DESIGN OF HEAT EXCHANGERS FOR WASTE HEAT RECOVERY WIHT THE KALINA CYCLE

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Abstract. *The purpose in the design of shell and tube heat exchangers is to calculate the heat transfer surface area aiming to waste heat recovery with Kalina cycles. The procedure for calculating the area was implemented in software Engineering Equation Solver (EES). We made the area calculation using the Logarithmic Mean Temperature Difference (LMTD) and determining the overall heat transfer coefficient. The calculation of the LMTD was held with the existing functions in EES for the correction factor calculation. The determination of the overall heat transfer coefficient was performed by calculating the convective heat transfer coefficients for the hot and cold sides of the heat exchanger. Additional correlations were implemented in the EES to calculate the thermodynamic properties of* NH_3-H_2O *mixture and to determine the transport properties of the phase change regions. As input data for the calculations were employed geometry (internal and external diameters of the tubes, the exchanger arrangement, geometrical arrangement of the pipes, etc.) and the heat balance of the heat exchanger (temperature, concentration of* NH_3-H_2O *mixture, etc.) obtained by simulation of the Kalina cycle. As an example of the results that were obtained for the evaporator, the overall heat transfer coefficient was found to be 108,5* W/m^2K *and the heat transfer surface area of 1170* m^2*. This value of the overall heat transfer coefficient is within the ranges reported in the literature by other researchers.*
Keywords*: overall heat transfer coefficient, heat exchanger, waste heat recovery, kalina cycle, LMTD.*

1. INTRODUCTION

The waste heat from production processes are usually at low temperature. Hence, it is necessary a good heat exchanger design for recovery and produce power in thermodynamic cycles. The Kalina cycle is excellent choice for waste heat recovery from production process. This cycle has as working fluid an ammonia-water (NH_3-H_2O) mixture that offers a better efficient on other cycles such as Rankine cycle. The use of NH_3-H_2O mixture is a complicating factor in determining the convective coefficient of heat transfer, the calculation of which has a certain complexity. Empirical correlations are used to find the convective coefficients the cold and hot side, and so determine the overall heat transfer coefficient. The determination of the heat transfer surface area is linked to the overall coefficient of heat transfer intrinsically. In this context the purpose in the design of shell and tube heat exchangers is to calculate the heat transfer surface area, considering the calculation of the global heat transfer coefficient, aiming to waste heat recovery with Kalina cycles for power generation from waste heat of cement production process.

2. LITERATURE REVIEW AND PROPOSED CYCLE

The determination of the heat transfer coefficient for mixtures is complex, particularly when the process involves phase change. Best and Rivera (1999) found experimentally that the heat transfer coefficient in boiling NH_3-H_2O mixture is 2 to 3 times higher than that of the ammonia-lithium nitrate mixture. Khir et al (2005) reported that heat and mass flow strongly influences the heat transfer coefficient in the NH_3-H_2O mixture, but the concentration of the mixture does not have great influence. Araújo et al (2013) studied the behavior of the coefficient of heat transfer of various fluids in the evaporation process, and fluid from the studies, R717 (ammonia) presented a more efficient heat transfer. Shah (2015) modified correlation shape to be applied in mixtures, and validated using the comparison with other correlations in literature. For Shah, Liu and Winterton, and Gungor and Winterton correlations average deviations were respectively 19.5%, 20.4% and 20.7%. The study of transport property (liquid phase and vapor phase) two cases of Kalina cycle performed by Kaern and others (2015) indicated a greater influence of the properties of the liquid phase in the estimation of the area of the exchangers. The NH_3-H_2O mixture has a boiling point below than pure water and boiling temperature variable promoting the reduction of losses in heat transfer. According Mirolli (2006), this provides an efficient use for waste heat recovery from exhaust gases in cement process. Kalina et al (1995) found a large ratio between the exchange surface of the evaporator and the power generated by the cycle. And in thermodynamic cycle for waste heat recovery is known that the heat exchangers are devices that influence over the initial investment cost and the amount of power generated by the system, hence the importance of a proper sizing of this equipment second Arrieta et al. (2015).

The Kalina cycle proposed by Silveira and Arrieta (2015) for the recycling of waste gases from a fictitious plant cement production is presented in Fig. 1. The thermodynamic calculation results of this cycle were used as a basis for the development of the employed method of calculation the determination of the convective heat transfer coefficient for heat exchangers which phase shift occurring NH_3-H_2O mixture, boiling process in the evaporator (EVAP) and condensation process in the condenser (COND).

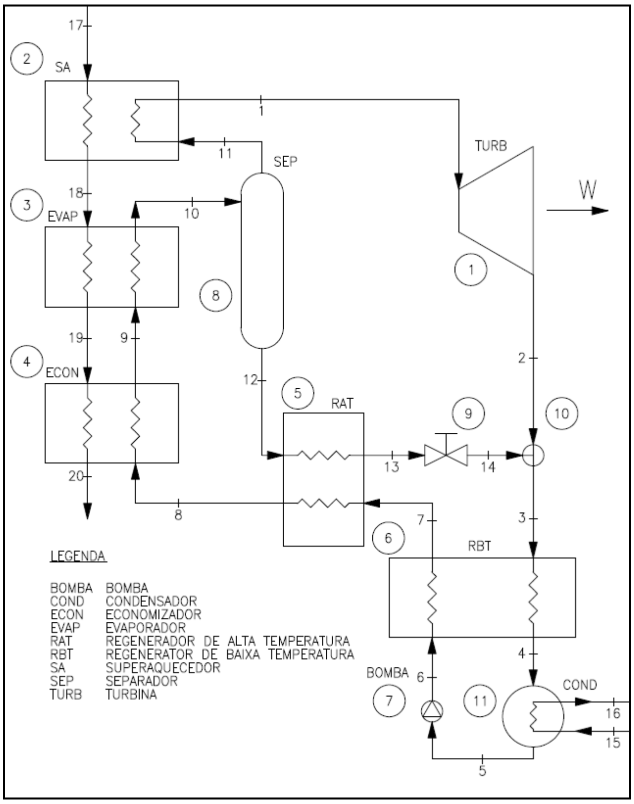

Figure 1. The Kalina cycle considered for heat exchanger sizing.

3. CONVECTIVE HEAT TRANSFER COEFFICIENT CALCULATION METHODOLOGY

Fig. 2 shows the algorithm with the parameters relevant to the calculation of the area of heat exchange used in each heat exchanger. Figure 2 shows the algorithm with the parameters relevant to the calculation of the area of heat exchange used in each heat exchanger. First set up the all the input data shown in the first step of the algorithm. The data are entered in two functions designed to calculate the convective heat transfer coefficients, one for the inside and one to the outside of the tube. In carrying out the duties are calculated some thermodynamic and transport properties of the mixture, using correlations that were described in detail later in this topic. The functions return the values found the heat of the inner and outer side transfer coefficients, completing the second step of the algorithm. In the third phase, defined the deposition efficiency factors and the fins for the case of finned tubes and calculate the overall coefficient of heat transfer. The last step consists in applying MLDT method for determining the surface area of heat exchange.

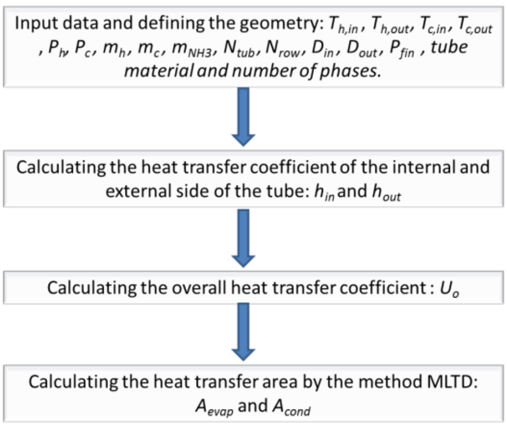

Figure 2. Calculation algorithm.

The calculations in EES were started with the input data related to thermodynamic and transport properties of each state involved presented in Tab.1, and geometric parameters of heat exchangers. This data come from the thermodynamic calculation of the cycle performed by Silveira and Arrieta (2015).

Table 1. Evaporator and condenser input data.

Fluid	Property	Evaporator		Condenser	
		Inlet	Outlet	Inlet	Outlet
Residual Gas (Evaporator) or Water (Condenser)	Temperaure (K)	557. 76	401.94	295.15	303.15
	Pressure (kPa)	101.32	101.32	250.00	250.00
	Mass flow (Kg/s)	47.747	47.747	265.034	265.034
NH_3-H_2O mixture	Temperaure (K)	379.68	431.45	340.28	300.15
	Pressure (kPa)	6375	6375	902.62	902.62
	Mass flow (Kg/s)	0.8677	0.8677	0.8677	0.8677

3.1 Convective heat transfer coefficient of the inside

The methodology for calculating the convective transfer coefficient for the process of change of phase mixtures, both boiling as condensation, was presented by Shah (2015) and is based on the addition of the two parameters, the nucleate boiling of the heat transfer mechanisms and effects of the forced convection in the anelar system.

$$h_{MIX} = h_{NB} + h_{FC} \tag{1}$$

The first step is to calculate the heat transfer coefficient for the contribution to nucleate boiling, the method uses the correlation Thome and Shakir (1987) as a correction factor applied to the correlation Dittus and Boelter (1985).

$$h_{NB} = h_l F_{TS}(230Bo^{0.5}) \tag{2}$$

Where:

$$h_l = 0.023\mathrm{Re}_l^{4/5}\,\mathrm{Pr}_l^n\left(\frac{k_l}{D_{in}}\right) \tag{3}$$

$$F_{TS} = \left\{1+\left(\frac{h_{PB}}{q''}\right)(T_{DP}-T_{BP})\left[1-\exp\left(\frac{-Bo}{\rho_l h_{\mathrm{lg}}}\right)\right]\right\}^{-1} \tag{4}$$

Where in equations 1 to 4 the h_{MIX} is the convective heat transfer coefficient of the mixture; h_{NB} is convective heat transfer coefficient of the nucleate boiling; h_{FC} is convective heat transfer coefficient of the forced convection; h_l is convective heat transfer coefficient of the liquid phase; F_{TS} correction factor of the Thome and Shakir; *Bo* is boiling number; *Re* is Reynolds number; *Pr* is Prandtl number; k_l is thermal conductivity of the liquid phase; D_{in} is inside diameter; h_{PB} is heat transfer coefficient of the pool boiling; q'' is heat flux, T_{DP} is dew point temperature of mixture; T_{BP} is bubble point temperature of mixture; ρ_l is density of the liquid phase; h_{lg} is latent heat of vaporization.

The second installment of the convective heat transfer coefficient of the mixture on the forced convection is calculated by the method of Bell and Ghaly (1973).

$$h_{FC} = \left(\frac{Co^{0.8}}{1.8h_l} + \frac{Y}{h_v}\right)^{-1} \tag{5}$$

Where:

$$Co = \left(\frac{1}{x-1}\right)^{0.8}\left(\frac{\rho_v}{\rho_l}\right)^{0.5} \tag{6}$$

$$Y = xC_{p,v}\frac{dT_{PO}}{dH} \tag{7}$$

Where in equations 5 to 7 the *Co* is Convection number; *Y* is factor in Bell-Ghaly method; h_v is convective heat transfer coefficient of the vapor phase; *x* is vapor quality; ρ_l is density of the vapor phase; $C_{p,v}$ specific heat of vapor at constant pressure; *H* is specific enthalpy.

3.2 Convective heat transfer coefficient of the outside

The staggered arrangement of circular tubes with fins is applied to the evaporator. Thulukkanam (2013) proposes the correlation Briggs and Young (1963) for calculating the heat transfer coefficient on the outside in these cases.

$$h_{out} = 0.134(\mathrm{Re}_D)^{0.681}(\mathrm{Pr})^{1/3}\left(\frac{s_a}{h_a}\right)^{0.2}\left(\frac{s_a}{\delta_a}\right)^{0.1134} \tag{8}$$

For staggered and smooth tubing used in the condenser, Bejan (2013) presents a number of correlations for the definition of convective heat transfer coefficient for the outside dependent on the Reynolds number. For Reynolds number between 1000 and 20,000, Nusselt number is calculated as:

$$\overline{Nu}_D = 0.35(\mathrm{Re}_D)^{0.6}(\mathrm{Pr})^{0,36}\left(\frac{\mathrm{Pr}}{\mathrm{Pr}_w}\right)^{1/4} \tag{9}$$

Where in equations 8 and 9 the h_{out} is convective heat transfer coefficient of the outside; s_a is spacing between fins; h_a is fin height; δ_a is thickness fin; Nu_D is Nusselt number; Pr_w is Prandtl number in the wall temperature.

3.3 Overall heat transfer coefficient

The overall heat transfer coefficient for smooth or finned tubes, by Incropera et al. (2008), which was modified to meet the particularities of the exchangers studied.

$$U_o = \left\{\frac{1}{h_{in}} + R_{d,c} + R_W A_c + \frac{R_{d,h}A_c}{\eta_a A_h} + \frac{A_c}{\eta_a h_{out} A_h}\right\}^{-1} \tag{10}$$

Where in equation 10 the $R_{d,c}$ is resistance to deposition of cold fluid; R_w is thermal resistance of the wall, A_c is area of the cold side; $R_{d,h}$ is resistance to deposition of hot fluid; η_a is Fin efficiency; A_h is area of the hot side.

3.4 Method of application Mean Logarithmic Temperature Difference (MLTD)

For specification of the area of necessary heat transfer to each heat exchanger, MLTD method was applied. The correction factor is calculated using proper function of the ESS software.

$$\dot{q} = U_o A \Delta T_{lm} F \tag{11}$$

Where ΔT_{lm} given by:

$$\Delta T_{lm} = \frac{\Delta T_I - \Delta T_{II}}{\ln\left(\frac{\Delta T_I}{\Delta T_{II}}\right)} \tag{12}$$

Where:

$$\Delta T_I = T_{h,in} - T_{c,out} \tag{13}$$

$$\Delta T_{II} = T_{h,out} - T_{c,in} \tag{14}$$

Where in equations 11 and 14 the $\dot{q}$ is heat rate; U_o is overall heat transfer coefficient; A is area of heat transfer; ΔT_{lm} is mean logarithmic temperature difference; F is factor correction; $T_{h,in}$ is the inlet temperature of the hot fluid; $T_{c,out}$ is the outlet temperature of the cold fluid; $T_{h,out}$ is the outlet temperature of the hot fluid; $T_{c,in}$ is the intlet temperature of the cold fluid.

After heat transfer surface area calculation finished a parametric analysis was performed, and the influence of geometrical parameters of the heat transfer surface and the overall coefficient of heat transfer was studied. Those results are presented in the next topic.

4. RESULTS AND DISCUSSION

In Tab. 2 the main results for the two heat exchangers dimensioned studies are presented. The overall heat transfer coefficient is found within the range specified by Walas (1990), for condenser type shell and tube of the overall coefficient of heat transfer is between 283.5 and 1134.9 W/m^{2}K. In the evaporator, the specified range is 141.75 to 340.47 W/m^{2}K.

Table 2. Evaporator and condenser results.

Property or Parameter	Evaporator	Condenser
	Values	Values
Convective heat transfer coefficient of the inside (W/m^{2}K)	584.8	943.9
Convective heat transfer coefficient of the outside (W/m^{2}K)	681.5	3538.0
Overall heat transfer coefficient (W/m^{2}K)	108.4	660.3
Number of tubes	80	80
Length of tubes (m)	6.834	5.194
Area (m^2)	1170.0	889.2

The parametric studies show the influence on global heat transfer coefficient, heat transfer surface area and average convective heat transfer coefficient inside and outside the tubes with the variation of the outer diameter ($0.0761 \leq D_{out} \leq 0.108$ m) and spacing between the tubes ($0.008 \leq s_{tub} \leq 0.014$ m). Note that the outer diameter is connected such to the inner diameter so as to keep constant the thickness of the tube wall, that is the variation in the outer diameter of the inner diameter also varies.

The calculations results are shown in Figure 3. The graphics (A), (B) and (C) present the values of the evaporator. The graphic (D), (E) and (F) present the values of the condenser.

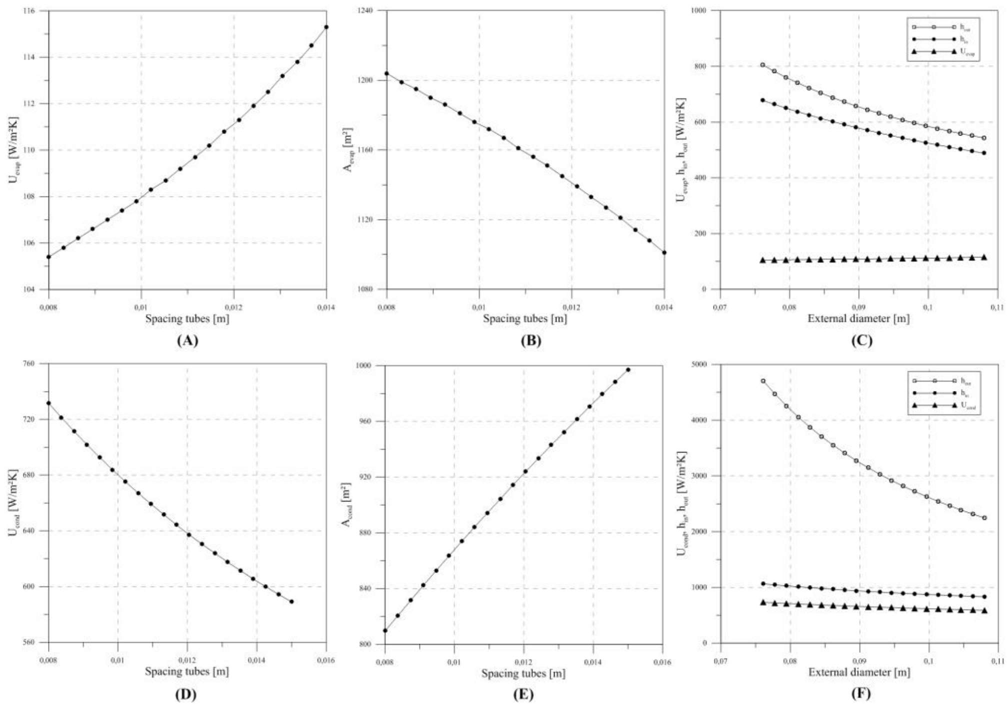

Figure 3. Results of parametric analysis

Figure 3 (A) shows the curves of the overall heat transfer coefficient for the evaporator by varying the spacing between the heat exchanger tubes. The overall heat transfer coefficient has an 8% increase in its value with increasing spacing. When increasing the spacing between the tubes, the gas velocity decreases and the heat transfer coefficient on the outside decreases. The reduction in heat transfer coefficient results in increased efficiency of the fins, which promotes increased overall heat transfer coefficient.

Figure 3 (B) shows the curves of the area of heat transfer for the evaporator by varying the spacing between the heat exchanger tubes. As expected, the opposite occurs because as the heat transfer coefficient increased with the spacing of the tubes the heat transfer area tends to decrease, this is because these two variables are inversely proportional.

Figure 3 (C) shows the conduct of the curves of convective heat transfer coefficients of inner and outer side of the evaporator, and the overall coefficient of heat transfer due to the external pipe diameter. There was a decrease of 27.93% and 32.54% for the heat transfer coefficients of inner and outer side respectively. In the heat transfer coefficient of the inner side, it is caused by reducing the mass flow caused by increased internal area. In the heat transfer coefficient on the outside, it is caused inverse proportionality of the external diameter and the heat transfer coefficient in calculating the Nusselt number. The same happens in Fig. 3 (F) for the heat transfer coefficients of inner and outer side.

Figure 3 (D) shows the curves of the overall heat transfer coefficient for the condenser by varying the spacing between the heat exchanger tubes. The overall heat transfer coefficient is reduced by 18.6% in value with increasing spacing. It occurs because reducing the mass flow caused by the increased area between the tubes. Because the condenser tubes are smooth, no interference efficiency of the fins in this case.

Figure 3 (E) shows the curves of the area of heat transfer for the condenser by varying the spacing between the heat exchanger tubes. As expected, the opposite occurs because as the heat transfer coefficient reduced with the spacing of the tubes the heat transfer area tends to decrease caused by the fact the two variables are inversely proportional.

Figure 3 (F) shows the curves of the convective heat transfer coefficients of inner and outer side of the condenser, and the overall heat transfer coefficient due to the external pipe diameter. There was a decrease of 21.9% and 49.5% for the heat transfer coefficients of inner and outer side respectively.

4. CONCLUSIONS

The overall heat transfer coefficient and heat transfer area to the evaporator were found 108.4 m^2 and 1170.0 W/m^2K respectively. In the condenser, the overall heat transfer coefficient and area were found to 660.3 W/m^2K and 889.2 m^2 respectively. For the parametric analysis, it is concluded that the external diameter of the pipe has a strong influence on the overall heat transfer coefficient, and this may be positive or negative depends on the intensity of the influence of this variable on convective coefficients of heat transfer from the inner side and outer the tube and the tube inner area. In the evaporator, the increased outer diameter provides an increase in the overall coefficient of heat transfer, since the reverse occurs in the condenser, mainly caused by construction differences of the two heat exchangers.

5. ACKNOWLEDGEMENTS

The authors thank FAPEMIG, FAPEMIG/CEMIG APQ-03422-12 research project, ANNEL Research and Developed program (P&D) research project GT0554 and Pontifical Catholic University of Minas Gerais – PUC Minas for the financial support to this work.

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