CONDENSER SURFACE AREA OF HEAT EXCHANGE ESTIMATION

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Abstract. *In order to determine the minimum surface area (or cost) of heat transfer of a shell-and-tube condenser, this article aimed to deeply study and understand this kind of equipment, and also to develop a couple of computer assisted models. At first, the calculation method proposed by the Heat Exchange Institute (HEI) was applied, being validated through a regarded software called Gate Cycle™. The modeling created was automatized and optimized (with genetic algorithm) using the software Engineering Equation Solver (EES), only valid for water as working fluid. Afterwards, a second method proposed by Kakaç and Liu, in the book "Heat Exchangers: Selection, rating and thermal design", which, theoretically, could be used for many different working fluids, was also validated once the input data from the first method was either used or straightly adapted when needed, permitting, after all, to obtain an equivalent surface area of heat transfer. The second method, just like the first one, was automatized and optimized in EES with the single purpose to decrease the overall surface area of heat transfer as well as to estimate the condenser cost. In both of them, variables such as cooling water velocity, tube gauge, outside tube diameter and tube material among other variables, behaved as iteration parameters in the EES' genetic algorithm, making possible to determine the optimum surface area of heat transfer with a theoretical cost range, simulating, therefore, a possible real scenario. From the results obtained it was easy to observe that the variables that must be considered to attain the surface area of heat transfer for condensers depend rightly on the constructive characteristics and thermodynamics features of the equipment. The calculation optimization resulted in a reduction of surface area of heat transfer about 37% and 26% for the first and the second modeling, respectively, validating the programming to be used as a first step analysis for shell-and-tube heat exchangers. The minimum cost was obtained with the Kakaç and Liu method.*

Keywords: *Condenser, Surface area of heat transfer, Optimization, Cost.*

1. INTRODUCTION

Condenser is an equipment of a cooling system used to refrigerate and to dissipate the heat absorbed by the process to the external environment, changing the phase of the working fluid from steam to condensed liquid. Heat exchangers have several applications, like in households or industries, widely used in domestic refrigerators, industrial and automotive cooling systems, cycles of electric generation, chemical processes and others. Depending on the cooling system, condensers can be divided in three different types, liquid cooled condensers, air cooled condenser and evaporative condensers. Shell-and-tube condensers use liquids or air as coolant, consisting in a chamber in which horizontal and parallel tubes are arranged within two plates on each extremity of the housing, where the coolant flows inside the tubes and the working fluid flows in the shell side. Particularly, virtually every thermodynamic cycle of electric generation by power plants uses shell-and-tube as heat transfer due the high amount of heat dissipated using concomitantly large arrangements of surface area of heat transfer but occupying relative small places. The condenser surface area of heat exchange estimation aims to develop a simple computer assisted calculation methodology to attain the minimum surface area of heat exchange of a shell-and-tube condenser as well as to estimate the equipment cost.

2. LITERATURE REVIEW

In a previous work, Tassou and Green (1981) developed a mathematic model to determine the heat transfer coefficient of a condenser. More recently Nebot *et al.*, 2007, developed a modeling to predict the maximum value of thermal resistance for fouling under different conditions. Queiroz and Costa (2008) presented a study about a shell-and-tube heat exchange design optimization in order to create an algorithm to minimize the surface area of heat exchange. Allem *et al.*, 2009, perform a study aiming the optimization of condenser arrangements using genetic algorithms. Wang *et al.*, 2009, made an experimental research for the better efficiency in shell and tube heat exchangers considering the circular sealing between the internal wall of the shell and the internal baffles. Sanaye and Hajabdollahi (2010) performed and presented an optimization model for a shell and tube heat exchanger aiming to improve the heat transfer efficiency and reducing the total cost. Pieve and Salvadori (2011) tested an air condenser installed in a heat recuperator using a mathematical model to predict the influence of the ambient temperature variation in the condenser performance. Walker *et al.*, 2012, showed a methodology to quantify the economic impact of fouling in the performance of a thermal power plant condenser. They included in the analysis several factors like fouling rate, condenser duty, cooling water inlet temperature and condenser cleanliness. Zeng *et al.*, 2012, used three different condenser tube arrangements in order to verify the average heat transfer coefficient, but none of the three reached the HEI's standards.

3. METHODOLOGY

This topic summarized the equations used for the surface area of heat transfer estimation by the HEI (2006) method and the Kakaç and Liu (2002) method, as well as the ones for the condenser cost estimation. Both methods and the cost calculation were develop in EES for optimization.

3.1 Surface area of heat exchange according to the HEI (2006) method

In order to design or predict the performance of a condenser it is extremely necessary to correlate the heat transfer rate to inlet and outlet fluid temperature, overall heat transfer coefficient and the surface are of heat exchange:

$$\dot{Q} = U \cdot As \cdot \Delta T_{MLTD} \tag{1}$$

Where:

$$\dot{Q} = \dot{m}_H \cdot (h_{H,i} - h_{H,o}) \tag{2}$$

Or for the cooling fluid,

$$\dot{Q} = \dot{m}_C \cdot \left(h_{C,o} - h_{C,i}\right) = \dot{m}_C \cdot Cp_C \cdot \left(T_{C,o} - T_{C,i}\right) \tag{3}$$

$$\Delta T_{MLDT} = \frac{\Delta T_1 - \Delta T_2}{ln\frac{\Delta T_1}{\Delta T_2}} \tag{4}$$

$$U = U_1 \cdot F_W \cdot F_M \cdot F_C \tag{5}$$

In Eq. (1) to Eq. (5) $\dot{Q}$ is the heat transfer rate, in J/s, U is the global heat transfer coefficient, in W/(m²K), As is the surface area of heat transfer, in m², ΔT_{MLTD} is the logarithmic mean temperature difference, in K, $\dot{m}_H$ is the hot fluid mass flow rate, in kg/s, $h_{H,i}$ is the specific enthalpy of the hot fluid at the condenser inlet, in J/kg, $h_{H,o}$ is the specific enthalpy of the hot fluid at the condenser outlet, in J/kg, $\dot{m}_C$ is the cooling fluid mass flow rate, in kg/s, $h_{C,i}$ is the specific enthalpy of the cooling fluid at the condenser inlet, in J/kg, $h_{C,o}$ is the specific enthalpy of the cooling fluid at the condenser outlet, in J/kg, Cp_C is the average specific heat at constant pressure of the cooling fluid, in J/(kgK), $T_{C,o}$ is the outlet temperature of the cooling flow, in K, $T_{C,i}$, is the inlet temperature of the cooling flow, in K, ΔT_1 is the temperature difference between the hot fluid temperature at the condenser outlet and the cooling fluid temperature at the condenser inlet, in K, ΔT_2 is the temperature difference between the hot fluid temperature at the condenser inlet and the cooling fluid temperature at the condenser outlet, in K, U_1 is the uncorrected global heat transfer coefficient due to the fluid velocity and the diameter of the tube, in W/(m²K), F_W is the dimensionless correction factor for inlet temperature, F_M is the dimensionless correction factor for the tube wall gauge and F_C is the dimensionless correction factor for cleanliness. The last four values used were based on experimental tests under real operation conditions.

The results obtained with this method were validated using the software Gate Cycle^{TM}. The validation was accomplished simulating a condenser in the software Gate Cycle^{TM} interface using the same data from the HEI (2006) method example.

3.2 Surface area of heat exchange according to Kakaç and Liu (2002) method

The correlation of the heat transfer rate to inlet and outlet fluid temperature, overall heat transfer coefficient and the surface are of heat exchange is:

$$\dot{Q} = U_M \cdot As_o \cdot \Delta T_{MLTD} \tag{6}$$

Where ΔT_{MLTD} is computed by Eq. (4) and As_o refers to the outside of the tubes and:

$$U_M = \frac{U_{o,i} + U_{o,o}}{2} \tag{7}$$

With,

$$U_{o,i} = \frac{1}{(\frac{1}{U_{o,i}} + R_t)} \tag{8}$$

$$U_{o,o} = \frac{1}{(\frac{1}{U_{o,o}} + R_t)} \tag{9}$$

Where,

$$U_{o,i} = 0.728 \cdot \left(\frac{{\rho_l}^2 \cdot g \cdot h_{lg} \cdot {k_l}^3}{\mu_l \cdot \Delta T_{o,i} \cdot d_o}\right)^{1/4} \cdot \frac{1}{{N_L}^{1/6}} \tag{10}$$

$$U_{o,o} = 0.728 \cdot \left(\frac{{\rho_l}^2 \cdot g \cdot h_{lg} \cdot {k_l}^3}{\mu_l \cdot \Delta T_{o,o} \cdot d_o}\right)^{1/4} \cdot \frac{1}{{N_L}^{1/6}} \tag{11}$$

And

$$\Delta T_{o,i} = \Delta T_i \cdot [1 - (R_t \cdot U_{o,i})] \tag{12}$$

$$\Delta T_{o,o} = \Delta T_0 \cdot [1 - (R_t \cdot U_{o,o})] \tag{13}$$

Where,

$$R_t = R_{fo} + \left[\left(\frac{1}{U_c} + R_{ft}\right) \cdot \frac{d_o}{d_i}\right] + \left(\frac{t_s}{k_{tube}} \cdot \frac{d_0}{D_{avg}}\right) \tag{14}$$

And

$$D_{avg} = \frac{d_o - d_i}{ln\frac{d_o}{d_i}} \cong \frac{1}{2} \cdot (d_o + d_i) \tag{15}$$

$$U_c = \frac{Nu \cdot k_c}{d_i} \tag{16}$$

With

$$Nu = \frac{(f/2) \cdot Re \cdot Pr_c}{1.07 + \left[12{,}7 \cdot (f/2)^{\frac{1}{2}} \cdot \left({Pr_c}^{\frac{2}{3}} - 1\right)\right]} \tag{17}$$

And

$$f = (1.58 \cdot [ln(Re)] - 3.28)^{-2} \tag{18}$$

$$Re = \frac{V_c \cdot \rho_c \cdot d_i}{\mu_c} \tag{19}$$

$$Pr_c = (\frac{Cp_c \cdot \mu_c}{k_c}) \tag{20}$$

In Eq. (6) to Eq. (20) $\dot{Q}$ is the heat transfer rate, in J/s, U_M is the average global heat transfer coefficient, in W/(m²K), As_o is the surface area of heat transfer, in m², ΔT_{MLTD} is the logarithmic mean temperature difference, in K, $U_{o,i}$ is the global heat transfer coefficient outside the tubes at the steam inlet section, in W/(m²K), $U_{o,o}$ is the global heat transfer coefficient outside the tubes at the (condensed) steam outlet section, in W/(m²K), U_C is the global heat transfer coefficient inside the tubes, in W/(m²K), R_t is the total thermal resistance, in m²K/W, R_{fo} is the fouling thermal resistance outside the tubes, in m²K/W, R_{ft} is the total thermal resistance inside the tubes, in m²K/W, $\Delta T_{o,i}$ is the temperature difference between the saturation temperature and the fouling surface temperature outside tubes at the inlet section, in K, $\Delta T_{o,o}$ is the temperature difference between the saturation temperature and the fouling surface temperature outside tubes at the outlet section, in K, ΔT_i is the temperature difference between the saturation temperature and the cooling water inlet temperature, in K, ΔT_o is the temperature difference between the saturation temperature and the cooling water outlet temperature, in K, N_L is the number of tube lines, ρ_l and ρ_C are the density of the saturated liquid and the cooling water, respectively, in kg/m³, μ_l and μ_C are the absolute viscosity of the saturated liquid and the cooling water, respectively, in Ns/m², k_l and k_C are the conductivity of the saturated liquid and the cooling water, respectively, in W/(mK), k_{tube} is the conductivity of the tube material, in W/(mK), h_{lg} is the specific enthalpy of the saturated liquid, in J/kg, Cp_C is the average specific heat at constant pressure of the cooling fluid, in J/(kgK), Pr_C is the average Prandtl number of the cooling fluid, d_o, d_i and D_{avg} are the outside, inside and average diameters, in m, V_c is the cooling water velocity inside the tubes, in m/s, *Re* is the Reynolds number inside the tubes, *Nu* is the Nusslet number inside tubes, g is the acceleration of gravity, in m/s², and f is the Fanning friction factor.

3.3 Condenser cost estimation

The condenser cost estimation was done with the information presented by Seider *et al.* (2009) and the equation:

$$C = F_P \cdot F_M \cdot C_B \tag{21}$$

Where

$$C_B = C_{B_{ref}} \cdot \left[\frac{As}{As_{ref}}\right]^N \tag{22}$$

$$F_M = a + \left[\frac{A}{100}\right]^b \tag{23}$$

$$F_P = 0.9803 + 0.018 \cdot \frac{P}{100} + 0.0017 \cdot \left[\frac{P}{100}\right]^2 \tag{24}$$

In Eq. (21) to Eq. (24) C is the condenser cost in US$, C_B is the uncorrected cost, in US$, F_P is the dimensionless operation pressure correction factor, F_M is the dimensionless material correction factor, $C_{B_{ref}}$ is the reference uncorrected cost, in US$, As_{ref} is the reference condenser heat transfer surface area, in m², As is the condenser heat transfer surface area, in m², the N factor is 0.53 accordingly to Seider *et al.* (2009) for shell and tube heat exchanger, the coefficients a and b are specified by the shell and tube materials. In the Eq. (23) the value of the area A must be expressed in ft², while in Eq. (24) the condenser pressure P must be expressed in psi.

3.4 Simulation data

The data assumed for the initial simulation was taken from the HEI (2006) method and presented in Tab. 1. The values in this table are typical for a thermal power plant condenser. The values were used to verify the computational code developed in EES for the HEI (2006) method, as well as for the validation in the software Gate Cycle™.

Table 1. Initial data for simulation.

Parameter	Value	Parameter	Value
Condenser pressure	3985.78 Pa	Cleanliness factor	0.8
Condenser temperature	302.04 K	Tube outside diameter	0.0254 m
Condenser duty	302697700 J/s	Tube inside diameter	0.02398 m
Steam flow	134.06 kg/s	Tube material	Stainless Steel 316
Cooling water flow	16.02 m^3/s	Cooling water	Not salt water
Cooling water inlet temperature	288.71 K	Cooling water density	1 g/cm^3
Water Velocity inside the tubes	2.74 m/s	Cooling water specific heat	4180 J/kg K

4. ANALYSIS AND RESULTS

The results are presented beginning with the results for the HEI (2006) method together with the validation of the software Gate Cycle™. After these results, the ones for the Kakaç and Liu (2002) method are presented. Finally the optimization results are shown. The optimization with the EES genetic algorithm was done in two stages, first we minimized the area changing the material for another with better thermal conductivity. In the second stage we minimized the area considering a cost restriction, which was possible only for the Kakaç and Liu (2002) method because in the HEI (2006) method the minimum area always converges to the material with the best thermal conductivity (F_M factor).

4.1 The HEI (2006) method

Figure 1 shows the condensing process that was simulated in the software Gate Cycle™ from the data presented in Tab. 1. In this figure it is possible to see all pressures, temperatures, mass flow and enthalpies for the steam and the water streams. The calculation with the EES model developed for the HEI (2006) method result in a surface heat transfer area of 9731 m², while the values with the Gate Cycle™ was different only in 0,0034 %. This way, it is possible to conclude that the developed model for the HEI (2006) method is valid.

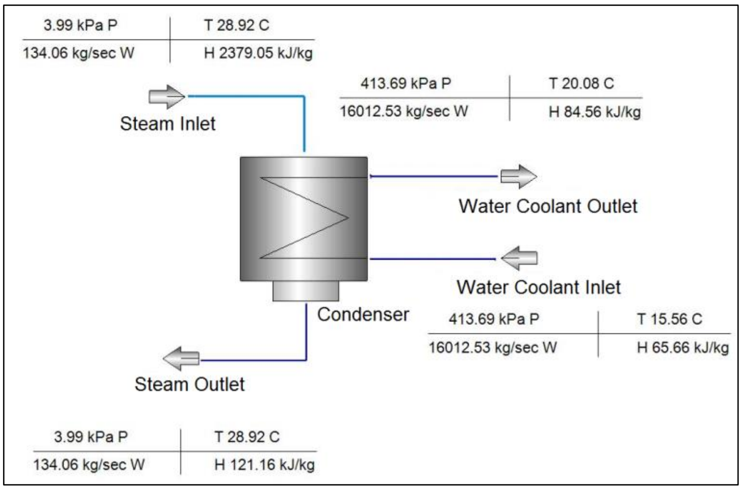

Figure 1. Validation results with the software Gate Cycle™.

Figure 2 shows the variation of the condenser heat exchange surface area due to the coolant velocity, varying the tube gauge (BWG), changing, therefore, the factor F_M. The results were obtained for a tube outside diameter and a material shown in Tab. 1. Once the working fluid exchanges heat with the tube and these with the cooling water through convection, hence, reducing tube gauge (increasing BWG factor) causes the F_M factor to increase, thus, thinner walls provide better heat exchange. From this point of view, tube material and wall thickness are important variables to be considered and chosen aiming at high F_M coefficients, consequently, lower area values.

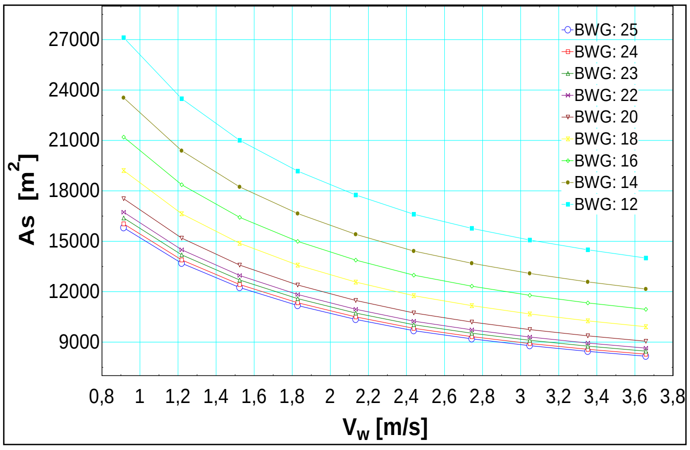

Figure 2. Surface area for different tube gauges.

Figure 3 shows the variation of the condenser heat exchange surface area due to the coolant velocity, varying the tube outside diameter, changing, therefore, the factor U_1. The results were obtained for a tube gauge and a material shown in Tab. 1. With higher diameters, the amount of fluid that flows in the center of the tube will lose heat flux, transferring less heat due to a bigger part of the fluid not being in direct contact with the tube wall, providing a slower heat exchange. In this way, lower values of outside the tube diameter contribute for better values for heat exchange and, thus, a higher U_1 coefficient, requiring less area.

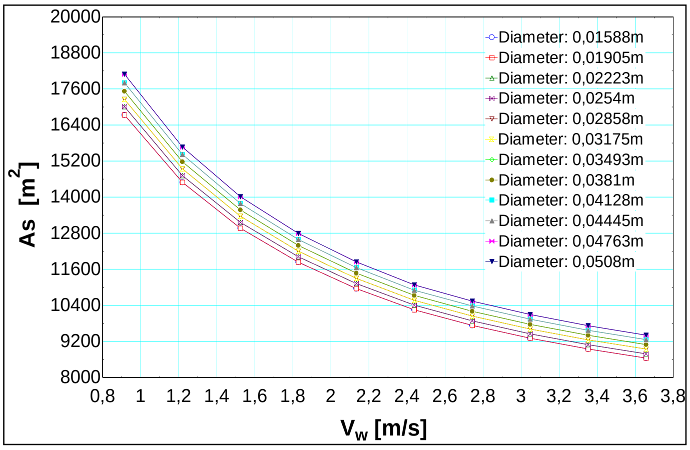

Figure 3. Surface area for different outside tube diameters.

It is remarkable that in the last two figures previously shown the area curves decrease with increasing of the cooling water velocity inside the tubes, hence, with a large amount of fluid in direct contact with the tubes, the heat transferred will be increased.

4.2 Kakaç and Liu (2002) method

From Fig. 4 is possible to see that with the outside diameter increasing, the area of heat exchange increases as well. As noted before in the HEI method, higher outside diameters values cause larger amounts of fluid flowing in the middle of the stream, transferring less heat. In the Kakaç and Liu (2002) method, a higher outside diameter raises the total thermal resistance, and also reduces the global heat transfer coefficient causing the increasing of the surface area of heat transfer.

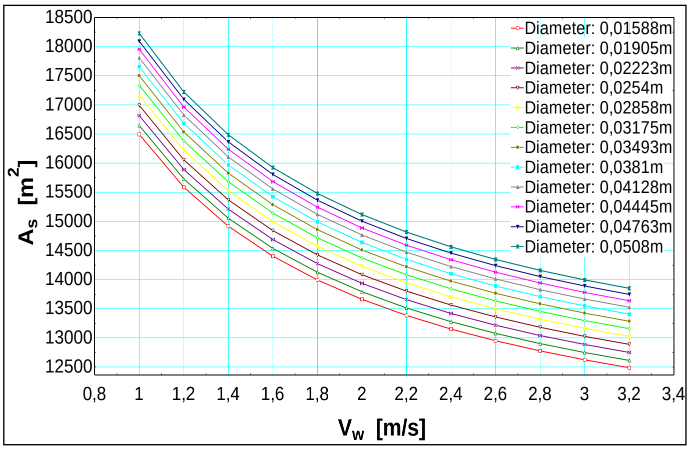

Figure 4. Surface area for different outside tube diameters.

Figure 5 shows the behavior of the surface area of heat transfer as function of the cooling water velocity inside the tubes with the variation of the tube gauge only. As the tube gauge increases, there is an increasing of the surface area of heat transfer. In addition, as seen before in the first HEI (2006) method, due to the process of heat transfer between the cooling fluid and the working fluid across the tube, as the tube wall is getting thinner, the heat exchange will occur easily between the fluids. This is possible because the tube wall works as a heat transfer barrier, slowing the heat transferred. Analyzing the equations according to the Kakaç and Liu (2002) method, with the increasing of the tube gauge the total thermal resistance will increase as well, reducing the global heat transfer coefficient and providing a larger surface area of heat transfer.

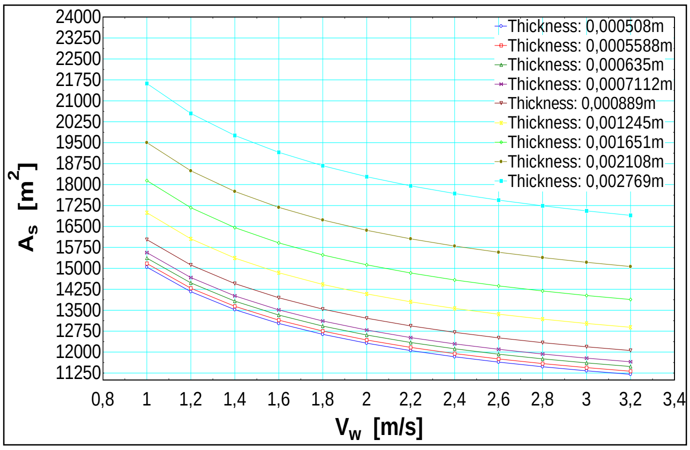

Figure 5. Surface area for different tube gauges.

Figure 6 shows the behavior of the surface area of heat transfer in function of the cooling water velocity inside tubes with the variation of the tube construction material only. Materials with high thermal conductivity like cooper tend to reduce the surface area of heat transfer. The material with high thermal conductivity reduces the total thermal resistance, increasing the global heat transfer coefficient with a fall of the surface area of heat transfer.

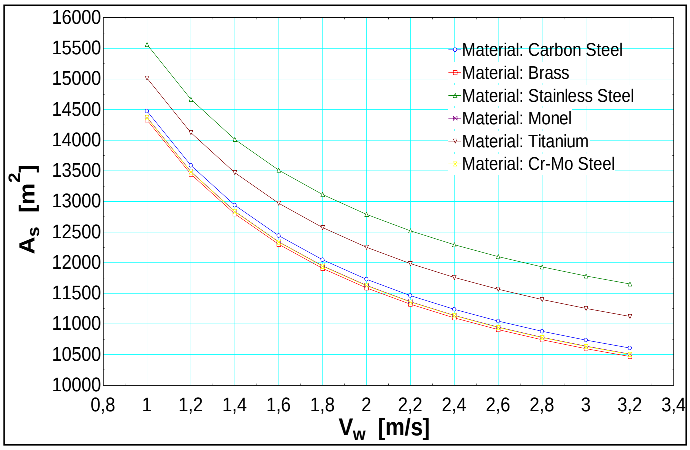

Figure 6. Surface area for different tube material.

4.3 Optimized results analysis

The optimization results were summarized in Tab. 2. For the HEI (2006) method the minimum surface area of heat transfer obtained through modeling was 7082 m². The reference parameters attained for such area were the cooling water velocity of 3.7 m/s, the outside tube diameter of 0.0159 m and the material correction factor of 1.042. The minimum cost was US$ 604,875.00. With optimization, the area was reduced in 37% from the initial value of 9731 m². The initial cost was around US$ 1,221,000.00.

For the Kakaç and Liu (2002) method the minimum surface area of heat transfer obtained through modeling was 9486 m². The reference parameters attained for such area were the cooling water velocity of 3.7 m/s, the outside tube diameter of 0.0159 m, the tube gauge of 0.000508 m, the number of tube rows equals to 30 and used Brass as the tube material. The cost for this parameter was US$ 712,069.00. With optimization, the area was reduced in 26% from the initial value of 11976 m². The initial cost was around US$ 1,385,000.00.

A further optimization was done for the Kakaç and Liu (2002) method intending to restrain the cost to a maximum of US$ 600.000,00, thus, the minimum surface area of heat transfer found was 9585 m² along with an approximately cost around US$ 287.099,00. The operational and design parameters were the same aforementioned except for the tube material that was changed for carbon steel. With the results obtained with the restriction cost for the optimization, it was noticed that the surface area of heat transfer suffered a reduction around 26%, changing from 11976 m² to 9486 m², or 9585 m², for non-restrained and restrained cost, respectively, where the difference between both is just 0,01%, but the difference in the cost is expressive, raging about 148% from the more expensive to the cheaper, and therefore, the restrained cost surface area of heat transfer calculated showed best cost/benefit over the other.

Table 2. Summary of optimization results.

Method	Comments	Area [m²]	Cost [US$]	Tube material
HEI (2006)	Area before optimization	9731	1,221,000.00	AISI 316
	Best minimized area	7082	604,875.00	Brass
Kakaç and Liu (2002)	Area before optimization	11976	1,385,000.00	AISI 316
	Best minimized area	9486	712,069.00	Brass
	Minimum cost with restriction optimization in US$ 600.000	9585	287,099.00	Carbon Steel

5. CONCLUSIONS

For the HEI method the optimization resulted in a reduction of surface area of heat transfer about 37% and cost reduction around 102% varying the input parameters, the optimization algorithm set lower tube diameter and gauge values, and defined brass as new tube construction material. For the Kakaç and Liu method the optimization is a reduction of surface area of heat transfer in 26% without cost restriction, with values of cost and surface area of heat transfer around US$ 712.069,00 and 9486 m², respectively. With restrained cost in US$ 600.000,00, the minimum surface area of heat transfer found was 9585 m², only 0,01% higher than the non-restrained area, with cost equivalent to US$ 287.099,00, hence, a cost was 148% smaller than the one obtained for the minimum surface area of heat transfer. Comparing both methods without optimization, Kakaç and Liu method results in surface area of heat transfer and cost greater in 23% and 13%, respectively, if compared with the results for the HEI method. With the Kakaç and Liu method the values predicted for cost are subject to more details in the thermal resistance calculations, with higher consistency in the result of surface area of heat transfer.

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