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ORIGINAL RESEARCH ARTICLE

Determination of overall heat transfer coefficients comparing LMTD and ε-NTU methods

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ABSTRACT

Thermal energy transfer processes are important problems to be solved in the field of engineering. In this field, heat exchangers are one of the most used equipment in the industry. The present investigation was carried out in an operating hydrogen sulfide cooler system, with the objective of determining the overall heat transfer coefficients by two methods, applying the passive experimentation procedure. With the Logarithmic Mean Temperature Difference (LMTD) method, values ranging from 11.1 to 73.3 W/(m^2·K) were obtained, compared to 11.0 to 58.9 W/(m^2·K) when applying the Effectiveness-Number of Transfer Units (ε-NTU) method. Although the results obtained were similar, for the thermal evaluation of the chiller system studied, it was recommended to employ the LMTD approach, used by most researchers.

Keywords: Heat Exchanger; Hydrogen Sulfide; Overall Heat Transfer Coefficient; LMTD; ε-NTU

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1. Introduction

Heat exchangers are present in most complex thermal systems and are the most widely used device for non-combustion heat transfer in industrial processes. They are used in chemical processing plants, steam generation, heating and air conditioning, food preparation, refrigeration, among other applications. Monitoring of their optimum operating parameters ensures process economy[1-3].

There are several criteria for evaluating the performance of heat exchangers. Of these, the behavior of the overall heat transfer coefficient over time is considered a reliable parameter to determine how quickly the conditions favoring heat exchange deteriorate[4,5]. Moreover, its prior calculation is necessary to determine the fouling factor and impact of depositions on the efficiency loss of the installation[6,7].

For the determination of global heat transfer coefficients from experimental data, the Logarithmic Mean Temperature Difference (LMTD) method is commonly used[2,7,8]. The calculation is straightforward, although for multipass heat exchangers (countercurrent-parallel) the LMTD correction factor must be considered, which leads to an extensive expression involving several parameters. Of the references consulted, only Gudmundsson[5,9] has used the Effectiveness-Number of Transfer Units (ε-NTU) method to experimentally determine the overall heat transfer coefficients, whose calculation is supported by published NTU ratios for different heat exchanger configurations[5,9].

Although it is known that the LMTD and ε-NTU methods share common parameters and concepts that arrive at a similar solution with respect to the thermal capacity of the equipment, few researchers have performed a detailed comparison of both procedures. The classical literature is limited to state that the LMTD approach is useful when the inlet and outlet temperatures of the fluids are known or can be easily determined, because otherwise the calculation involves an iterative trial-and-error process. In these cases, the analysis is more easily performed using the ε-NTU method, based on the performance or effectiveness of the heat exchanger during the transfer of a given amount of thermal energy[10-12].

Jeter[13] presented the theoretical foundations of three conventional methods for analyzing cross-flow heat exchangers for pedagogical purposes. According to the author, the Mean Temperature Difference (MTD) approach is practically obsolete. The LMTD method is preferred for sizing heat exchangers, while the ε-NTU is selected for performance analysis and simulation works[13]. Another study was conducted by Ramana and Sudheerpremkumar[14] with the purpose of calculating the effectiveness in a double tube heat exchanger and comparing the results of the LMTD and ε-NTU methods with the graphically determined values. Although they conclude that the results obtained are equivalent, the LMTD method provided better approximation for the countercurrent arrangement of the fluids, the opposite occurring for the parallel arrangement of the streams. The evaluation was performed for a single data set, without modifying any of the independent variables[14]. Although the methodology for the analysis of two-fluid heat exchangers has been established, the literature consulted does not refer to the study of jacketed shell and tube heat exchangers, where three fluids interact. When the specialists of the production plants need to perform a thermal evaluation of this type of heat exchangers, they do not know which method to use to obtain accurate results.

Considering the above, the objective of this research is to determine the overall heat transfer coefficients in a system of hydrogen sulfide coolers in operation, establishing a comparison between the LMTD and ε-NTU methods. The heat exchangers under study have industrial use in high purity hydrogen sulfide production plants and in sulfur recovery units from the conversion of the above mentioned chemical reagent (Claus process).

2. Materials and methods

2.1 Methodology

The evaluation of the hydrogen sulfide cooling process in jacketed shell and tube exchangers was performed by analyzing the overall heat transfer coefficients, determined from experimental data. The LMTD and ε-NTU methods were used for the calculation.

In heat exchangers with three fluids and two main heat exchange paths, two global heat transfer coefficients are determined[15,16]. One characterizes the internal heat exchange, between the fluid flowing through the tubes and the fluid flowing through the shell; while the other corresponds to the external exchange, between the fluid flowing through the shell and the fluid flowing through the jacket.

For simplification of the calculations, the following assumptions were made[10,17]:

- Heat exchangers operate under steady state conditions.
- The overall heat transfer coefficients, as well as the specific heat of each fluid, remain constant throughout the heat exchanger.
- Heat transfer to the environment is neglected.
- Potential and kinetic energy changes are negligible.
- Heat transfer by longitudinal conduction in the fluids, and in the wall of the tubes and shell, is negligible.
- There are no phase changes.
- During the same operating shift, the hydrogen sulfide flow is constant.
- Measurements of the flow rate of water circulating on the tube side and jacket side were made on common branches, so half

the flow rate is assumed for each exchanger, assuming that the pressure drops in the equipment are similar.

2.2 Description of the installation and experimental technique

The system under study consists of four jacketed shell and tube heat exchangers. Each pair (two units in series) was designed to transfer 138 kW of heat over an area of 49.2 m^2. In each unit, the hydrogen sulfide travels on the shell side, in a single pass, while the water circulates on the tube side, with four passes, and also through the shell jacket. See **Figure 1**. The heat exchangers operate for eight hours in gas cooling mode, and then are taken out of operation to supply steam (tube side and jacket) for four to six hours to remove the sulfur embedded in the gas.

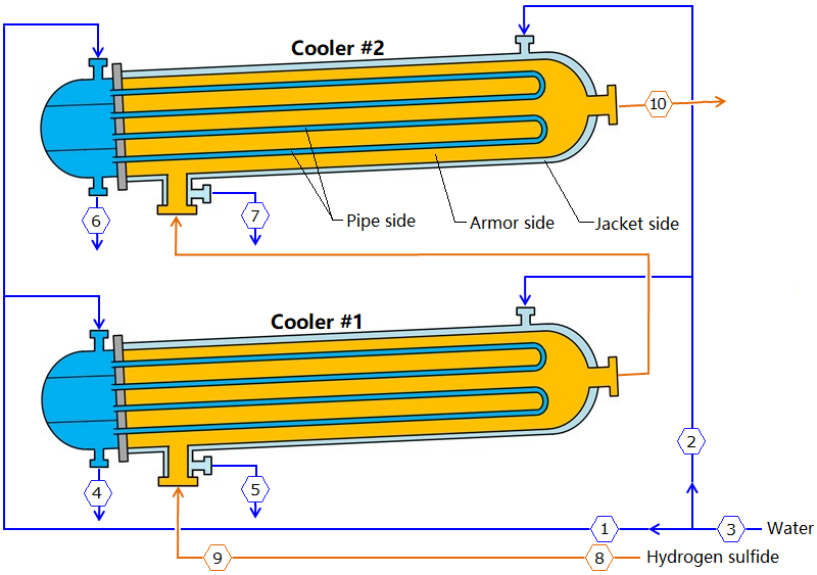


Figure 1. Schematic of a pair of hydrogen sulfide coolers and measurement points.

Due to the uninterrupted production regime in which the object of study is found, a passive experiment was applied (non-experimental research design, of the longitudinal, trend type). This procedure consists in the observation and recording of the input and output variables of the process in the normal working regime of the investigated object, as well as in the observation of the natural arbitrary variations of all technological variables without the active intervention of the researcher in the course of the technological process and without the introduction of preconceived disturbances. Under these premises, the measurements of the fundamental parameters involved in the heat exchange process were performed without manipulation of the variables, analyzing the heat transfer mechanisms as they manifest themselves in their context[18].

The parameters recorded (measurement points as shown in **Figure 1**) are listed below:

- Water flow fed from pipe side.
- Water flow fed from jacket side.
- Water temperature at the inlet of the chiller bank.
- Water temperature through the pipes at the outlet of cooler #1.
- Water temperature through the jacket at the outlet of cooler #1.
- Water temperature through the pipes at the outlet of cooler #2.
- Water temperature through the jacket at the outlet of cooler #2.
- Flow of hydrogen sulfide fed to the coolers.
- Hydrogen sulfide temperature at cooler #1 inlet.
- Hydrogen sulfide temperature at cooler #2 outlet.

The corresponding instruments and their technical characteristics are as follows:

- Temperatures: Ashcroft industrial bimetallic thermowells and thermometers, accurate to 0.1 K.
- Water flow: Proline Prosonic Flow 93T ultrasonic flow meter, accurate to $6.3 \cdot 10^{-6}$ m^3/s.
- Hydrogen sulfide flow: Process signal is sent to a Siemens S7-400 PLC and through Citect SCADA 7.10 the variable is stored and displayed on the control panel computer, accurate to 10^{-4} kg/s.

In the experiment, three observations were carried out on alternate days, during eight hours after the same pair of heat exchangers were put into operation in cooling mode. During each cycle, 20 measurements of the technological variables were taken in each heat exchanger, obtaining a data set with 120 records. In order to reduce random and accidental observation errors (parallax, physical phenomenon and reflection), three replicates were carried out.

The gas temperature at the outlet of cooler #1 was determined by energy balance, as shown in equation (1)[15]. Similarly, the gas temperature at the outlet of cooler #2 was checked using the same equation.

$$T_{b2} = T_{b1} - \left[\frac{\dot{m}_a \cdot Cp_a \cdot (T_{a2} - T_{a1}) + \dot{m}_c \cdot Cp_c \cdot (T_{c2} - T_{c1})}{\dot{m}_b \cdot Cp_b} \right] \quad (1)$$

Where: T [K] is the temperature; $\dot{m}$ [kg/s] is the mass flow rate; and Cp [J/(kg·K)] is the specific heat at constant pressure. The subscripts a , b and c identify the fluids on the tube, shell and jacket side respectively; while 1 and 2 refer to the inlet and outlet conditions of each stream.

2.3 Determination of the overall coefficients using the LMTD method

The determination of the overall heat transfer coefficients, using the LMTD method, is performed by equation (2). The heat transfer area is known by catalog, while the amount of heat transferred during the process, the logarithmic mean temperature difference and its correction factor are calculated from the experimental data: mass flows of each stream, as well as the inlet and outlet temperatures of the fluids[3,7,14].

$$U_{(DTML)} = \frac{Q}{A \cdot \Delta T_{ml} \cdot F} \quad (2)$$

Where: U [W/(m^2·K)] is the overall heat transfer coefficient; Q [W] represents the heat transferred; A [m^2] is the transfer area; ΔT_{ml} [K] is the logarithmic mean temperature difference; and F is its correction factor.

For the internal heat exchange (shell-tubes), the logarithmic mean temperature difference (ΔT_{mli}) is determined by equation (3), based on a multipass equipment. The heat transferred (Q_i) is absorbed by the water circulating on the side of the tubes, and equation (4) is used in its calculation, since no phase changes occur[10].

$$\Delta T_{ml\ i} = \frac{(T_{b2} - T_{a1}) - (T_{b1} - T_{a2})}{\ln\left[(T_{b2} - T_{a1}) / (T_{b1} - T_{a2})\right]} \quad (3)$$

$$Q_i = \dot{m}_a \cdot Cp_a \cdot (T_{a2} - T_{a1}) \quad (4)$$

On the other hand, for external heat exchange (shell-jacket), the logarithmic mean temperature difference (ΔT_{mle}) is calculated by equation (5), established for a heat exchanger with countercurrent flows. In this case, the heat transferred (Q_e) is absorbed by the water on the jacket side, and is determined according to equation (6)[10].

$$\Delta T_{ml\ e} = \frac{(T_{b1} - T_{c2}) - (T_{b2} - T_{c1})}{\ln\left[(T_{b1} - T_{c2}) / (T_{b2} - T_{c1})\right]} \quad (5)$$

$$Q_e = \dot{m}_c \cdot Cp_c \cdot (T_{c2} - T_{c1}) \quad (6)$$

The logarithmic mean temperature difference correction factor is equal to unity (F = 1) for countercurrent or parallel flows. However, in multipass shell-and-tube heat exchangers, it is determined by equations (7) and (8), for any number of shell passages and even number of tube passages, when $R \neq 1$[11,19].

$$F = \left\{\sqrt{R^2+1}\cdot ln\left(\frac{1-S}{1-R\cdot S}\right)\right\}\cdot\left\{(R-1)\cdot ln\left[\frac{2-S\left(R+1-\sqrt{R^2+1}\right)}{2-S\left(R+1+\sqrt{R^2+1}\right)}\right]\right\}^{-1} \quad (7)$$

$$S = \left\{\left[\left(\frac{1-R\cdot P}{1-P}\right)^{1/N}\right]-1\right\}\cdot\left\{\left[\left(\frac{1-R\cdot P}{1-P}\right)^{1/N}\right]-R\right\}^{-1} \quad (8)$$

Where: the parameter *S* is used to simplify the equation for calculating the correction factor; *R* is the ratio between the temperature differences, calculated by equation (9); *P* is the effectiveness of the temperatures, according to equation (10); and *N* is the number of passes through the shell.

$$R = (T_{b1} - T_{b2}) / (T_{a2} - T_{a1}) \quad (9)$$

$$P = (T_{a2} - T_{a1}) / (T_{b1} - T_{a1}) \quad (10)$$

2.4 Determination of the overall coefficients using the ε-NTU method

The determination of the overall heat transfer coefficients, employing the ε-NTU method, is performed by equation (11). The heat transfer area is known, while the minimum thermal capacitance and the ratio of thermal capacitances are calculated from experimental data. The number of transfer units is determined as a function of the exchanger type, thermal effectiveness and the ratio of thermal capacitances[9].

$$U_{(\varepsilon-NUT)} = \frac{C_{min}\cdot NUT}{A} \quad (11)$$

Where: C_{min} [J/(s·K)] is the minimum thermal capacitance; and *NUT* is the number of transfer units. The thermal capacitance of each current is determined through equation (12).

$$C = \dot{m}\cdot Cp \quad (12)$$

In the case of internal heat transfer (shell-tube), the number of transfer units (NUT_i) is calculated based on a multipass heat exchanger, according to equation (13)[10].

$$NUT_i = \frac{1}{\sqrt{1+Cr_i^2}}\cdot ln\left[\frac{2-\varepsilon_i\left(1+Cr_i-\sqrt{1+Cr_i^2}\right)}{2-\varepsilon_i\left(1+Cr_i+\sqrt{1+Cr_i^2}\right)}\right] \quad (13)$$

However, for external heat transfer (shell-shell), when determining the number of transfer units (NUT_e), a countercurrent heat exchanger is considered and equation (14) is used[10].

$$NUT_e = \frac{1}{1-Cr_e}\cdot ln\left(\frac{1-\varepsilon_e\cdot Cr_e}{1-\varepsilon_e}\right) \quad (14)$$

Where: *e* [%] is the thermal efficiency; and *Cr* is the ratio of the thermal capacitances. The subscript *i* refers to the internal heat exchange; while *e* represents the external one.

Thermal efficiency is defined as the ratio between the actual heat transfer magnitude and the maximum possible heat transfer[10-12]. Therefore, for internal heat exchange, it is determined by equation (15), and for external heat exchange by equation (16).

$$\varepsilon_i = Q_i / [C_{mín\ i}\cdot(T_{b1} - T_{a1})] \quad (15)$$

$$\varepsilon_e = Q_e / [C_{mín\ e}\cdot(T_{b1} - T_{c1})] \quad (16)$$

The ratio of the thermal capacitances is calculated by equation (17)[10,11].

$$Cr = C_{mín} / C_{máx} \quad (17)$$

Where: C_{max} [J/(s·K)] is the maximum thermal capacitance.

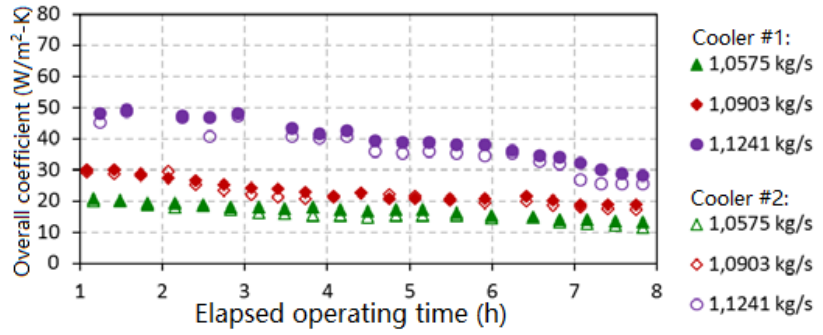
When the ε-NTU method is used to determine the overall heat transfer coefficients based on experimental data, the inlet and outlet temperatures of both fluids must be known or able to be estimated, unlike when the method is used to calculate the heat transferred and the outlet temperatures in the exchanger (*Rating* problem).

3. Results and discussion

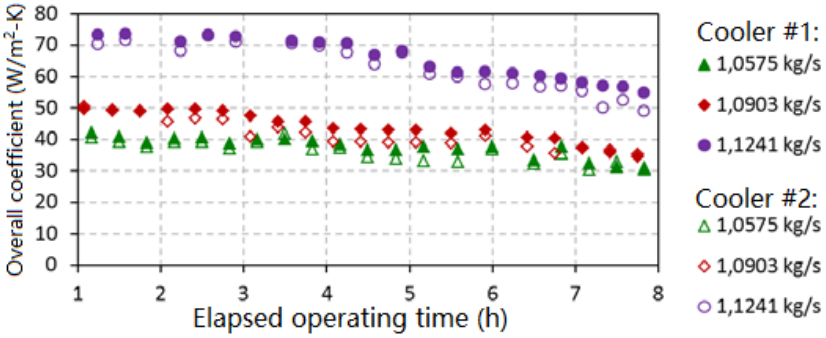
3.1 Application of the LMTD method

Figure 2 shows the behavior of the overall heat transfer coefficients determined by the LMTD method. The calculations were performed for three

hydrogen sulfide flux values.



a) Shell-tube exchange



b) Heart-jacket exchange

Figure 2. Behavior of the overall heat transfer coefficient (LMTD method).

During the eight-hour cooling cycle, a decreasing trend in the overall heat transfer coefficient was observed, mainly due to an increase in sulfur incrustations inside the heat exchangers. Over time, the accumulation of sulfur particles that separate from the gas grows on the walls of the tubes and the shell, forming "insulation" layers on the heat transfer surfaces that act to the detriment of heat exchange and cause a decrease in the overall coefficient between 7.5 and 20.8 W/(m^2·K). Other causes of changes in the overall coefficient are variations in flow and in the thermo-physical properties of the fluids, but their incidence is minor compared to the influence of fouling. It was determined that the changes in pressure and temperature of the fluids, by affecting their thermo-physical properties, cause maximum variations in the global heat transfer coefficient equivalent to 1.1 W/(m^2·K) for the tube-shell heat exchange and 3.6 W/(m^2·K) for the heart-jacket exchange.

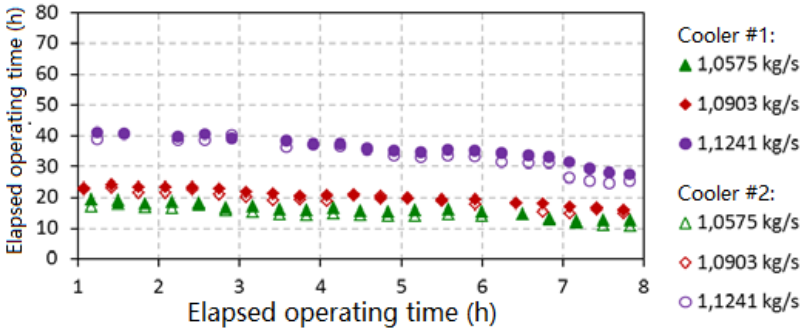
The overall heat transfer coefficient values improve with increasing hydrogen sulfide mass flow rate, since increasing the gas velocity on the shell side increases the individual convective transfer coefficient and decreases the thermal resistance of the scale. Although the highest heat transfer takes place in cooler #1, the behavior of the overall coefficient is similar in cooler #2.

3.2 Application of the ε-NTU method

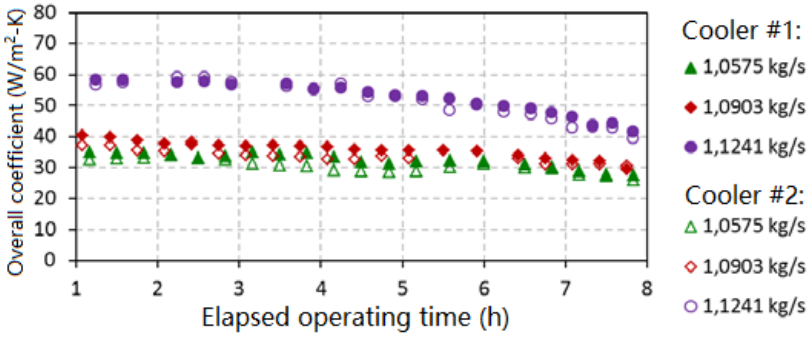
Figure 3 shows the behavior of the overall heat transfer coefficients determined by the ε-NTU method. By applying this procedure (for the same experimental data), values comparable to those obtained using the LMTD method were obtained. The decrease in the overall heat transfer coefficients at the end of the duty cycles ranged from 6.9 to 16.7 W/(m^2·K).

To increase the overall heat transfer coefficients and, consequently, improve the heat exchange process in hydrogen sulfide coolers, it is recommended to: shorten the planned time for the cooling cycle; disassemble the tube bundle of each heat exchanger, perform cleaning and reassemble; or replace the tube bundles in operation with new

units. These actions will help minimize the impact of fouling on the thermal efficiency loss of the facility. It is also suggested to increase the flow of water fed to each cooler above 1.167 kg/s, on the tube side, to reach the turbulent regime (*Re* > 4,000).



a) Shell-tube exchange



b) Heart-jacket exchange

Figure 3. Behavior of the overall heat transfer coefficient (ε-NUT method).

Table 1. Comparison of values obtained by LMTD and ε-NTU methods

Exchange route	Gas flow [kg/s]	$U_{(DTML)}$–$U_{(\varepsilon\text{-NUT})}$ [W/m²·K]		
		Minimum	Average	Maximum
Internal exchange (shell-tubes)	1.0575	0.1	0.9	2.7
	1.0903	0.4	2.8	7.8
	1.1241	0.2	3.4	8.8
External interchange (heart-jacket)	1.0575	2.2	5.4	11.0
	1.0903	3.4	8.0	13.6
	1.1241	7.1	12.2	15.9

Both methods reveal decreasing trend of the overall heat transfer coefficient with the time course, as well as higher values of this parameter with increasing hydrogen sulfide mass flow rate. However, the values of the overall coefficients calculated using the LMTD method are higher than those determined using the ε-NTU method. **Table 1** shows the variations calculated during the quantitative comparison of the two procedures.

Most of the authors who have evaluated the influence of fouling on the efficiency loss of heat exchangers, based on the determination of global coefficients, used the LMTD method[1,2,6,8,12]. Using the global heat transfer coefficients calculated by the ε-NTU method, lower than those determined by the LMTD, leads to the estimation of conservative values of thermal resistance of the fouling, which leads to the oversizing of the installation. In the case of the external heat exchange pathway, the determination of the global coefficients dispenses with the LMTD correction factor, since it is considered as an exchanger with countercurrent flows (F = 1). This makes the calculation based on the LMTD method, according to equations (2) and

(5), more direct and precise when compared to equations (11) and (14), based on the ε-NTU method. The propagation of measurement errors is accentuated with the latter solution. The ε-NTU method is mainly used in heat exchangers with cross-flow, since for this configuration, no analytical expression was ascertained that allows accurately program and determines the LMTD correction factor. An essential requirement for obtaining reliable results using the ε-NTU method is to properly select the function that best characterizes the heat exchanger under analysis. Several authors have established ranges of preliminary values of the overall coefficient in tubular heat exchangers for heat transfer between gases and water, but no study consulted refers to the heat exchange between hydrogen sulfide and water. **Table 2** compares the results obtained with those published by other researchers. These values are used during the evaluation of heat exchangers to make a quick estimate of the required transfer area by clearing in equation (2), so assuming a more accurate value of the overall heat transfer coefficient will improve the accuracy of the calculations.

Table 2. Comparison of the results obtained with other references

Reference	Heat exchanging fluids	U [W/(m^2·K)]
Ludwig (1993)[20]	Gases–water	17.0–284.0
Kern (1999)[17]	Gases–water	11.0–284.0
Kakaç and Liu (2002)[10]	Gases–water	10.0–250.0
Serth (2007)[11]	Air, nitrogen, etc.–water or brine	57.0–454.0
Present research	Hydrogen sulfide–water (LMTD method)	11.1–73.3
	Hydrogen sulfide–water (ε-NTU method)	11.0–58.9

4. Conclusions

Both the LMTD and ε-NTU methods can be used to determine the overall heat transfer coefficients from experimental data. However, the LMTD method is the one used by most researchers and is recommended to perform the thermal evaluation of the chiller system under study.

Using the LMTD method, global heat transfer coefficients values ranging from 11.1 to 73.3 W/(m^2·K) were obtained, while applying the ε-NTU method, the results ranged from 11.0 to 58.9 W/(m^2·K). The coefficients determined for the heat exchange between hydrogen sulfide and water allow delimiting the range of preliminary values published by other authors for gases and water, in tubular exchangers.

Conflict of interest

The authors declared no conflict of interest.

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