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Calculation of the Heat Transfer Surface Area of Heat Exchangers for Waste Heat Recovery with the Kaline Cycle

Felipe Raúl Ponce Arrieta, Josme de Sousa Santos, Wagner Henrique Saldanha

Department of Mechanical Engineering, Pontifical Catholic University of Minas Gerais, Av. Dom José Gaspar, 500 – CEP 30535-901 – Belo Horizonte – MG – Brazil

***Abstract*— *The purpose is to calculate the heat transfer surface area of the shell and tube heat exchanger aiming recovery to waste heat with Kalina cycles. The procedure for calculating the area was implemented in software Engineering Equation Solver (EES), where the calculation of the LMTD was held with the existing functions in EES for the correction factor calculation, and 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 tubes) 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*—heat exchanger, waste heat recovery, Kalina cycle.**

I. 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 [1-4] 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 implemented in software Engineering Equation Solver (EES), aiming recovery to waste heat with Kalina cycles for power generation from waste heat of cement production process.

II. LITERATURE REVIEW AND PROPOSED CYCLE

The determination of the heat transfer coefficient for mixtures is complex, particularly when the process involves phase change. Rivera and Best [5] 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 [6] 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 [7] 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 [8] modified correlation shape to be applied in mixtures, and validated using the comparison with other correlations in literature.

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 [9], this provides an efficient use for waste heat recovery from exhaust gases in cement process. Kalina et al [10] 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. [11].

The Kalina cycle proposed by Arrieta et al. [11]. 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 and condensation process in the condenser.

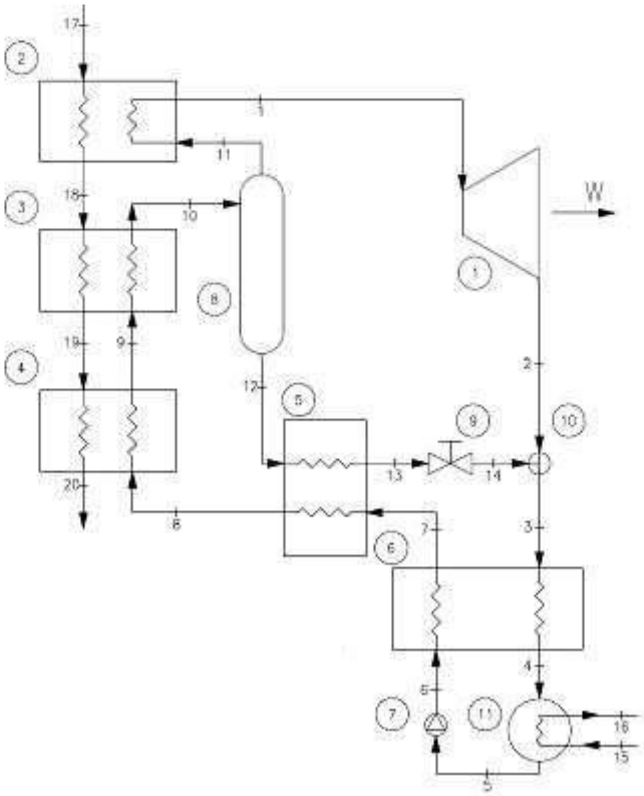

Fig. 1: Kaline cycle used for heat exchanger sizing

In Fig. 1 each number corresponds to an equipment, being:

- 1 turbine,
- 2 superheater,
- 3 evaporator,
- 4 economizer,
- 5 high temperature regenerator,
- 6 low temperature regenerator,
- 7 pump,
- 8 separator,
- 9 valve,
- 11 condenser.

III. INPUT DATA IN EES

The calculations in EES were started with the input data related to thermodynamic and transport properties of each state involved presented in Table 1 and Table 2. This data come from the thermodynamic calculation of the cycle performed by Arrieta et al. [11].

Table 1: Input data for evaporator

	Property	Evaporator	
		Inlet	Outlet
Residual Gas	Temperaure (K)	557. 76	401.94
	Pressure (kPa)	101.32	101.32
	Mass flow (Kg/s)	47.747	47.747
NH_3-H_2O mixture	Fluid	379.68	431.45
	Pressure (kPa)	6375	6375
	Mass flow (Kg/s)	0.8677	0.8677

Table 2: Input data for condenser

Fluid	Property	Condenser	
		Inlet	Outlet
Water	Temperaure (K)	295.15	303.15
	Pressure (kPa)	250.00	250.00
	Mass flow (Kg/s)	265.034	265.034
NH_3-H_2O mixture	Temperaure (K)	340.28	300.15
	Pressure (kPa)	902.62	902.62
	Mass flow (Kg/s)	0.8677	0.8677

IV. INTERNAL HEAT TRANSFER COEFFICIENT

The methodology for calculating the convective transfer coefficient for the process of change of phase mixtures, both boiling as condensation, was presented by Shah [8] and is based on the addition of the two parameters, the nucleate boiling of the heat transfer mechanisms and effects of the forced convection.

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

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

Where:

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

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

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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:

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

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

$$Y = x C_{p,v} \frac{dT_{PO}}{dH} \quad (7)$$

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.

V. EXTERNAL HEAT TRANSFER COEFFICIENT

The staggered arrangement of circular tubes with fins is applied to the evaporator. Thulukkanam [12] proposes the following correlation for external heat transfer coefficient:

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

For staggered and smooth tubing used in the condenser, Bejan [13] 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:

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

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.

VI. CALCULATION OF THE AREA OF THE HEAT TRANSFER

The LMTD method was applied for specification of the thermal exchange area to each heat exchanger. The correction factor is calculated using proper function of the ESS software.

$$q = U_o A \Delta T_{lm} F \quad (10)$$

The overall heat transfer coefficient (U_o) for smooth or finned tubes was used the formulation of Incropera et al. [14], which was modified to meet the particularities of the exchangers studied.

$$U_o = \left\{ \frac{1}{h_{in}} + R_{d,c} + R_W A_c + \frac{R_{d,h} A_c}{\eta_a A_h} + \frac{A_c}{\eta_a h_{out} A_h} \right\}^{-1} \quad (11)$$

Where in equation 11 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.

And ΔT_{lm} given by:

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

$$\Delta T_I = T_{h,in} - T_{c,out} \quad (13)$$

$$\Delta T_{II} = T_{h,out} - T_{c,in} \quad (14)$$

Where in equations 10 to 14 the *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.

VII. RESULTS

The overall heat transfer coefficient is found within the range specified by Walas [16], for condenser type shell and tube of the overall coefficient of heat transfer is between 283.5 and 1134.9 W/m²K. In the evaporator, the specified range is 141.75 to 340.47 W/m²K. Table 3 and Table 4 shows the values found for internal heat transfer coefficient, external heat transfer coefficient, overall heat transfer coefficient and area of the heat transfer for the evaporator and condenser, respectively.

Table 3: Results for Evaporator

Parâmetro	Values
Internal heat transfer coefficient (W/m²K)	584.8
External heat transfer coefficient (W/m²K)	681.5
Overall heat transfer coefficient (W/m²K)	108.4

Area of the heat transfer (m^2)	1170.0

Table 4: Results for Condenser

Parâmetro	Values
Internal heat transfer coefficient (W/m^2K)	943.9
External heat transfer coefficient (W/m^2K)	3538.0
Overall heat transfer coefficient (W/m^2K)	660.3
Area of the heat transfer (m^2)	889.2

The parametric studies performed in EES 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). Note that the outer diameter was 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.

VIII. CONCLUSION

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 tube has a strong influence on the overall heat transfer coefficient. 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.

It is suggested as a future study the optimization of the evaporator and condenser that are used in the Kalina cycle, as in the case study found in Saldanha et al. [16] for shell and tube heat exchanger with single phase flow.

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