Thermal Convexity of Tubular Heat Exchangers in Steady State

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Abstract: This paper deals with the thermal convexity of heat exchangers in steady state: ($T_{h,c}(x)=\gamma_{h,c}(x)T_{c,in}+(1-\gamma_{h,c}(x))T_{h,in}$ with $0 \leq \gamma_{h,c}(x) \leq 1$). This method assesses the spatial distribution of the thermal convexity factors of both fluids along a tubular heat exchanger in counter-current flow and co-current flow arrangement. Analytical expressions of the thermal convexity coefficients are in exponential form. According to the flow configuration two linear functions are proposed for the hot and the cold fluid. The slope of these two functions corresponds to the exponential factor. The estimations of the exponential factor thanks to the steady state convexity coefficient profile provide results that are in good agreement with those obtained from correlations.

Keywords: Heat exchanger, thermal convexity, steady state.

1. INTRODUCTION

Heat exchangers are widely used in all kinds of industries to control heat transfer and or thermal equipment temperature. They are most often connected to other equipment that can cause changes in parameters such as inlet temperatures and mass flow rates. Generally, heat exchangers design is based on the calculation of their characteristics in steady state. In the open literature, the steady state solutions are commonly based on global heat transfer coefficient (Kakaç and Liu [1]; Shah [2], Serth [3]). As explained by Claesson [4], the Logarithmic Mean Temperature Difference (LTMD) method is simple to use when the inlet temperatures are known and the outlet temperatures are specified. A Modified LMTD method has been recently proposed to study indirect evaporative heat exchangers (Cui *et al.* [5]). If the outlet temperatures are unknown, the effectiveness-Number of Transfer Unit (ε-NTU) method should be used (Noie [6]; Ranong and Roetzel [7]). This method is based on calculation of the maximum possible heat transfer rate and the number of transfer units. Fakheri [8, 9] proposed a simplified concept based on the efficiency of heat exchangers obtained from a single algebraic expression. These methods give a global approach for heat exchangers analysis. The spatial temperature distribution is required in some applications, particularly when local temperature calculation is necessary for safety analysis. In this case, the usual techniques are not sufficient to analyze this kind of assessment of this thermal equipment. Indeed, the spatial temperature distribution can be useful for thermal analysis of such a device. In this sense, a modelling approach with partial differential equations is needed. In this paper, a new steady state method based on the thermal convexity property of heat exchangers is presented. This method is based on the assessment of the steady state thermal convexity coefficients for both fluids along a tubular heat exchanger. Analytical expressions of the convexity coefficients for both fluids are presented. Validation of these expressions with experimental results is also illustrated. The exponential factor, strongly dependent of convective heat transfer coefficients of both fluids, is obtainable from the best fitting of two proposed functions according to the flow axis. It is also obtainable from the outlets convexity coefficients. This alternative method represents the linear link between the outlet temperatures and the inlet temperatures and the nonlinear relation to the flow rates through the thermal convexity coefficients.

2. DESCRIPTION AND FORMULATION

2.1. Experimental Device and Description

In order to characterise the spatial distribution of the steady state convexity coefficients of the two fluids, the experimental device is sufficiently instrumented. Figure **1** shows the experimental set-up used to validate the theoretical results. Temperature probes are alternatively placed along the insulated tubular heat exchanger and are spaced of 0.36 m for both fluids. The sensors are connected to a data acquisition card inserted in a computer that allows the recording of the temperatures along the heat exchanger. The inner tube is in copper and the outer one is in steel. The physical and geometrical characteristics of the tubes are reported on Table **1**. The two streams are water-water flow.

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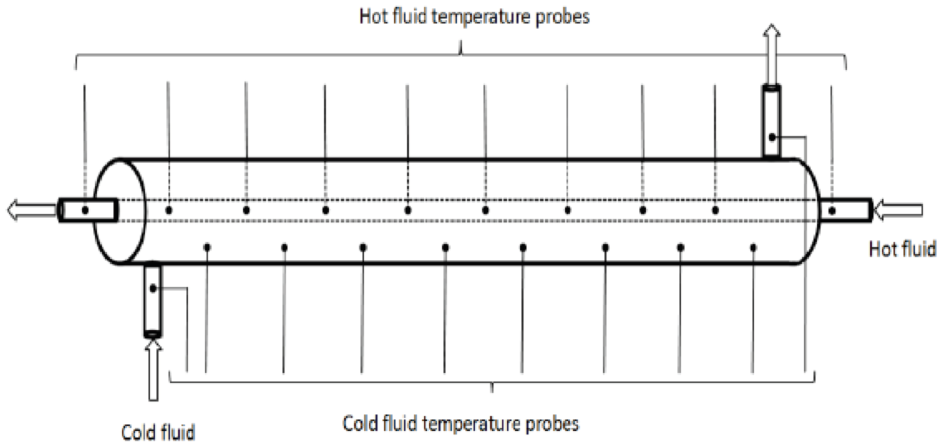


Figure 1: Experimental set up.

Table 1: Physical and Geometrical Characteristics of Inner and Outer Tubes

	k [W.m^{-1}.K^{-1}]	C_p [J.kg^{-1}.K^{-1}]	ρ [kg.m^{-3}]	D [m]	a [m]	L [m]
Inner tube	384	394	8900	2 10^{-2}	10^{-3}	4.5
Outer tube	45	490	7850	4 10^{-2}	3 10^{-3}	4.5

2.2. Assumptions and Modelling

The mathematical model used in this study is proposed by Patankar *et al.* [10]. The assumptions made in our case are:

- Fluids are in turbulent flow
- Heat conduction along the flow axis is neglected
- Fluids are incompressible and single phased
- Thermo physical properties of the fluids are assumed to be constant
- Separating wall is assumed to be isothermal along the radial axis. The Biot numbers of both sides, $Bi_{h,c} = \frac{h_{h,c}\, a}{k_w}$, corresponding to experimental conditions are lower than 0.01 ($h_{h,c}$ < 5000 W.m^{-2}.K^{-1}).

In order to obtain the equations that govern the thermal behaviour, the heat exchanger is subdivided in several elementary volumes with a length of dx as indicated on Figure **2.** While crossing elementary volume, the hot fluid transfers heat to the wall by the convection. This contributes to the reduction in its outlet enthalpy and internal stored thermal energy. Energy balance applied to a differential volume of hot fluid leads to the first equation of system (1) after simplification and rearrangement. Similar energy balance applied to the cold fluid and the separating wall gives the second and the third equations of system (1).

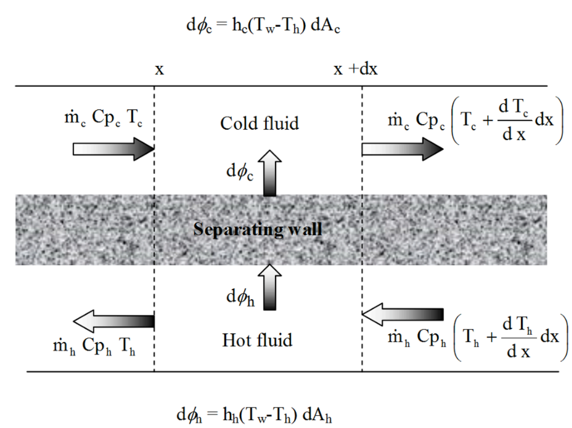


Figure 2: Elementary energy balance taking into account convective heat transfer.

$$\begin{cases} \pm \dot{m}_h\, Cp_h\, L \dfrac{d T_h}{d x}(x) + h_h A_h \left(T_w(x) - T_h(x)\right) = 0 \\ \dot{m}_c\, Cp_c\, L \dfrac{d T_c}{d x}(x) + h_c A_c \left(T_w(x) - T_c(x)\right) = 0 \\ h_h A_h \left(T_h(x) - T_w(x)\right) + h_c A_c \left(T_c(x) - T_w(x)\right) = 0 \end{cases} \quad (1)$$

The ± expresses the co-current or counter current flow.

Let us define the following parameters:

- Dimensionless axial position

$$x^* = x / L \tag{2}$$

- Thermal rate ratio

$$C^* = \frac{\dot{m}_h \, Cp_h}{\dot{m}_c \, Cp_c} \tag{3}$$

$$N_h = \frac{h_h \, A_h}{\dot{m}_h \, Cp_h} \tag{4}$$

$$N_c = \frac{h_c \, A_c}{\dot{m}_c \, Cp_c} \tag{5}$$

Using the above parameters, system (1) can be rewritten in the following form:

$$\begin{cases} \pm \dfrac{d\,T_h}{d\,x^*}(x) + N_h \left(T_w(x) - T_h(x)\right) = 0 \\ \dfrac{d\,T_c}{d\,x^*} + N_c \left(T_w(x) - T_c(x)\right) = 0 \\ C^* N_h \left(T_h(x) - T_w(x)\right) + N_c \left(T_c(x) - T_w(x)\right) = 0 \end{cases} \tag{6}$$

The heat exchanger is subject to the following boundary conditions:

- In counter-current flow configuration:

$$\begin{cases} T_h(x^* = 1) = T_{h,in} \\ T_c(x^* = 0) = T_{c,in} \end{cases} \tag{7}$$

- In co-current flow configuration:

$$\begin{cases} T_h(x^* = 0) = T_{h,in} \\ T_c(x^* = 0) = T_{c,in} \end{cases} \tag{8}$$

$T_{h,in}$ and $T_{c,in}$ are the inlet temperatures of the hot and the cold fluids respectively.

In this formulation, the groups N_h and N_c are very similar to the number of transfer units used in the steady-state NTU method (Kakaç and Liu [1]; Rohsenow *et al.* [11]. The NTU is linked to the global heat transfer while N_h and N_c depend on the convective heat transfer coefficients.

Determination of the convective heat transfer coefficients h_h and h_c constitutes an important point of this model. These coefficients can be assessed in two ways. The first one consists in determining the best fit between the experimental and theoretical steady state results according to the good values of h_h and the h_c. The second one corresponds to the use of the Nusselt number correlations that are linked to the heat transfer coefficients.

3. THERMAL CONVEXITY PROPERTY

Heat transfer follows the conservation law of energy and mass. Let us consider for instance the single phase mixing of two identical fluids with different flow rates and temperatures as depicted in Figure **3**. The following relations could be then obtained:

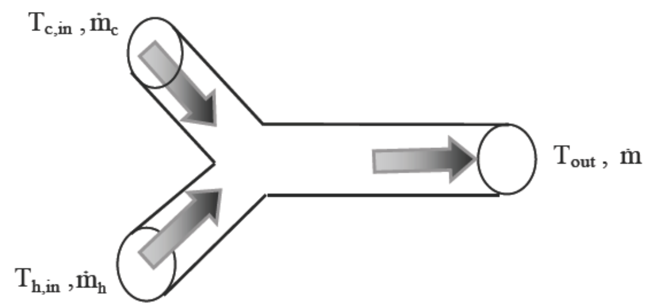


Figure 3: Mixing of two fluids.

$$\dot{m} = \dot{m}_h + \dot{m}_c \tag{9}$$

$$T_{out} = \gamma \, T_{h,in} + (1-\gamma) \, T_{c,in} \tag{10}$$

where the thermal convexity coefficient is:

$$\gamma = \frac{1}{1 + \dfrac{\dot{m}_c}{\dot{m}_h}} \tag{11}$$

with: $0 \le \gamma \le 1$ (12)

The relation (12) is valid for any values of the flow rates. γ depends on the flow rates through a non linear (hyperbolic) function and T depends linearly on temperatures $T_{h,in}$ and $T_{c,in}$. The outlet temperature is convex with regard to the inlet temperatures as long as the relation (12) is valid.

Let us now consider the heat transfer from a fluid through a surface A with a flow rate $\dot{m}$. U is the global heat transfer coefficient. Its inlet and outlet temperature are respectively $T_{h,in}$ and $T_{h,out}$. The other side is assumed to be at uniform temperature T_c. The classical "one compartment" representation (Incopera and DeWitt [12]; Kays and London [13]) gives:

$$U\,A\,(T_c - T_{h,out}) + \dot{m}\,C_p\,(T_{h,in} - T_{h,out}) = 0 \tag{13}$$

Equations (13) can be rearranged in the following form:

$$T_{h,out} = \gamma\, T_{h,in} + (1 - \gamma)\, T_c \tag{14}$$

Where the thermal convexity coefficient is given as:

$$\gamma = \frac{1}{1 + NTU}, \text{ with } 0 \le \gamma \le 1 \tag{15}$$

Figure **4** shows the variation of the convexity coefficient as function of the number of transfer units NTU.

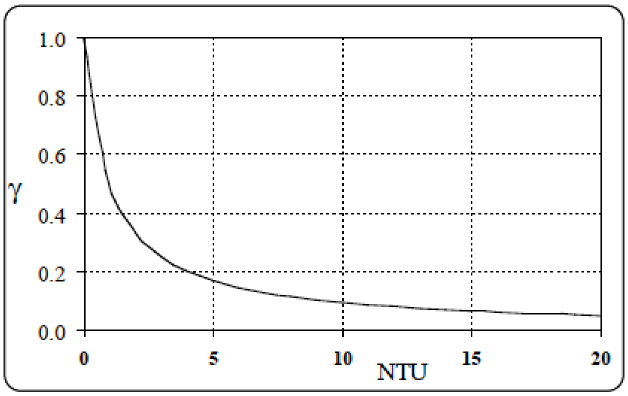


Figure 4: Thermal convexity coefficient of one classical compartment heat transfer as function of the number of transfer units.

A similar structure to describe the heat exchangers is also possible. Indeed, the exact steady state solutions of the system of equations (6) are obtained, after rearrangement, by the following expressions:

$$\begin{cases} T_h(x^*) = \gamma_h(x^*)\, T_{h,in} + \left(1 - \gamma_h(x^*)\right) T_{c,in} \\ T_c(x^*) = \gamma_c(x^*)\, T_{h,in} + \left(1 - \gamma_c(x^*)\right) T_{c,in} \end{cases} \tag{16}$$

The thermal convexity coefficients of the hot and cold fluid are given as follows:

- In counter-current flow configuration:

$$\gamma_h(x^*) = \frac{e^{-\alpha_1 x^*} - C^*}{e^{-\alpha_1} - C^*} \tag{17}$$

$$\gamma_c(x^*) = C^* \left(\frac{e^{-\alpha_1 x^*} - 1}{e^{-\alpha_1} - C^*} \right) \tag{18}$$

The exponential factor is given the expression:

$$\alpha_1 = N\left(C^* - 1\right) \tag{19}$$

- In co-current flow configuration:

$$\gamma_h(x^*) = \frac{e^{-\alpha_2 x^*} + C^*}{1 + C^*} \tag{20}$$

$$\gamma_c(x^*) = C^* \left(\frac{e^{-\alpha_2 x^*} - 1}{1 + C^*} \right) \tag{21}$$

The exponential factor is given as follows:

$$\alpha_2 = N\left(C^* + 1\right) \tag{22}$$

The parameter N is defined with the same expression for both co-current and counter-current configuration and is given as follows:

$$N = \frac{N_h\, N_c}{N_c + N_h C^*} \tag{23}$$

It is worth noting that, the thermal convexity property is valid for each position along the heat exchanger. ($0 \le \gamma_h(x^*) \le 1$ and $0 \le \gamma_c(x^*) \le 1$) These relations show the influence of the convective heat transfer coefficients on the thermal convexity coefficient profiles. They appear through the dimensionless groups Nh and Nc. The best fit of experimental results in steady state enables a good estimation of the exponential factor but cannot give separately Nh and Nc. One way to distinguish the heat transfer coefficients is the use of correlation for the hot fluid flowing in the inner tube, what makes it possible to extract the hh and to thereafter deduce the hc from the exponential factor. In this case, the modified Colburn correlation widely used in the literature for circular ducts was given by Sieder and Tate [14] as follows:

Nu_h = ,,h-h.,D-i.-,k-h.. = 0.027, Re-h-4/5., Pr-h-1/3.,,,μ-,μ-...-0.14. (24)

where μ / μ_w is the ratio of dynamic viscosity at the flow center and at the wall proximity. The Reynolds number Re_h of the hot fluid is given by the following expression:

$$Re_h = \frac{4\, \dot{m}_h}{\pi\, D_i\, \mu} \tag{25}$$

The Prandtl number Pr_h is widely tabulated in the literature for different liquids.

Many correlations exist in the literature for the circular and annular ducts (Jacob [15]). As mentioned before, the modified Colburn correlation for the circular-ducts is usually employed in the literature while for the annular ducts, other kind of correlations are investigated. Kawamura [16] proposed two correlations for turbulent annular flow according to the radius ratio ($r^* = r_e/r_i$). The first one concerning the radius of maximum velocity r_m is used in the second treating the Nusselt number Nu. These relations are:

$$r_m = r_{m\infty} + \left(r_{ml} - r_{m\infty}\right)\left(\frac{2300}{Re_c}\right)^{0.8} \tag{26}$$

where r_{ml} is the radius of maximum velocity in laminar flow:

$$r_{ml} = \sqrt{\frac{r^{*2} - 1}{2\ \ln(r^*)}} \tag{27}$$

$r_{m\infty}$ is the radius of maximum velocity for fully developed turbulent annular flow and is expressed as:

$$r_{m\infty} = \frac{r^{*n} + r^*}{r^{*n} + 1} \tag{28}$$

The empirical constant n is n = 0.415.

The second correlation corresponding to the Nusselt number is given by:

$$Nu_c = 0.022 \quad \varphi_i \quad Re_c^{0.8}\, Pr_c^{0.5} \tag{29}$$

where φ_i is given by the following expression:

$$\varphi = \frac{r_m^{\ 2} - 1}{r^{*2} - 1} \tag{30}$$

This accommodation factor φ_i represents the ratio of the shear stress on the inner wall to the average shear stress on the inner and outer walls.

4. RESULTS AND DISCUSSION

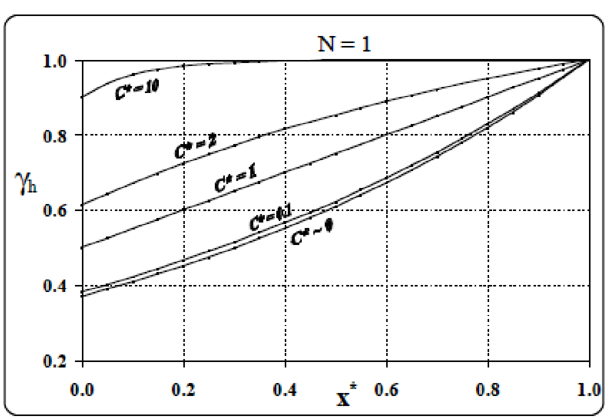
In this section, the thermal convexity coefficients calculation is given for both fluids in several configurations.

4.1. Counter-Current Configuration

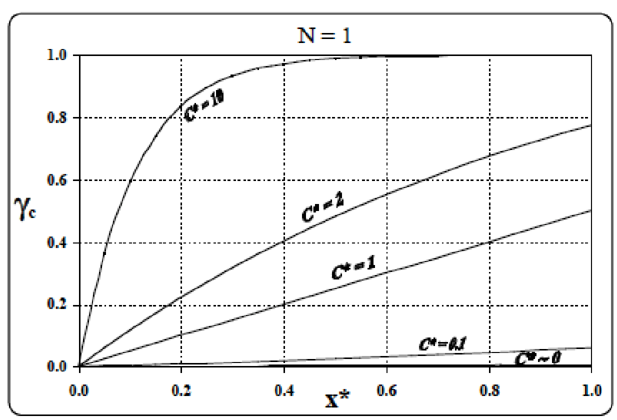
Figure **5** illustrates the convexity coefficients of tubular heat exchangers in counter-current flow configuration for different values of C^* with N = 1. Figure **5a** relates to the hot fluid and Figure **5b** corresponds to the cold fluid. The thermal convexity coefficients increase with the thermal rate ratio. The singularity appears when C* = 1. In this case, the thermal convexity coefficients are linear according to the axial position along the heat exchanger. When C* tends towards very high values (C* >> 1), the convexity coefficient of the hot fluid tends uniformly towards 1 along the heat exchanger. The hot fluid enters with the inlet temperature and remains at this temperature almost along the exchanger of heat. The convexity coefficient of cold fluid attains its maximum profile. When C* >> 1, the thermal convexity coefficients become:

$$\begin{cases} \gamma_h(x^*) \approx 1 \\ \gamma_c(x^*) \approx 1 - e^{-N_c\ x^*} \end{cases} \tag{31}$$

On the other hand, when C* << 1, the convexity coefficient of the cold fluid tends towards 0. So the temperature of the cold fluid is quasi constant along the heat exchanger. The convexity coefficients can be approximated by the following expressions:



(**a**)



(**b**)

Figure 5: Thermal convexity coefficient along counter-current tubular heat exchanger for different values of thermal rate ratio with N = 1. (**a**) Hot fluid – (**b**) Cold fluid.

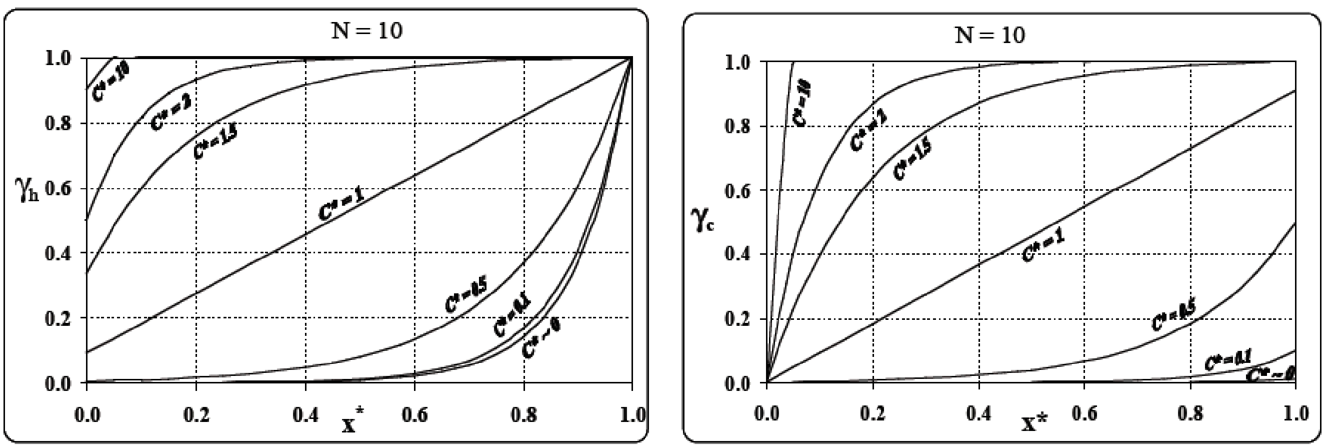
$$\begin{cases} \gamma_h(x^*) \approx e^{N_h\,(x^*-1)} \\ \gamma_c(x^*) \approx 0 \end{cases} \quad (32)$$

Figure **6** shows the heat exchanger thermal convexity coefficients for different values of C* with N=10. The curves of Figure **6a** relate to the hot fluid and the curves of Figure **6b** correspond to the cold fluid. The convexity coefficient seems to be more sensitive to the variation of the thermal rate ratio when N increases. The assessment of the heat exchangers convexity coefficients gives the exponential factor, which is linked to the convective heat transfer coefficients. For that, we define in the present paper two linear functions extracted from the steady state convexity coefficients:

$$\Gamma_h(x^*) = -\ln\left\{(1-C^*)\,\frac{\gamma_h(x^*)}{\gamma_{h,out}} + C^*\right\} \quad (33)$$

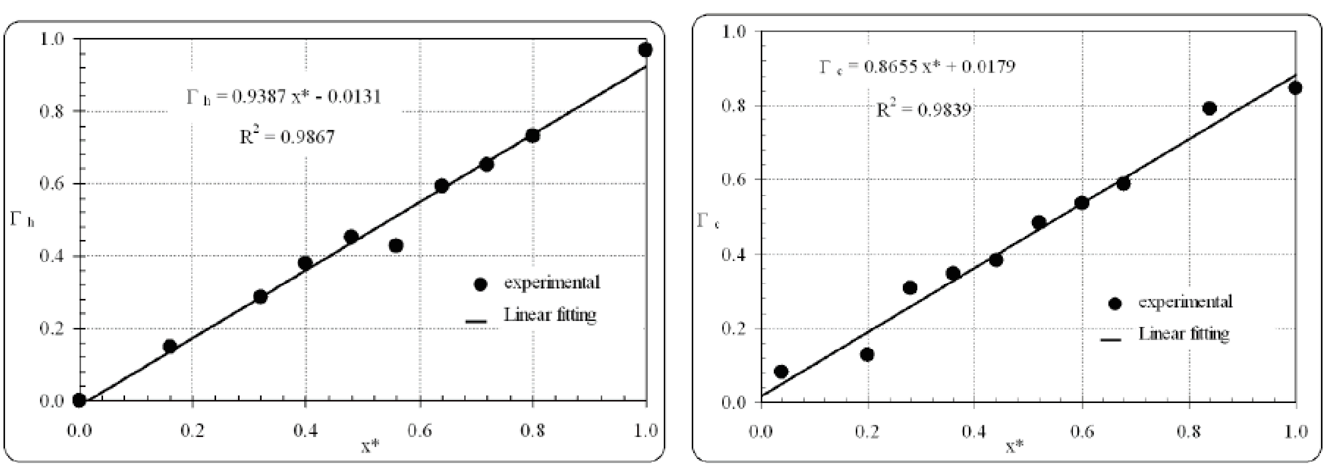
$$\Gamma_c(x^*) = -\ln\left\{\left(\frac{1-C^*}{C^*}\right)\frac{\gamma_c(x^*)}{\gamma_{h,out}} + 1\right\} \quad (34)$$

The slopes of Γ_h and Γ_c as function of x^* correspond precisely to the exponential factor α_1 given in relation (19). The steady state best fit of the linear functions (33) and (34) with experimental results gives the exponential factor α_1. The results obtained by this method are compared with those deduced from the correlations discussed in the previous section. Figure **7a** shows an example of experimental Γ_h along the heat exchanger for the hot fluid. Figure **7b** corresponds to the cold fluid. The best linear fitting with the experimental points of Γ_h and Γ_c allows the evaluation of the slopes, which correspond to α_1. Results obtained by this method and those deduced from correlations



(**a**) (**b**)

Figure 6: Thermal convexity coefficient along counter-current tubular heat exchanger for different values of thermal rate ratio with N = 10. (**a**) Hot fluid – (**b**) Cold fluid.

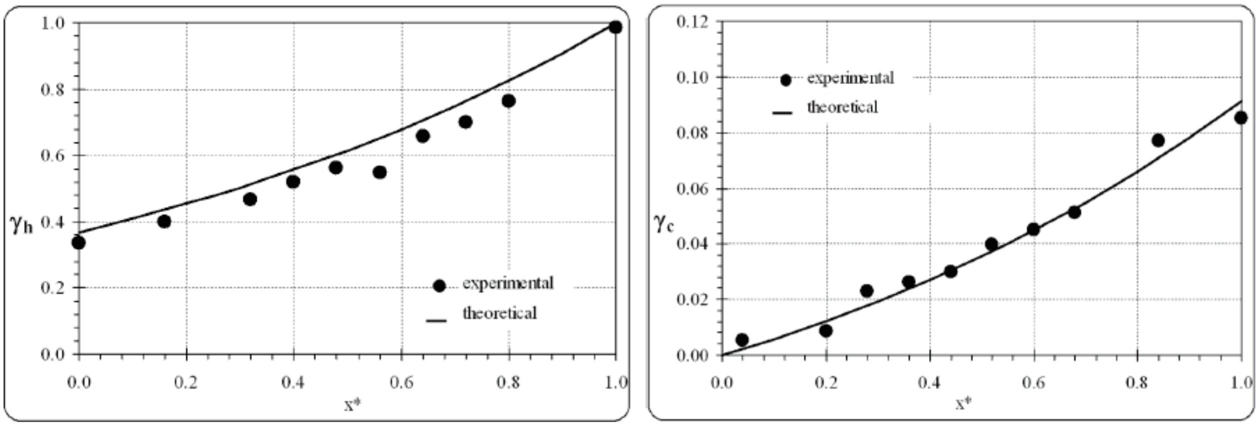


(**a**) (**b**)

Figure 7: Experimental Γ as a function of x* with its linear fitting. (**a**) Hot fluid. (**b**) Cold fluid.

Table 2: Exponential Factor Extracted from Linear Fitting and Correlations

Linear Fitting				Correlations				
C^*	α_h	α_c	α_1	N_h	N_c	N	α	$\Delta\alpha_1/\alpha_1$
0.162	0.94	0.86	0.90	1.52	0.53	1.038	0.89	1.1 %



(**a**) (**b**)

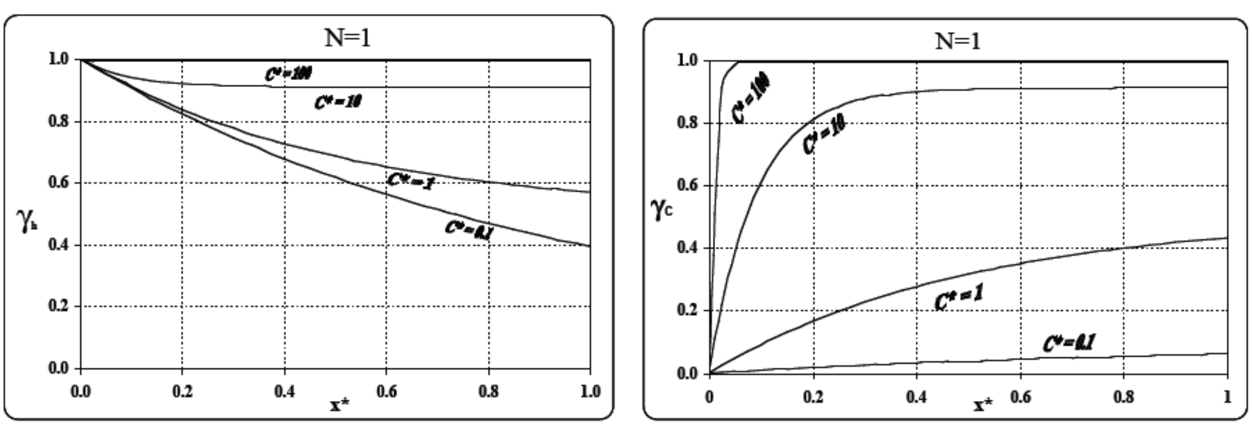
Figure 8: Experimental and theoretical steady state thermal convexity coefficient along the heat exchanger. (**a**) Hot fluid. (**b**) Cold fluid.

presented previously are shown in Table **2**. The convective heat transfer coefficients obtained from the correlations are in agreement with those extracted from the best fit. Figure **8a** shows the agreement between the experimental and theoretical thermal convexity coefficient of the hot fluid along the tubular heat exchanger in steady state. Also, the agreement is illustrated on Figure **8b** for the cold fluid.

It is important to note that in the industrial applications of the heat exchangers, measurements of the temperatures are usually taken only at the inlets and the outlets. In this case, the experimental exponential factor α_1 can be extracted from the following expression:

$$\alpha_1 = -\ln\left\{\sqrt{\frac{\left[1 - C^*\left(1 - \gamma_{h,out}\right)\right]\left[\gamma_{h,out} + C^*\left(\gamma_{h,out} - \gamma_{c,out}\right)\right]}{C^*\gamma^2_{h,out}}}\right\} \quad (35)$$

The expression (35) corresponds to an average value of α_1 deduced from the measured steady state outlet thermal convexity coefficients of the hot and cold fluids.



(**a**) (**b**)

Figure 9: Thermal convexity coefficient along co-current tubular heat exchanger for different values of thermal rate ratio with N = 1. (**a**) Hot fluid – (**b**) Cold fluid.

4.2. Co-Current Configuration

In the same way as the counter-current configuration, Figure **9** shows the steady state convexity coefficients along the heat exchanger in the co-current flow configuration with N = 1. The curves (a) of this Figure are relative to the hot fluid and the curves (b) correspond to the cold fluid. The convexity coefficients of the hot and cold fluids increase according the thermal rate ratio. When C* attains high values, the convexity coefficient of the hot fluid tends to 1. The convexity coefficient of the cold fluid takes the following expression:

$$\begin{cases} \gamma_h(x^*) \approx 1 \\ \gamma_c(x^*) \approx e^{-N C^* x^*} - 1 \end{cases} \quad (36)$$

When C* is negligible, the convexity coefficient of the cold fluid tends to 0 and that of the hot fluid tends towards a limiting profile. The expressions of the convexity coefficients can be approximated by:

$$\begin{cases} \gamma_h(x^*) \approx e^{-N_h x^*} \\ \gamma_c(x^*) \approx 0 \end{cases} \quad (37)$$

Figure **10** illustrates the steady state convexity coefficients along the heat exchanger with N = 10. When N increases, the span of the convexity coefficient increases.

The same analysis for determination of the exponential factor can be applied to co-current configuration by defining the two following linear functions:

$$\Gamma_h(x^*) = -\ln\left\{ (1-C^*)\ \gamma_h(x^*) - C^* \right\} \quad (38)$$

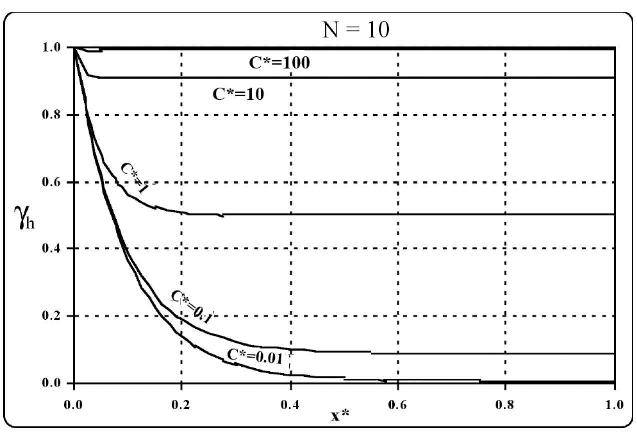
$$\Gamma_c(x^*) = -\ln\left\{ \left(\frac{1+C^*}{C^*}\right) \gamma_c(x^*) + 1 \right\} \quad (39)$$

The slopes of the linear functions (38) and (39) represent the exponential factor when the heat exchanger is in co-current configuration.

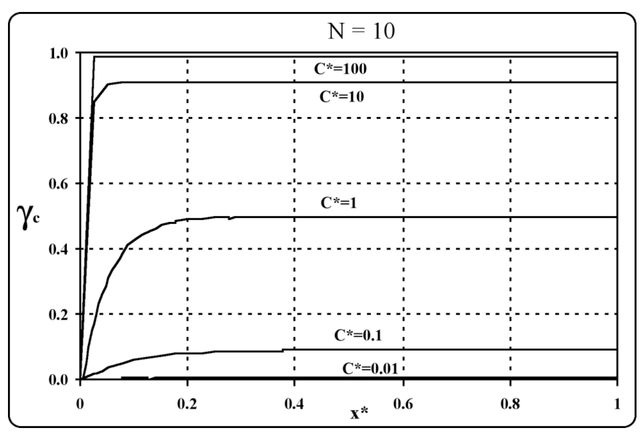
When the only outlets convexity coefficients are accessible, the experimental exponential factor α_2 can be extracted from the following expression:

$$\alpha_2 = -\ln\left\{ \sqrt{\frac{\left[(1+C^*)\ \gamma_{h,out} - C^*\right]\left[(1+C^*)\gamma_{c,out} + C^*\right]}{C^*}} \right\} \quad (40)$$

It is worth noting that this steady state method gives an alternative approach to the well known procedures for sizing and rating heat exchangers in industrial applications like ε-NTU and LMTD methods. It is based on the average convective heat transfer coefficients of both sides while the other methods are based on the overall heat transfer coefficient. The thermal convexity property shows the linear relation between the inlet and outlet temperatures and the nonlinear link to the flow rates through the thermal convexity coefficients. It represents a convenient steady state formulation for the heat exchanger networks analysis and optimization. Note that this method can be extended to other kinds of heat exchangers by introducing the Fin Analogy number well defined in the open literature for the practical use for all common multipass, shell-and-tube and cross flow heat exchangers.



(**a**)



(**b**)

Figure 10: Thermal convexity coefficient along co-current tubular heat exchanger for different values of thermal rate ratio with N = 10. (**a**) Hot fluid – (**b**) Cold fluid.

5. CONCLUSION

In this paper, a new steady state formulation introducing thermal convexity coefficients of the heat exchangers is presented. This method assesses the spatial profile of the thermal convexity coefficients along the tubular heat exchanger in counter-current flow and co-current flow. Two linear functions corresponding to the hot and cold fluids are proposed according to the flow configuration. This method, based on estimating the exponential factor of steady state convexity coefficients profile, is in good agreement with those obtained from correlations. Based on the average convective heat transfer coefficients, this method gives an alternative approach to the well known procedures for sizing and rating heat exchange-rs in industrial applications.

NOMENCLATURE

a wall thickness [m]

A heat transfer area [m^2]

Bi Biot number

C^* thermal rate ratio

C_p specific heat [$J.kg^{-1}.K^{-1}$]

D diameter [m]

h heat transfer coefficient [$W.m^{-2}.K^{-1}$]

k thermal conductivity [$W.m^{-1}.K^{-1}$]

L heat exchanger length [m]

$\dot{m}$ mass flow rate [$kg.s^{-1}$]

N dimensionless number

NTU Number of Transfer Units

Nu Nusselt number

Pr Prandtl number

r radius [m]

Re Reynolds number

T temperature [K]

U overall heat transfer coefficient

x axial position [m]

GREEK SYMBOLS

α dimensionless exponential factor

γ convexity coefficient

ϕ accommodation factor

μ dynamic viscosity [$kg.m^{-1}.s^{-1}$]

Γ logarithmic law

SUBSCRIPT

c cold fluid

e external tube

h hot fluid

i inner tube

in input stream

m maximum

ml maximum in laminar flow

$m\infty$ maximum for fully developed turbulent annular flow

out output stream

w separating wall

SUPERSCRIPT

* dimensionless form

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