Determination Global Heat Transfer Coefficient in Shell and Tube Type and Plates Heat Exchangers

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THEORETICAL AND EXPERIMENTAL SIMULATION FOR DETERMINATION OF THE OVERALL HEAT TRANSFER COEFFICIENT

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ABSTRACT

The literature on heat exchangers is very wide because of numerous existing configurations, several types of fluids used, as well as the variety of applications. On the other hand, when we need to calculate a heat exchanger, a similar procedure is hardly found. Therefore, we propose this educational work in order to facilitate the calculation procedures, when the student or the professional in the area needs a script for its design. The heat exchanger is installed in the Laboratory of Thermal Systems and Biodiesel at UNIMAR - University of Marilia - SP. In the theoretical simulation we used the EES Software. The results for the turbulent regime were as expected , while the values obtained for the laminar regime will be analyzed afterwards.
Keywords: overall heat transfer coefficient, heat exchanger, errors analysis, Brazil, Unimar.

1. INTRODUCTION

The overall heat transfer coefficient depends on the construction, operation, duration of use, type of heat exchanger, properties and fluid flow.
The construction of a heat exchanger with thin tubes and high values of thermal conductivity, coupled with turbulent flow, keeping the energy loss in acceptable values are desirable factors for better performance of a heat exchanger. Viscous fluids, fouling on the walls of the exchanger and undesirable presence of gases within the liquid (mainly the air), decrease the efficiency of the exchanger and consequently there is also a decrease in the overall coefficient. The air form films that act as excellent thermal insulators.
The heat exchangers lose efficiency over time due to the formation of deposits in the tubes which act as an insulator. It depends on:
• the time which the exchanger is in operation;
• the nature of the fluid;
• the flow rate (high speed delays the formation of deposits due to the fluid drag).
The global resistance is the sum of multiple individual resistances of heat transfer, which are:
• resistance to external convection;
• resistance to internal convection;
• resistance to external fouling ;
• resistance to internal fouling ;
• thermal resistance due to the conduction of the wall material in the tube;
The overall heat transfer coefficient, according to Incropera and Dewitt (2008), is calculated by the Equation (1):

$$Q = U.A.\Delta TML \tag{1}$$

The heat flow between two fluids can be schematized by the illustration, Figure 1.

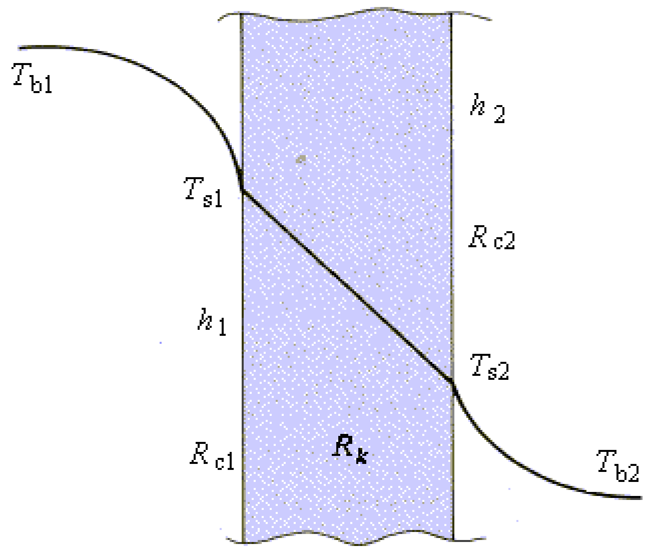

Figure 1: temperature profile through a wall between two fluids

Originating the equivalent thermal circuit, Figure 2.

Figure 2: Equivalent Thermal circuit

The heat exchanged between the fluids across the surface of the tubes can be obtained considering the thermal resistances, Figure 3.

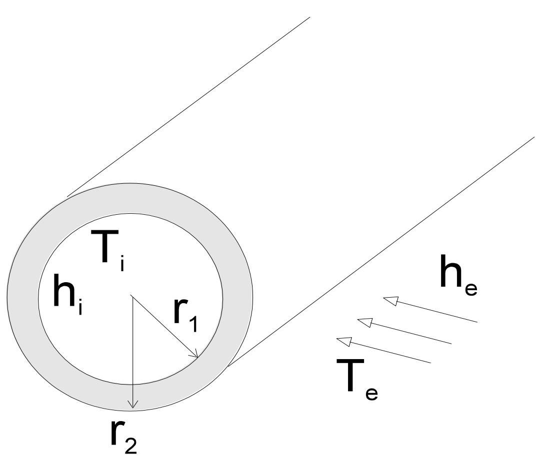

Figure 3: thermal resistance due to conduction and convection.

Thus, the Equation (2) for the heat flow, according to Stoecker and Jones (1985), is as following:

$$Q = \frac{\Delta TML}{R_t} = \frac{\Delta TML}{\frac{1}{h_i.A_i} + R_{cond} + \frac{1}{h_e.A_e} + R_{inc\_1} + R_{inc\_2}} \quad (2)$$

For a heat exchanger of parallel streams, the input is obvious. However, for exchangers of opposing or cross flows, the situation is somewhat more complex, Figure 4. Therefore, it is common to use Equation (3) for calculating the Δ TML, as it is given:

$$\Delta TML = \frac{\Delta T_{máxima} - \Delta T_{mínima}}{\ln\left(\Delta T_{máxima} / \Delta T_{mínima}\right)} \quad (3)$$

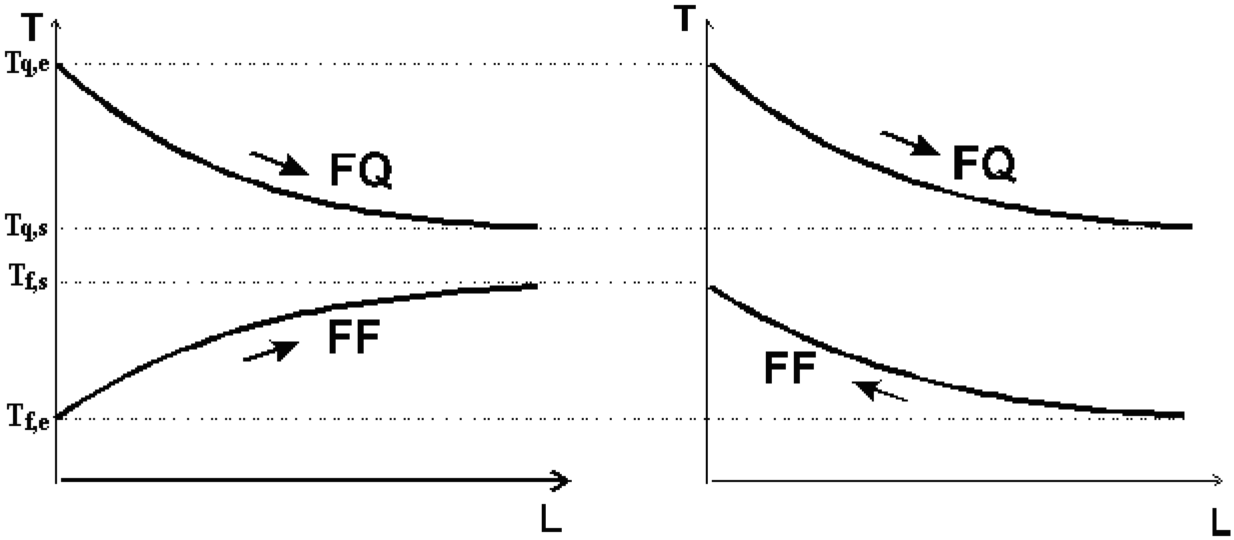

Figure 4: Configuration of heat exchangers for calculating the ΔTML.

The heat flow in a heat exchanger for the shell and tube type is as in Equation (4)

$$Q = U.A.F.\Delta TML \quad (4)$$

The values for Factor F are obtained in function of the S and R dimensionless ratios, calculated by Equation (5). For each configuration of exchanger, there is an abacus such as the type shown in Figure 5.

$$S = \frac{t_2 - t_1}{T_1 - t_1} \quad \text{e} \quad R = \frac{T_1 - T_2}{t_2 - t_1} \quad (5)$$

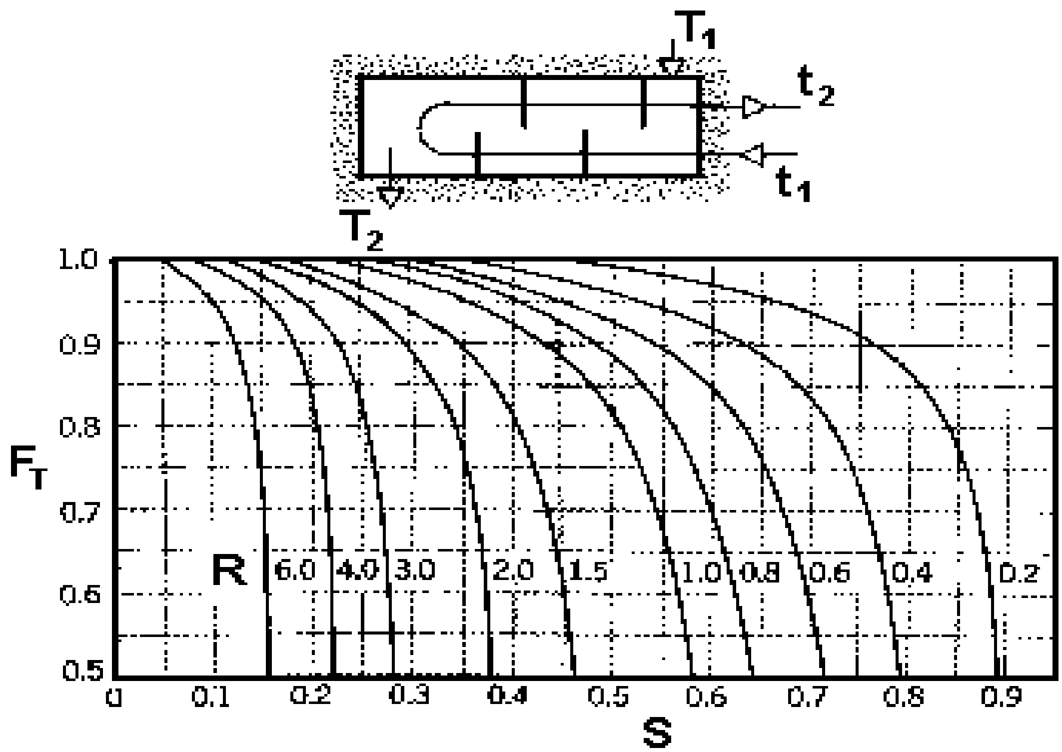

Figure 5: F factor in function of dimensionless ratios S and R.

2. FLOW WITHIN TUBES

The heat transfer inside a tube is classified in accordance with the flow regime of three kinds whose limits of each type is defined by the Reynolds number.
• Laminar regime Re <2100
• Regime of Transition 2100 <Re <3500
• Turbulent Regime Re> 3500
In the laminar regime, heat transfer happens mainly because of conduction between the layers of the fluid. In the transition and turbulent regimes, heat transfer occurs mainly by forced convection. The higher the turbulence, the better the heat transfer will be. Thus, baffles are used in the heat exchangers in order to promote this turbulence.
The Reynolds number, which is dimensionless, is calculated by Equation (6):

$$\mathrm{Re} = \frac{\rho V D}{\mu} \tag{6}$$

2.1 Calculation of forced convection coefficients within the tubes

The overall heat transfer coefficient , based on the external area, Ue, is given by Equation (7):

$$\frac{1}{U_e.A_e} = \frac{1}{h_i.A_i} + R_{cond} + \frac{1}{h_e.A_e} + R_{inc\_1} + R_{inc\_2} \tag{7}$$

2.1.1 Laminar flow within tubes

For the flow in the laminar regime within tubes, the equations (8), (9) and (10), listed below, according to Singh and Heldman (2009), are applied for calculating the heat transfer coefficient by means of convection.

$$\mathrm{Nu} = 3{,}66 + \frac{0{,}085.\mathrm{Gz}}{1 + 0{,}045.\mathrm{Gz}^{2/3}} \left(\frac{\mu}{\mu_w} \right)^{0{,}14} \quad \text{para}\, \mathrm{Gz} < 100 \tag{8}$$

$$\mathrm{Nu} = 1{,}86.\mathrm{Gz}^{1/3}.\left(\frac{\mu}{\mu_w} \right)^{0{,}14} \quad \text{para}\, \mathrm{Gz} > 100 \tag{9}$$

$$\mathrm{Gz} = \mathrm{Re}.\mathrm{Pr}.\frac{D}{L} \tag{10}$$

The fluid properties must be measured at an average temperature of the mixture. The formulas may be used for non-circular ducts by applying the equivalent hydraulic diameter.

2.1.2 Turbulent flow within tubes

The generic formula applied to the turbulent flow of fluid within tubes, considering the actual conditions in the layer resistance, according to ASHRAE (2004), is the Equation(11):

$$\mathrm{Nu} = 0{,}023.\mathrm{Re}^{0{,}8}.\mathrm{Pr}^{1/3}.\left(\frac{\mu}{\mu_w} \right)^{0{,}14} \tag{11}$$

The fluid properties must be measured at an average temperature of the mixture. The formulas may be used for non-circular ducts by applying the equivalent hydraulic diameter.

3. EXPERIMENTAL WORK

The experimental procedure is to establish flow rates set for Steam Heating Systems and cooling water. Measuring pressure and temperature of the steam (inlet and outlet), water temperature (inlet and outlet) of both fluids flow.

The heat exchanger used in the experiments is the shell and tube type, with one-pass shell side, two-pass tube side, according to the scheme below, Figure 6:

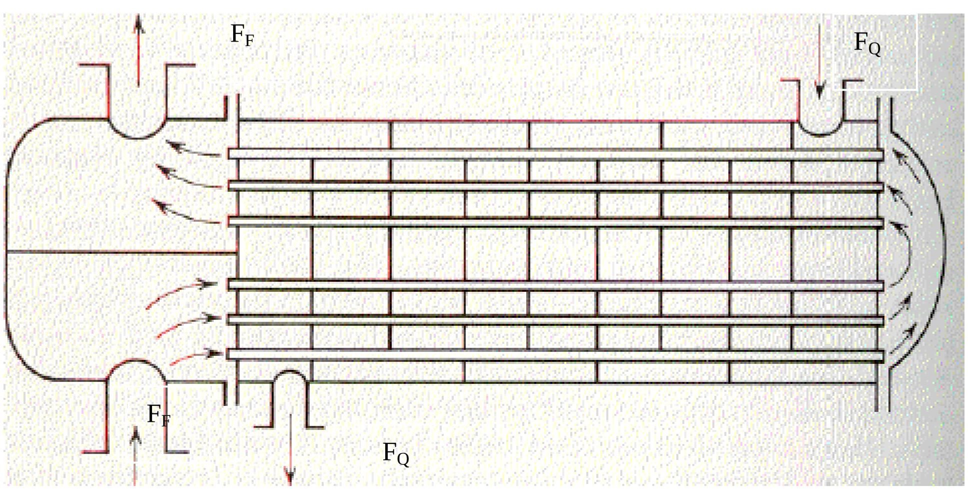

Figure 6: Heat exchanger shell and tube type

The measuring instruments used to evaluate the thermodynamic properties were: Resistance Temperature Detectors (RTDs), pressure transducer to evaluate the steam pressure, Differential Pressure Transducer associated with a measuring device for evaluating the venturi type flow of steam.
The experiments are to determine the heat flux, the overall heat transfer coefficient and mass flow rate of steam flow for many settings, as design of measurement systems of the experimental apparatus, Figures 7 and 8:

Figure 7: Experimental Apparatus

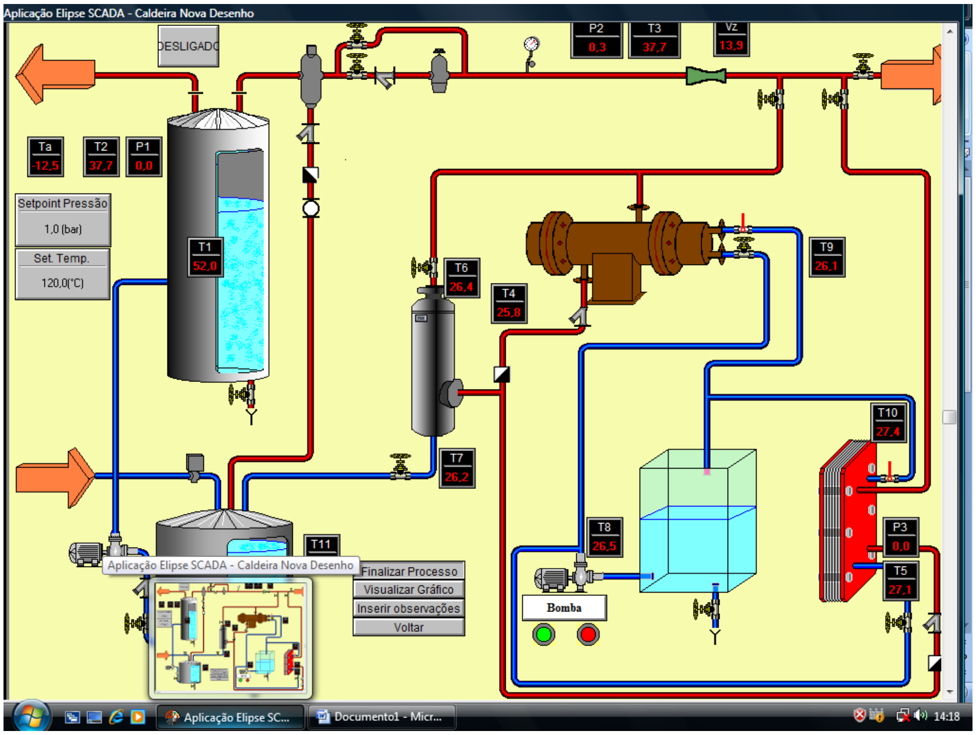

Figure 8: **Design of** measurement **systems** shown in the s**creen computer** monitor

Variables shown in the s**creen computer** monitor:
Ta is the ambient temperature
T1 is the liquid water temperature inside the boiler
T2 is the steam temperature in the boiler
T3 is the steam temperature inlet piping
T4 is the condensate temperature
T5 is the steam temperature inlet heat exchanger plates of the type
T6 is the steam temperature inlet flash tank vapor
T7 is the condensate temperature outlet flash tank vapor
T8 is the water temperature inlet heat exchanger shell and tube
T9 is the water temperature outlet heat exchanger shell and tube
T10 is the water temperature inlet heat exchanger plates
T11 is the liquid water temperature inside the boiler feed reservoir
P1 is the steam pressure inside the boiler
P2 is the steam pressure outlet of the boiler
P3 is the condensate pressure outlet heat exchanger
Vz is the steam flow meter (apply multiplier to $5,7.10^{-5}$)

4. MEASUREMENT PROCEDURES AND ERROR ANALYSIS

An analysis of errors following criteria has been adopted in accordance with information provided by each measuring instrument:

- Temperature: + 0.5 ° C;
- Pressure: + 2%;
- Vernier caliper to measure the length: + 0.05 mm;
- Time measurements: + 0.1 s;
- Mass measurements: + 0.1 g.

We can generalize the propagation of errors theory for a given function using the differential calculus. F is a simple function in the range of interest and slow variation as the same of Equation (12):

$$\mathrm{F} = \mathrm{f}(\mathrm{x}) \tag{12}$$

In order to determine the limiting F, calculate Fi for each component, ie, expand Fi and then calculate the average value of F and its deviation. F **can** be prescribed in the form of Equation (13):

$$F = f(\bar{x} \pm \delta x) \tag{13}$$

where δ x symbolizes a mean deviation or pattern. Expanding in Taylor series, we have Equation (14):

$$f(\bar{x} \pm \delta x) = f(\bar{x}) \pm \left.\frac{df(x)}{dx}\right|_{x=\bar{x}} \delta x + \left.\frac{d^2 f(x)}{2dx^2}\right|_{x=\bar{x}} (\delta x)^2 \pm .... \tag{14}$$

After neglecting **second-order terms**, one can determine the deviation of the function in Equation (15):

$$\delta f = \left|\frac{df}{dx}\right| \delta \tag{15}$$

This simplification is implied that the function is monotonous and has a slow variation. It is the same as considering that the confidence interval of x, in the graph, that is **taken** to be a **straight-line function** . The derivative should be calculated for x = x in absolute value, so we can express F in the confidence interval, according to Equation(16):

$$F = \overline{F} \pm \delta F \text{ onde } \overline{F} = f(\bar{x}) \text{ e } \delta F = \delta f \tag{16}$$

Generally, the function of several variables based on the total differential function can be written to Equation (17):

$$F=f(x,y,z,....) \tag{17}$$

The differential equation of f is given by Equation (18):

$$dF = \frac{\partial f}{\partial x}\delta x + \frac{\partial f}{\partial y}\delta y + ... \frac{\partial f}{\partial z}\delta z \tag{18}$$

where dF the infinitesimal change in the function F and new coordinate system with origin at this (x,y,z) and call the new coordinates dx, dy, and dz.
Assuming that x, y and z are measured, and replacing the differences by their infinitesimal deviations δ x, δ y and δ z, then the last expression will be the formula for the deviation of F. The maximum deviation occurs when all the partial contributions occur in the same direction of growth. Therefore, we take the partial derivatives absolute value, as Equation (19), ie:

$$\delta F = \left|\frac{\partial f}{\partial x}\right|\delta x + \left|\frac{\partial f}{\partial y}\right|\delta y + ... \left|\frac{\partial f}{\partial z}\right|\delta z \tag{19}$$

Thus, recalling that is an approximate expression for the terms have been discarded from the second order function F can be written as Equation (20):

$$F = \overline{F} \pm \delta F \tag{20}$$

Thus, it is possible to determine the limiting values in the calculation of steam flow rate in each estimate of the same.
For the mass balance in the heat exchanger, one can write Equation (21):

$$m_S = \frac{m_w c_{p_w}\left(T_9 - T_8\right)}{\left(h_s - h_c\right)} \tag{21}$$

Therefore, the steam flow rate depends of water flow, the water temperature difference, the properties of water and the difference between enthalpy . Thus, the maximum deviations for calculation the mass flow are given by Equation (22):

$$\delta m_S = \left|\frac{\partial m_S}{\partial m_w}\right|\delta m_w + \left|\frac{\partial m_S}{\partial c_{p_w}}\right|\delta c_{p_w} + \left|\frac{\partial m_S}{\partial \Delta T_w}\right|\delta \Delta T_w + \left|\frac{\partial m_S}{\partial \Delta h}\right|\delta \Delta h \tag{22}$$

The uncertainty in each input variable is expressed by Equations (23), (24), (25) and (26):

$$\delta m_w = \delta m_w \pm 0{,}02 \tag{23}$$

$$\delta c_{p_w} = \delta c_{p_w} \pm 1{,}5 \tag{24}$$

$$\delta \Delta T_w = \delta \Delta T_w \pm 0{,}5 \tag{25}$$

$$\delta\Delta h = \delta\Delta h \pm 1{,}017 \qquad (26)$$

and steam mass flow in the balance of the heat exchanger is given by Equation(27):

$$m_S = \overline{m_S} \pm \delta m_S \qquad (27)$$

The bounds for the steam mass flow calculated by different tests, using the process discussed above, is given by Equation (28):

$$m_S = \overline{m_S} \pm 7{,}5\% \qquad (28)$$

5. EXPERIMENTAL RESULTS

The experimental results were within expectations. The overall heat transfer coefficient calculated in the tests, presented an **average error** rate of 15% compared with the value calculated by mass balance in the heat exchanger, as shown in Figure 9. The error analysis of the measures showed a maximum variation of 7.5%, according to section 5 of this work.

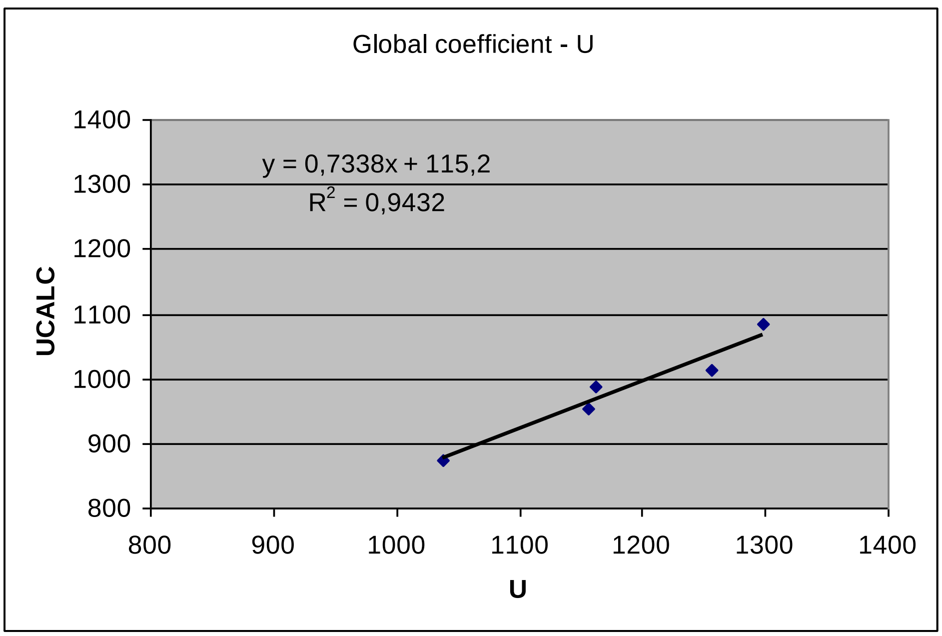

Figure 9: Overall heat transfer coefficient (Ucalc) compared with the theoretical value obtained by mass balance in the heat exchanger (U)

Moreover, the assessment of the overall heat transfer coefficient calculated without taking into account the scale on the tube wall, showed very similar results compared with the value calculated by mass balance in the heat exchanger, Figure 10. The dispersion has an error average value of 2%. This suggests that the maintenance of the heat exchanger, which is performed periodically, is made properly.

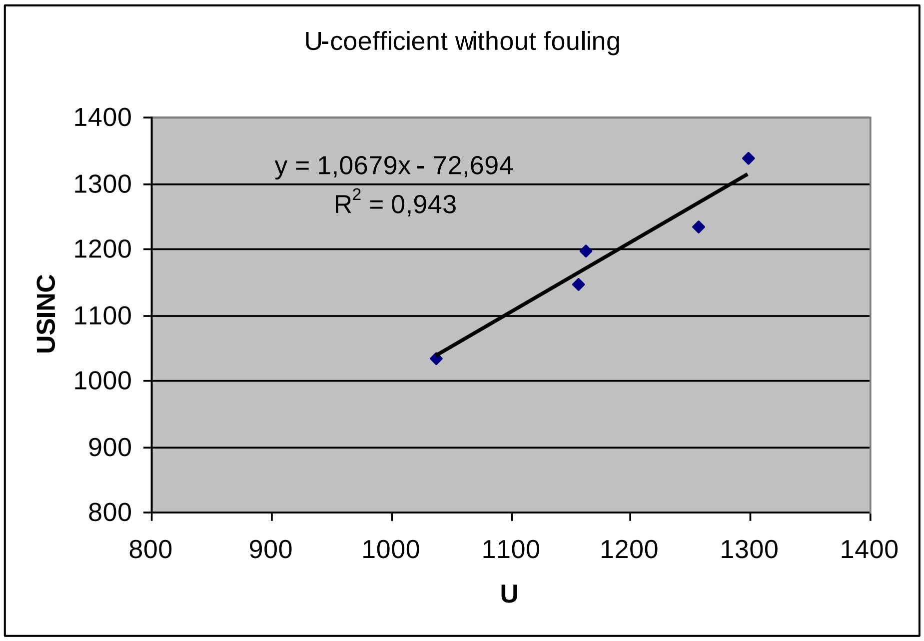

Figure 10: Overall heat transfer coefficient (without fouling, Usinc) compared with the theoretical value obtained by mass balance in the heat exchanger (U)

The steam mass flow (Ms) is calculated by mass balance in the heat exchanger, while the value Vz is provided by the data acquisition software. The value of Vz must be corrected by the coefficient of flow, provided by the installer, who made the comparison with a calibration standard meter. This weighting factor has the value of $5,7.10^{-5.}$ Comparing the values obtained with the mass balance provided by the software revealed that there is a perfect agreement, with a mean error of 2.8%, as shown in Figure 11.

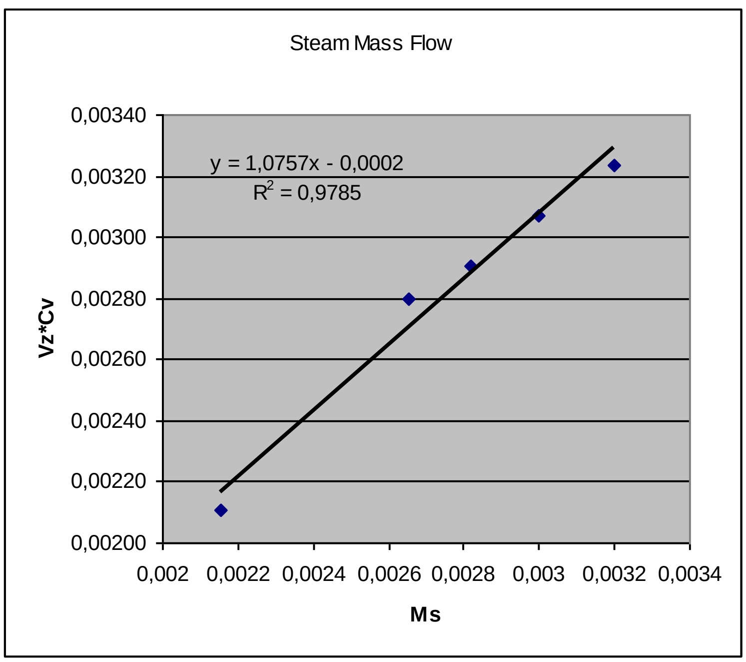

Figure 11: Correlation measuring the mass flow (Vz) compared with the value obtained by the mass balance in the heat exchanger (Ms)

6. CONCLUSION

The calculation of the overall heat transfer coefficient presents various methodologies, such as the LMTD and the effectiveness method. We presented the LMTD method, because the implementation, is more simpler than effectiveness method., due to convergence around the Logarithmic Functions.
We hope that this work is useful for those who want to improve the knowledge about the subject. The results presented were within the expected theoretical analysis. This work continues with a focus of study within the laminar regime and also in order to make comparisons with other heat exchangers such as the type plates.

NOMENCLATURE

Symbol - Measure Greatness
A is the area of heat exchange (always perpendicular to the direction of fluid flow), m^2;
Ai and Ae are the areas of heat exchange inside and outside, respectively, m^2.
D is the diameter of the tube, m;
ΔTML is the logarithmic mean temperature difference, K
Gz is the Graetz, Number (-);
hi is the coefficient of heat exchange by convection within the tube; $W/m^2.K$;
he is the coefficient of heat exchange by convection outside the tube $W/m^2.K$;
L is the length of the tube, m;
μ is the fluid viscosity, Pa.s;
μw is the viscosity of the fluid measured at the temperature of the tube wall, Pa.s;
Nu is the Nusselt Number, (-)
PR is the Prandtl number, (-);
Q is the flow across the heat exchanger, W;
Rcond is the thermal resistance due to heat conduction of the tube wall material , K / W;
Re is the Reynolds number, (-).
Rinc_1 is resistance to fouling on inner tube wall, K / W;
Rinc_2 is the resistance to fouling on the outside tube wall, K / W;
RT is the global resistance to heat transfer, K / W
ρ is the fluid density, kg/m^3;
t1 is the temperature of the fluid inlet tube, K
t2 is the temperature of the fluid outlet tube, K
T1 is the inlet temperature of the shell-side fluid, K
T2 is the outlet temperature of the shell-side fluid, K
U is the overall heat transfer coefficient, $W/m^2.K$;
V is the fluid velocity, m / s;

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