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Experimental Study of the Heat transfer coefficient in a Plate Heat exchanger

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Abstract : The way to evaluate a device in charge of heat transfer under certain working conditions is through the study of the global coefficient of heat transfer, it was carried out using a test bench that includes the base unit where the heating and movement of water is carried out and a unit where the heat exchange is carried out connected by means of flexible tubes and using the fundamental equations that are required, the results obtained were respectively the global transfer coefficient for both fluids of work with their respective Nusselt number, finally it is observed that the relationships between other models for plate exchangers, the comparative error is very large (40%) and to have more concise results. the scale factor must be taken into account.
Keywords : heat transfer coefficients, plate heat exchanger, empirical model, Nusselt number.

1. Introduction

In the thermal applications of mechanical systems, one of the most outstanding points is energy studies, since they represent a great impact on costs in the industry, as well as on the environment. Heat exchangers guarantee a large part of the success in these applications because they are designed to offer better efficiency in heat transfer between two fluids, optimizing energy consumption and maintenance and installation costs [1],[2].

Annular flow analysis is very common in plate heat exchangers, Reynolds liquid crystal thermography heat was analyzed from 2000 to 7500 for Reynolds, obtaining results to create new correlations for annular flow [3]. There are works concerning new ways to combine local heat transfer measurements and flow visualization using an FPHE simultaneously, obtaining viable solutions to obtain local heat transfer with a flow regime without a plate channel, preserving real geometry and operating condition [4]. Unprecedented cross-corrugated plate characterizations have been performed for aero-thermal performance for aerospace engine cycles, using a transient liquid crystal heat transfer (TLC) technique, with Reynolds 1300 to 13000, resulting in new Nusselt and Prandtl correlations for future plate heat exchanger optimizations [5],[6]. They have also conducted experimental comparative studies for sealed plate heat exchangers using three different plates, based on Reynolds numbers between 300-5000 and Prandtl numbers depending on Chevron's angle, correlations were obtained for each plate, deriving for the number of Nusselt and the friction factor [7]. Also, correlations were

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determined for the Colburn factor and friction and optimisation of the shape of Chevron plate heat exchangers, using a numerical analysis with a simulation of large eddies (LES); in order to validate, the experiment was made, with a maximum error of 10%, the factors were found with geometric parameters (chevron angle, p inclination, and h height), obtaining optimal values (Chevron angle = 66.5, p/h = 2.73), where they are constant and checked by Reynolds number [8]. Analyzing the modeling of a turbulent flow for an air-gas tube for Chevron plate heat exchangers, it was shown that a flat plate can be optimized to a Chevron plate, improving the velocity of the vectors and the shape of the eddies. It was also shown that the greater angle of Chevron plates widened by 18% at the outlet, with an increase in pressure drop of 68% compared to the flat plate heat exchanger [9]. The models for plate heat exchangers, using an organic Rankine cycle system, have been studied in different areas and different substances [10]–[12]. The study of an application of a $TiO2$/water nanofluid in a plate heat exchanger for milk pasteurization is a very common application to investigate simply because the use of $TiO2$ helps considerably in pressure drop [13],[14]. In other cases of application, studies have been carried out on the coefficients of convective transfers of $CO2$ hydrate mixtures in refrigerating devices. The results obtained for the mentioned mixture are between 0 and 14% of solid fraction, besides that the $CO2$ hydrate mixture is 2.5 more effective than water [15]. Experimental studies have also been conducted on welded plate heat exchangers to evaluate new correlations to improve the efficiency of these heat exchangers [11],[16]. They have also been made in $SOC2$/water heat transfer in a plate heat exchanger with a printed circuit board with direct channels with Brayton cycle, obtaining results of 17.6% when operating in a critical transient state [17]. In some studies, fin-shaped plate heat exchangers are analyzed, working with Reynolds from 1500 to 2600, obtaining results of diameter optimization, friction factor and Colburn and drying [18]–[21]. The heat transfer of a supercritical coolant with high variations in critical temperatures has been tested, using two plates with different corrugation angles of 30° and 60° respectively, examined in Reynolds (320-8000) and Prandtl (3.2-4.2), obtaining results, using the Jackson-Hall correlation factor, that the effects of buoyancy have much influence on the new correlations found for 30° [22]. Single-phase studies were conducted for forced convection heat transfer using CFD analysis to demonstrate experimental results and see the behavior of the Nusselt number in microchannels [23]. An investigation of dimensional constants representing the secondary aspects of the distribution of Nusselt curves was carried out using simulations to vary heat, fluid velocity, to find said constant as the nozzle radius analyzed by Reynolds number, giving an approximate value of 6000 [24]. n an experimental investigation of nanofluids, the effect of adding nano-particles of $SiO2$ in water is studied, where new correlations were found to predict viscosity in this nanofluids [25].

The analysis carried out in the previous articles makes clear the need to always estimate or obtain heat transfer coefficients in heat exchangers, precisely the selection and evaluation of these devices are carried out thanks to these coefficients that are related in a dependent way with the dimensional numbers: Prandtl, Nusselt and Reynolds. The lower the experimental error, the better the efficiency and the installation of the heat exchanger in a system, the contribution will be to calculate and validate experimentally the heat coefficients of a plate heat exchanger working with cold and hot water.

2. Methodology

This section of the article provides a brief description of the equipment to be used and the fundamental equations needed to obtain results from the experiment.

2.1. Equipment Description

The equipment used for the experimental study of heat exchange consists of an EDIBON plate heat exchanger shown in Figure 1 allows us to study the heat transfer between hot and cold water circulating through alternating channels formed between parallel plates. The base unit and heat exchanger are connected by means of flexible hoses to ensure the circulation of hot and cold water. The interface together with the control software (SCADA) allows us to visualize on screen the measurements made during the test: temperatures in the heat exchanger, temperature in the heat exchanger, temperature of the heating tank and water flow rates as shown in Figure 1

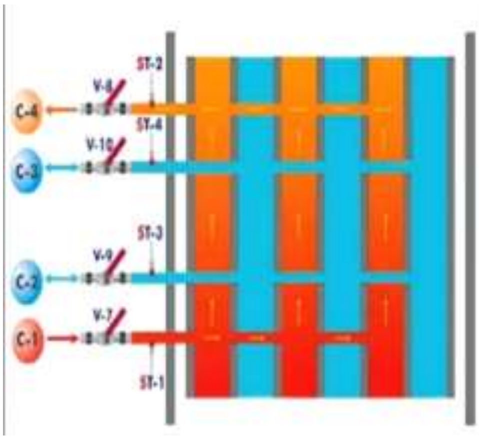

Figure 1. Plate heat exchanger TIPLA

The base unit fulfills various functions such as water heating, which is provided by a device with an electric resistance of 3000 W, thermocouple type J for measuring the water temperature, level switch for controlling the water level of the tank, lid and drain valve; measurement of cold and hot water flow rates, using two electronic flow transducers, one for hot water and another one for cold water; the pumping of hot water by a centrifugal pump with adjustable speed adapter; and the variation of the cold water circulation direction, for this by having four ball valves that are configured if it is in parallel flow or countercurrent in the exchanger and four ball valves that allow us to empty the set of pipes as shown in Figure 2.

Figure 2. Unit base of the plate heat exchanger

The EDIBON® plate heat exchanger shown in Figure 3 allows us to study the heat transfer between hot and cold water circulating through alternating channels formed between parallel plates. It has its aluminum structure anodized and painted steel panel. It consists of 10 corrugated stainless steel plates, each plate has dimensions of 29x12x16 cm with 4 connections for water inlet and outlet.

Figure 3. Extended plate heat exchanger TIPLA

For further information on the sensors and their respective equipment, please refer to the manual [26].

2.2. Fundamental Equations

The use of convection transfer coefficients for both fluids (hot and cold water) made it possible to find the overall heat transfer coefficient using the following equation from the literature

$$\frac{1}{\mathrm{U}*\mathrm{A}}=\frac{1}{hl*A}+Rcond+\frac{1}{ha*A} \tag{1}$$

Where U is the global heat transfer coefficient (W/m^2·K), h_l,h_a are the convection heat transfer coefficients for hot and cold water; (W/m^2·K), R_{cond} is the conductive resistance (K/W), A is the heat exchange area (m^2).

The overall coefficient of heat transfer depends on the properties of the fluid and its regime, the working parameters chosen and the coefficients of heat transfer by convection, experiments done show how to obtain several coefficients with different fluids, the detail is that there is a need to resort to experimental methods because there is not a literature that works for all. The number of Nusselt can be left in terms of the number of Reynolds and Prandtl as follows

Nu=f(Re, Pr, admensional geometry)

It is only valid if working in a laminar flow area between 0.6 ≤ Pr ≤ 50. In order to calculate the convection coefficients, a relationship is made with dimensional numbers such as Reynolds Number (Re), Nusselt (Nu), Prantdl (Pr). Having as Nusselt of the form

$$h=\frac{Nu*K}{Lc} \tag{2}$$

And the characteristic length, as follows

$$L_c=\frac{Area}{Perimeter} \tag{3}$$

The equation that relates them is the following equation

$$h=\frac{c*R_e^{\,n}*\mathrm{Pr}^{\frac{1}{3}}*k}{L_c} \tag{4}$$

The values of the coefficients n and c are experimental and depend on the type of flow and. The characteristic length of the channel (Lc). Where k is the thermal conductivity of the fluid (W/m·K), b is the width of the channel or distance between plates (m). As can be seen from the equation, different dimensional numbers are needed (Reynolds (Re), Nusselt (Nu), Prantdl (Pr)), the Reynolds number relates the inertial forces to the viscous forces and is important in order to be able to determine the regime to which the flow is found and is represented by the following formula

$$\mathrm{Re}=\frac{\dot{m}*L_c}{S*\mu} \tag{5}$$

Where R_e is Reynolds number, $\dot{m}$ is the mass flow [Kg/s], S is the cross section of the exchanger [m^2]$L_c$is the pipe diameter or its equivalent diameter [m] μ is the dynamic or absolute viscosit[Pa·s].

These are the standard parameters with the Reynolds number

Re < 2300 Flow follows laminar behaviour.
2300 < Re < 4000 Transition zone form laminar to turbulent fluid.
Re > 4000 The fluid is turbulent.

The Prandtl number relates the velocity of diffusion of the amount of motion to the velocity of heat diffusion, which means that it controls the relative thickness of the moment and thermal boundary layers. Values were taken from literature. The higher the Prandtl number, the more efficient the convection heat transfer is than conduction, the more common values found in practice are:

- For gas (Pr = 0.7)
- Non viscous fluids (Pr=1-7)
- Viscous fluids: Motor oil, glycerine (Pr = 3060-3400)

For a Prandtl number that is small means that heat diffuses very fast compared to speed (or time).

There are different correlations for this number of Nusselt depending on the flow rate, what type of geometry the heat exchanger has and the range of Prandtl values. Having a fully developed laminar flow between parallel plates, which is our case, the correlation will be used for the calculation of Nu[27]

$$Nu = 7.54 + \frac{0.03 * \frac{D_h}{L} * \mathrm{Re} * \mathrm{Pr}}{1 + 0.016\left(\frac{D_h}{L} * \mathrm{Re} * \mathrm{Pr}\right)^{2/3}} \tag{6}$$

2.3. Equipment Description

According to the calculation procedure, the main variables involved in the hot water cooling process are as follows:

- Inlet temperature of hot water and cooling water
- Hot water and cooling water outlet temperature
- Mass flow of hot water and cooling wáter

Experiments to determine heat transfer coefficients were carried out by setting two variables: the mass flow of hot water and the mass flow of cold water, the remaining variables were considered random. The heat exchange area is 0.192 m2 and the plates used are straight.The experimental results of the variables measured in the plate heat exchanger, to determine the coefficients of heat transfer by convection.as shown in Table 1.

Table 1. Inlet and outlet temperature measurement of the two fluids

Operational Condition	**Request Temperature (°C)**	**Volumetric flow cold water (L/min)**	**Volumetric flow hot water (L/min)**
Parallel/Counterflow	45-47-50-52-55-58-60-63-65	1.2 - 1.4 - 1.6	1.2 – 1.4 – 1.6

3. Results

Figure 4 shows the behavior of the Nusselt number as a function of Reynolds for the fluids involved in the heat exchange process. An increase in Nusselt values is observed with the increase in Reynolds number, which is associated with the velocity of the fluid by the plate heat exchanger.

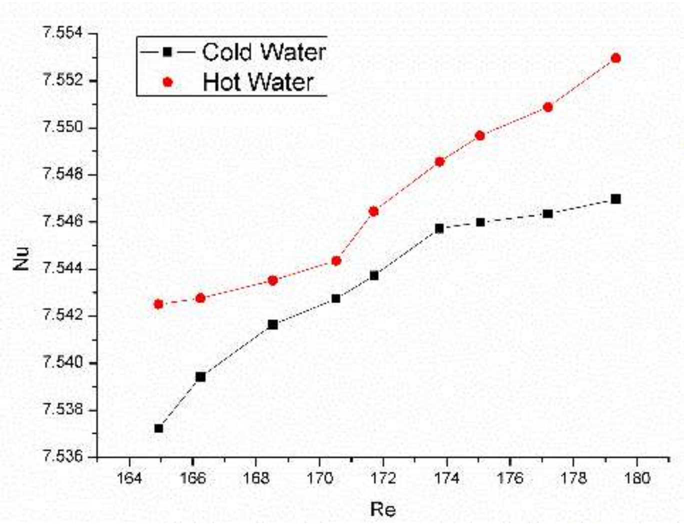

Figure 4. Performance of the number of Nusselt according to Reynolds for hot and cold water

Figure 5 shows the behaviour of the number of Nusselt as a function of the number of Reynolds in plate heat exchangers according to experiments by several researchers [28] and that obtained experimentally in this work marked with the name "Hot Water". The model that resembles the behavior obtained in this research is the Buonopane model, with a comparative error of 40%.

Figure 5. Performance of the number of Nusseltccording to Reynolds for hot and cold water

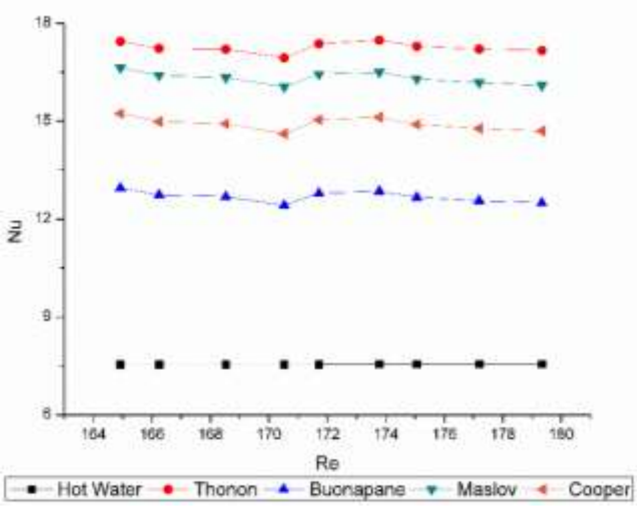

Figure 5. Performance of the number of Nusseltccording to Reynolds for hot and cold water

The results obtained by the comparison models (Thonon, Buonapane, Maslov and Cooper) differ from the values presented in the heat transfer coefficients, with a comparative error of over 40%, so it is not advisable to use these results in subsequent analysis.

Table 2. Comparative values with models obtained for plate heat exchangers

Name	Equation	Systematic Error (%)	Fluid Type
Hot Water	$Nu = 7.54 + \dfrac{0.03 * \frac{D_h}{L} * \mathrm{Re} * \mathrm{Pr}}{1 + 0.016\left(\frac{D_h}{L} * \mathrm{Re} * \mathrm{Pr}\right)^{2/3}}$	-	-
Thonon	$Nu = 0.2946 * \mathrm{Re}_a^{\ 0.7} * \mathrm{Pr}_a^{\ \frac{1}{3}}$	56.27%	Water
Buonapane	$Nu = 0.2538 * \mathrm{Re}_a^{\ 0.65} * \mathrm{Pr}_a^{\ 0.4}$	40.49%	Water
Maslov	$Nu = 0.78 * \mathrm{Re}_a^{\ 0.5} * \mathrm{Pr}_a^{\ \frac{1}{3}}$	53.76%	Any fluid
Cooper	$Nu = 0.2983 * \mathrm{Re}_a^{\ 0.65} * \mathrm{Pr}_a^{\ 0.4}$	49.40%	Any fluid

The experiments carried out had certain drawbacks with respect to the base unit, mainly with the centrifugal pump, the previous equipment described above had limited performance in the analysis of higher flow rates and thus vary the flow rate.

4. Conclusions

It is concluded that the global heat transfer coefficient obtained from an EDIBON® plate exchanger handling hot and cold water has a dispersion of 40%, this shows that the experiment despite having had certain disadvantages and assumptions, generates a value not very scattered and that the efficiency of work was higher in counter-flow.

Likewise, we can see the importance of these analyses when characterizing a heat exchanger of any kind, so that it can be concluded closer to reality should be further tested and take into account the incrustations. These dimensional studies must always be carried out in a manner consistent with those that have been studied in order to continue contributing to the study of heat exchangers and the development of this particular area.

Prolonged use of the heat exchanger increases the incrustation effect, whereby the fouling coefficient could reach 20% of the heat transfer that occurs in the exchanger, and so it is advisable to use the fouling factor for more concise results.

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