.5.1 Internal Flow

To determine the properties of mineral oil was used to operating temperature would be given by the arithmetic mean of inlet and outlet temperatures in the steam generator.

- **Calculation of Reynolds number**

The Reynolds number for a tube is given by:

$$\mathrm{Re}_{(\mathrm{int})} = \frac{4\left(\dot{m}_{(óleo)} \big/ N_T\right)}{\pi . d_{(\mathrm{int})} . \mu_{(óleo)}} \tag{33}$$

- **Calculation of the Nusselt number**

Found for Re, Nusselt is given by equation (16) being the number of Prandlt given by:

$$\mathrm{Pr} = \frac{cp_{(m,óleo)} . \mu_{(óleo)}}{k_{(óleo)}} \tag{34}$$

- **Calculation of the convection coefficient**

Known Nusselt number, the internal convection coefficient is determined by:

$$h_{(\mathrm{int})} = \frac{Nu_{(\mathrm{int})} . k_{(óleo)}}{d_{(\mathrm{int})}} \tag{35}$$

4.5.2 External Flow

The steam generator will receive a high concentration solution will be weak and output streams and water vapor with concentration solution out through separate pipes. In this case, it is necessary to identify an optimum temperature so as to obtain the solution properties of steam and considering that there are two exits instead of just one. This was done through the conservation of energy which resulted in the equation:

$$T_{(14,17)} = \frac{\dot{m}_{14} . T_{14} + \dot{m}_{17} . T_{17}}{\dot{m}_{13}} \tag{36}$$

Then determines the temperature at a point distant from the surface of the tubes, ie, the temperature at infinity is given by:

$$T_{(\mathrm{inf})} = \frac{T_{(14,17)} + T_{13}}{2} \tag{37}$$

The surface temperature of the tubes was taken as the arithmetic mean of inlet and outlet temperatures of the mineral oil inside the tubes, given by:

$$T_{(\mathrm{sup})} = \frac{T_{21} + T_{22}}{2} \tag{38}$$

The temperature of film used to determine the properties of the solution and vapor leaving the generator was defined according to Eq (7). The average concentration is calculated by the equation:

$$x_m = \frac{\dot{m}_{14}\ x_{14} + \dot{m}_{17}\ x_{17}}{\dot{m}_{13}} \tag{39}$$

In possession of the temperature of the film and the mean concentration and using the EES, met the necessary properties in order to give continuity to the other calculations involved in sizing.

- **Calculation of the convection coefficient**

The calculation of the external convection coefficient in the boiling process on film for horizontally placed cylinder was developed based on equation (Bejan, 1948, p. 429),

$$\frac{h_{(ext)} . D_T}{k_v} = 0{,}62\left[\frac{D_T^3 . g . (\rho_l - \rho_v)\, h'_{lv}}{k_v . \nu_v \left(T_{(filme)} - T_{sat}\right)}\right]^{\frac{1}{4}} \tag{40}$$

- **Calculation of the global coefficient of heat transfer:**

To calculate the overall coefficient was made use of Equation (11):

- **Cálculation of the ΔT_{ml}:**

The calculation of logarithmic mean of temperature differences is based on the heat exchangers in counter flow resulting in:

$$\Delta T_{ml} = \frac{(T_{21} - T_{13}) - (T_{22} - T_{(14,17)})}{\ln\left(\frac{(T_{21} - T_{13})}{(T_{22} - T_{(14,17)})}\right)} \tag{41}$$

- **Calculating the total area of heat transfer:**

The total area of heat exchange tubes corresponds to the surface and the relationship between tube length and number of tubes is determined by combining Equation (14), (15) and (31).

4.6 DESIGN OF THE GENERATOR LOW

The heat supplied to the operation of the steam generator is derived from the low water vapor coming from the generator high. It is considered that all the heat supplied by steam will be absorbed by the solution of water / lithium bromide

$$\dot{Q}_{gII} = \dot{m}_{17}(h_{17} - h_{18}) = UA_t \Delta T_{ml} \tag{42}$$

4.6.1 Internal Flow

The water vapor that passes through the tubes will transfer heat to the solution that lies between the tubes and the shell of the generator. There is a tendency, therefore, what portion of vapor under goes a phase change during the run off. At low speeds the flow of steam where the equations are valid: (Cengel, 2009, p.591)

$$\mathrm{Re}_{v,e} = \left(\frac{\rho_v . u_{(m,v)} . D_T}{\mu_v} \right) < 35000 \tag{43}$$

$$\overline{h_{(\mathrm{int})}} = 0{,}555 \left[\frac{k_l^3 . g . (\rho_l - \rho_v) h'_{lv} . \rho_l}{\mu_l . (T_{(sat)} - T_{(\mathrm{sup})}) . D_T} \right]^{\frac{1}{4}} \tag{44}$$

$$h'_{lv} \equiv h_{lv} + \frac{3}{8} . cp_l . (T_{sat} - T_{\mathrm{sup}}) \tag{45}$$

Since the properties in these equations are evaluated based on the temperature of the film and is prized in saturation temperature.

4.6.2 External Flow

The procedure for the design should be similar to that developed earlier for the high generator. The temperature of the flow generator outlet considering, once again, the steam and the low concentration solution exits through a single pipe is given by;

$$T_{(4,7)} = \frac{\dot{m}_4 . T_4 + \dot{m}_7 . T_7}{\dot{m}_{16}} \tag{46}$$

The temperature at a point distant from the surface of the tubes which is given by;

$$T_{(\inf)} = \frac{T_{(4,7)} + T_{16}}{2} \tag{47}$$

The surface temperature of the tubes was considered as the arithmetic average of the temperatures of inlet and outlet steam from the generator which high pass inside the tubes, given by;

$$T_{(\mathrm{sup})} = \frac{T_{17} + T_{18}}{2} \tag{48}$$

The temperature of the film is then given by equation (7), while the average concentration is calculated by the equation,

$$x_m = \frac{\dot{m}_{16}\, x_{16} + \dot{m}_4\, x_4}{\dot{m}_{16} + \dot{m}_4} \tag{49}$$

- **Calculation of the convection coefficient**

The calculation of the external convection coefficient in the boiling process on film for cylinders arranged horizontally was developed based on Eq (40).

- **Calculation of the global coefficient of heat transfer**

To calculate the overall coefficient was made use of Equation (11):

- **Calculation of Δ Tml**

The calculation of logarithmic mean of temperature differences is based on the heat exchangers resulting in counter-current:

$$\Delta T_{ml} = \frac{(T_{24} - T_{(4,7)}) - (T_{23} - T_{22})}{\ln\left(\frac{(T_{24} - T_{(4,7)})}{(T_{23} - T_{22})} \right)} \tag{50}$$

Calculating the total area of heat transfer:

The total area of heat exchange tubes corresponds to the surface and the relationship between tube length and number of tubes is determined by combining Equation (14), (15) and (42).

5. RESULTS AND DISCUSSION

Table. 6.1 illustrates the values of temperature, pressure and flow measuring instruments read in six different times of operation of the pilot plant.

Table 1 - Preliminary results

Temp. (ºC)	1º	2º	3º	4º	5□	6º
T_3	29,5	30,6	31,0	30,1	31,0	32,4
T_8	30,7	32,0	32,1	32,3	30,9	33,3
T_{13}	29,5	29,4	30,7	32,0	32,6	32,9
T_{14}	30,8	29,1	30,1	31,6	31,2	32,5
T_{15}	28,5	30,9	30,9	31,7	32,0	32,2
T_{17}	33,0	37,6	41,7	56,9	56,9	62,4
T_{21}	63,0	74,5	91,4	104,9	185,1	116,0
T_{22}	49,3	60,7	79,3	93,5	89,4	108,3
T_{24}	30,1	30,1	30,1	31,2	32,4	32,4
T_{26}	30,3	30,3	30,3	31,1	31,1	34,1
T_{27}	28,2	30,1	30,2	30,4	30,5	30,5
T_{28}	28,2	28,7	29,3	30,7	31,3	32,4

Pressure (mBar)	1º	2º	3º	4º	5º	6º
High	149,0	156,0	169,0	182,0	202,0	202,0
Média	80,14	80,14	80,14	80,14	80,14	80,14
Low	10,19	10,19	10,19	10,19	10,19	10,19

Flow in (l/s)	1º	2º	3º	4º	5º	6º
V_3	-	-	-	-	-	-
V_4	-	-	-	-	-	-
V_{14}	-	-	-	-	-	-
$V_{(oil)}$	0,08	0,08	0,08	0,08	0,08	0,08

(Readings taken every 15min)

5.1 Analyzing the heat supplied to the generator of high

Among the points 21 and 22 that match the entry and exit of mineral oil in the generator I, dropping to 5 reading, there is a variation of average temperature of 11.7 º C. Knowing the density of mineral oil that is expressed by the manufacturer of 0.83 g / ml and having a flow rate of mineral oil measure, calculate the mass flow whose value is = 6.64 x 10-2 kg / s. Having also the specific heat also provided by the manufacturer of 0.55 cal / g º C, one reaches the value of Q = 1.74 kW which would be roughly estimated by the heat supplied oil to the generator I even working at a temperature below the proposal for mineral oil in computer simulation.

5.2 Analyzing the pressures:

It can be seen very well that the pressure readings, low and medium did not change during the test. A priori, we identified the following assumptions: an indicator of pressure transducers or defective; small amount of working fluid, pump with low pumping capacity, blocking the holes for expansion; crystallization of the solution.

5.3 Internal flows:

Table 1 does not contain the values of the flows V_3, V_4 and V_{14} which are, respectively: total flow of the solution that I sent to the generator, the flow of the solution that leaves the generator II; the flow of the solution leaving the generator I. These values were not obtained because there a dings on the instruments used were totally unstable, there were large and constant oscillation signal loss. It is believed that the fact leading to this episode lies in diameter below the minimum indicated for the use of equipment that is 13 mm, supplied by the manufacturer. To the measure of the mineral oil flow was not found no obstacle, since the tubing used had a diameter of 19 mm, this measure than the minimum required by the manufacturer of flow measuring instrument. Moreover, the pipe remained completely filled with oil, no waste solids and also air bubbles that could interfere with the reading.

In order to obtain more significant results were taken the following decisions: installation of windows, cleaning inside the machine and clearing holes for expansion; increase in the volume of working fluid, conducting vacuum better equalization of pressures. The installation allowed viewers to see at once the lock hole expansion, opening of the pipe to remove the particles that promote the lock, display or easy replacement of the hose if necessary and reinstall without the need for welding and perfect sealing. By contrast, it is able to break the vacuum in the machine and the occurrence of loss of a small amount of working fluid for possible interventions. After these interventions were carried out further tests and the results shown in Table 2.

Table 2 shows the results of the pilot plant in operation after intervention.

Temperatureem (ºC)	1º	2º	3º	4º
T3	32,0	37,1	57,5	50,2
T8	32,8	37,5	40,4	42,3
T13	33,6	65,5	76,0	59,9
T14	33,6	78,4	84,0	56,4
T15	34,8	44,9	72,8	52,4
T17	34,5	66,8	76,8	78,5
T21	33,3	111,8	156,5	162,8
T22	33,8	105,7	149,4	157,9
T23=T25	31,8	32,3	39,7	40,7
T24	31,9	32,3	41,8	40,9
T26	31,9	33,4	41,6	43,1
T27	30,9	30,3	31,7	31,2
T28	30,7	29,7	30,8	29,9

Pressure (mBar)	1º	2º	3º	4º
High	43,0	335,0	479,0	483,0
Média	33,4	55,2	71,9	75,5
Low	8,8	23,1	62,4	61,9

Flow(l/s)	1º	2º	3º	4º
V3	-	-	-	-
V4	-	-	-	-
V14	-	-	-	-
$V_{(oil)}$	0,08	0,08	0,08	0,08

Readings taken every 15 min.

5.4 Analyzing the results

Comparing the results of Table (2) with those of Table (1), it is easy to verify that the values of temperatures measured at various points of the machine showed significant variations, in the points 3, 13, 14 and 15. These results point to say that there is movement of fluid in the pipes and occurrence of heat exchange between the solutions in order to assess the functioning of the heat exchanger of high, which is not observed previously. The effectiveness of the heat exchanger of high experimentally determined and building on the values of the third reading of Table (2) is 0.698,while the estimated value of the work of Santos (2005) is 0.7. The T_{27} and T_{28} temperatures also show a small effect fridge, but far short of what is the target. As for the pressures obtained in the system is clearly visible that there was a significant improvement. The values obtained in Table (2), we can see a better equalization between the three levels of pressure achieved.

6. CONCLUSIONS

The amplitude of this research work goes beyond the aspects of practical interest in engineering as a whole still reflect theoretical knowledge acquired in its methodology. In an experimental work like this is not always possible to achieve the proposed objectives in full, however, although we have not managed to get the perfect operation of the pilot plant some important conclusions can be described:

Even the pilot plant has not reached the steady state of operation, the cogeneration system used was shown to be able to provide the necessary power required in the steam generator to perform the high-absorption cycle, ie estimated 1.75kw 1.74kw against experimentally determined even outside the steady state. The temperatures reached at the entrances and exits of the heat exchanger led to a high heat exchanger effectiveness consistent with that proposed by computer simulation of the work of Santos (2005).

The holes that were used as expansion devices that reduce the pressure of the system were not effective, since in general constant system crashes during the operation of the pilot plant and is not therefore possible to observe the cooling effect desired. There are valves sold in the market capable of performing this function, however the cost to acquire them is huge which makes it unfeasible. So one of the ideas that lance is trying to develop a device that aggregates the hole with a larger diameter tube into "U" in an attempt to obtain movement of the working fluid without the occurrence of pipe blockage and promotes the decrease of pressure. The knowledge and experience acquired on the basis of participation in the development of the machine that goes from the design of a Project grounded in laws and theoretical concepts, manufacturing of components, preliminary testing of the operation, adjustments and corrections, discussions related to the improvement of components, discovery of defects and solutions to address them, and also new ideas for improvement and future studies, this research did an excellent laboratory.

It should be noted also that the theoretical results obtained by the analysis of thermodynamics and heat transfer are consistent with the literature and therefore do not cease to be validated by experimental results obtained in the pilot plant. On the other hand, so you can get a better match between theory and experiment, it is necessary to make any new interventions, be they in the design of a practical or even referring to some components of the unit that was built.

7. REFERENCES

BEJAN, A., 1948, *Heat Transfer*, JohnWiley & Sons, Inc. , N.Y.

BOSNJAKOVIC, F. & BLACKSHEAR JR. , P. L. , 1965, *Thecnical Thermodynamic* , editado por Holt, Rinehart and Winston, Inc. , N.Y.

ÇENGEL, Y. A., BOLES, M. A., 2006, *Termodinâmica*. 5ª ed. São Paulo, McGraw-Hill.

ÇENGEL, Y. , CIMBALA, J. M. , 2007, *Mecânica dos Fluidos Fundamentos e Aplicações*. 1ª ed. São Paulo: McGraw Hill.

GOMRI, Rabah, 2009, "Second Law Comparasion of Single Effect and Double Effect Vapour Absorption Refrigeration Systems", *Energy Conversion and Management*, v. 50, pp. 1279-1287.

HEROLD, K. E. , RADERMACHER, R. , KLEIN, S. A. , 1996, *Absorption Chillers and Heat Pumps.* CRC Press, Inc. , N.Y.

INCROPERA, F. P. , *et al* , 2008, *Fundamentos de Transferência de Calor e Massa*. 6ª ed. Rio de Janeiro: LTC.

KHALIQ, A. & KUMAR, R., 2008, "Exergy Analysis of Double Effect Vapor Absorption Refrigeration System", *International Journal of Energy Research*, v. 32, pp.161-174.

KAUSHIK, S. C. & ARORA, A., 2009, "Energy and Exergy Analysis of Single Effect and Series Flow Double Effect Water-Lithium Bromide Absorption Refrigeration System", *International Jornal of Refrigeration*, v. 32, pp. 1247-1258.

SANTOS, P. H. D. dos ., 2005,. *Análise exergética de sistemas de refrigeração por absorção de múltiplos estágios com o par água-brometo de lítio.* Dissertação de M.S.c , João Pessoa-PB:

SOUZA, Wilson L. de ., 2007, *Análise Exergoeconômica dos Ciclos de Refrigeração por Absorção de Única Pressão.* Tese de D.S.c , – Universidade Federal da Paraíba/UFPB, João Pessoa – PB.

VARANI, C. M. R. , 2001, *Avaliação Energética e Exergética de uma Unidade de Refrigeração por Absorção Agua/Brometo de Lítio Utilizando Gás Natural.* Tese de D.S.c ,— Universidade Federal da Paraíba, João Pessoa – PB.

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