TEMPERATURE EFFECT ON THERMAL-HYDRAULIC PERFORMANCE OF ONE-PASS COUNTER-CURRENT FLOW SHELL-AND-TUBE HEAT EXCHANGER AND UPON ITS DESIGN

By

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A methodology of design and analysis of thermal-hydraulic performance for a single pass 1-1 counter-current flow shell and tube heat exchanger (CCFSTHE) TEMA E type has been established. The temperature effect on the thermo-physical properties of flowing fluids and on the overall coefficient of heat transfer along the heat exchanger is incorporated in our approach, as well as the coupling between different thermal and hydraulic parameters. It has been noted that the correction factor (F) in the HAUSBRAND formula is not included. Our method brings us to a new dimensionless quantity (MKA) which links the calculation parameters of the heat exchanger to the thermo-physical properties. This dimensionless quantity relates the number of transfer units (NTU) to the heat flow ratio (R). The results based on our models show a pronounced deviation compared to the model reported in the literature (NTU method). This deviation may be related to a temperature effect not included in the literature model. It has been shown that the results derived from our models are in a good agreement with experimental data. Our new method, named MKA - method, could be a useful tool for theoretical and experimental studies of the design and analysis of the single pass 1-1 CCFSTHE thermal and hydraulic performance for $0 \leq R \leq 1$.

Keywords: *Heat transfer, single pass 1-1, CCFSTHE, Thermal design, Modeling, Effectiveness, MKA method.*

1. Introduction

Heat exchangers are classified according to many parameters such as transfer process, flow types, heat transfer mechanisms, construction, degrees of surface compactness, pass arrangements, and the phase of the process fluids [1, 2, 3]. The selection of a heat exchanger takes into consideration many factors including capital and operating cost, fouling, corrosion tendency, pressure drop, temperature ranges, and safety issues. Because of its structural simplicity, wide range of operational temperatures and pressures, with relatively low cost and design adaptability, the counter-current flow shell and tube heat exchanger (CCFSTHE) TEMA E type is largely used in various industrial fields such as petro-chemical industry, electrical power production, food preservation, manufacturing industry, and energy conservation systems. The CCFSTHE accounts for more than 35–40% of the heat exchangers used in global heat transfer processes [4, 5].

Much work directed at enhancing the shell-and-tube heat exchangers (STHE) performances have focused on the effects of the variation of their different thermal, hydraulic and mechanical parameters, such as fluids temperatures, flow rates, flow arrangement, materials types, heat exchanger type, tube length, shell and tube diameters, numbers of tubes and baffles, cost, and/or process optimization considerations [1, 6]. These studies were unfortunately restricted to certain parameters as reported in ref. [7], for example. Other studies have been carried out to investigate the effect of some thermo-physical properties [8] and the location of the tubes in the shell [9] on heat transfer. The coupling between thermal, hydraulic and mechanical parameters was neglected. The baffle forms and locations are also important elements; they are used in different geometrical shapes, and lead to fluid flow that increase its turbulence. The presence of baffles permits to enhance heat exchanger efficiency [10, 11]. Furthermore, a rise in the heat transfer rate increases the pressure drop [12]. The heat transfer rate and pressure drop inter-dependence should be taken into account in the STHE designing. In this sense, to further improve the CCFSTHE design, considerable efforts using different optimization methodologies have been developed to determine the thermo-hydraulic and mechanical parameters that optimise the efficiency [6, 13 - 18].

In conventional methods of the heat exchanger thermo-hydraulic performance analysis and design, thermo-physical properties of fluids and the overall heat transfer coefficient [19 - 25] are considered to be temperature-invariant and assumed to be uniform along the heat exchanger, which may induce in calculation errors. Additionally, industrial scale-up implies an increase in the inadequacy of calculations and errors due to the change of the thermo-physical properties of the fluids as a function of the temperature of hot and cold fluids throughout the heat exchanger [19]. The significant deviations between experimental and theoretical results have driven researchers to study the effect of thermo-physical property variations on the CCFSTHE thermal-hydraulic performance analysis and upon its design [20, 21]. The integration of methods, taking into account the temperature effect, leads to better results when compared to other approaches such as Colburn [21] and Roetzel et spang. [22]. These methods do not all incorporate the different parameters involved.

Therefore, in order to improve the accuracy of the performance analysis of the heat exchanger, we have developed MKA-method taking into account the combination of the thermo-physical properties, the overall heat transfer coefficient and their variation as functions of the temperature along the exchanger. The importance of this MKA - method is justified by comparison of our results with both those of the conventional method and the experimental results.

2- Principal equations governing the design of STHEs

Heat exchangers are devices commonly used in a wide range of applications. The most common design of STHE is the type TEMA E because of its design simplicity, robustness and a wide range of operational temperatures and pressures. Figure 1 shows an example of a single pass 1-1 CCFSTHE TEMA E type with baffles. To improve the heat transfer, baffles are used to direct fluid flow through the shell. The presence of baffles is important to obtain turbulences that boost up the overall heat transfer coefficients [1-3].

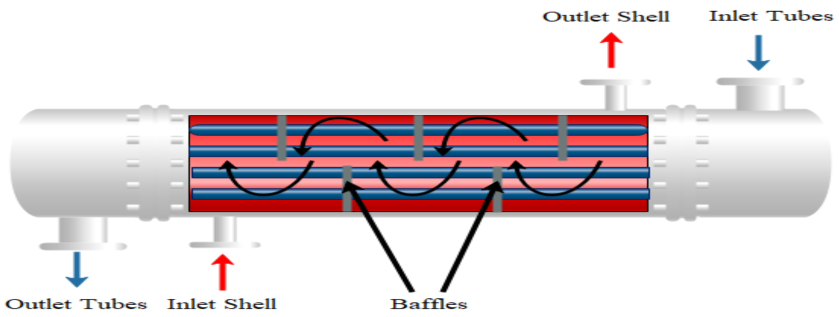

Figure 1. One-pass (1-1) CCFSTHE TEMA E type with baffling

Among simple types of heat exchangers, mono-tubular heat exchanger in which the temperature distribution of fluids throughout a tube is depicted in figure 2.

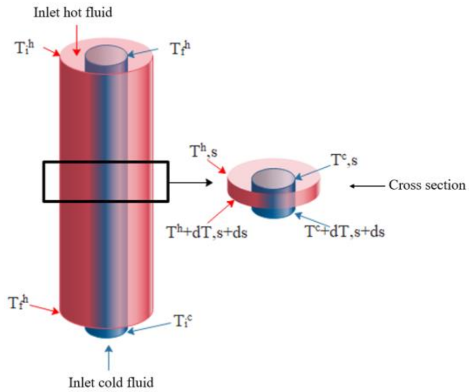

Figure 2. Temperature distribution of fluids in a single tube heat exchanger

The heat exchange calculation is based on the following formula [5]:

$$\Phi = K\,A\;\Delta T_{lm} \tag{1}$$

This equation provides a relationship between heat flux (Φ), overall heat transfer coefficient (K), exchange surface (A), and the logarithmic mean temperature difference (ΔT_{lm}). These parameters constitute key parameters of the heat exchanger and they should be determined as described below.

Generally, the calculation of the total heat exchanger area A refers to the following equation:

$$A = \int_{T_i^c}^{T_f^c} \frac{q^c c_p^c}{k\Delta T}\, dT^c \tag{2}$$

where q^c, c_p^c and T^c are, respectively, locale mass flow, specific heat and temperature of cold fluid, corresponding to the rectangular image shown in Fig. 2, k is the local heat transfer coefficient.

The overall heat transfer coefficient (K) is determined on the basis of thermodynamic or transport considerations and it covers almost all fluid thermal resistances and the conductivity of the tube material.

$$K = \frac{1}{\frac{A}{A_i h_i} + \frac{A}{A_i} R_{fi} + \frac{A}{2\pi L\lambda} \ln\frac{r_e}{r_i} + \frac{A}{A_e} R_{fe} + \frac{A}{A_e h_e}} \tag{3}$$

where h_i, h_e, R_i, R_e, r_i , r_e and λ are, respectively, the internal and external convective heat transfer coefficients, fouling coefficients, internal tube radius, external tube radius and the conductivity thermal of the tube wall. The hydrodynamic and thermal conditions of the circulating fluid depend on these resistances which subsequently amplify errors in the calculation of K [26]. Note that all these resistances are temperature dependent [19].

The logarithmic mean temperature difference (LMTD) is determined using the following formula:

$$\Delta T_{lm} = \frac{\Delta T_i - \Delta T_o}{\ln \frac{\Delta T_i}{\Delta T_o}} \tag{4}$$

with $\Delta T_i = T_i^h - T_f^c$ and $\Delta T_o = T_f^h - T_i^c$

Where ΔT_i and ΔT_o are the temperature differences between the two fluids in the inlet and outlet of the heat exchanger. The use of LMTD represents an averaging of the driving force since the temperature difference between the two streams as they flow through the exchanger.

The LMTD method can be used for the heat exchanger design when the mass flow rates, the inlet and outlet temperatures of the hot and cold fluids are specified [26]. This method assumes that K is constant along the heat exchanger. In the case of STHE, LMTD must be multiplied by a correction factor (F). Then eq. (1) takes the following form:

$$\Phi = KAF\, \Delta T_{lm} \tag{5}$$

The correction factor F is empirically determined at a mean temperature between the inlet and the outlet of the heat exchanger, or can also be expressed as a function of the heat flow ratio (R) and the heat exchanger efficiency (G).

When the LMTD method cannot be used, the G-NTU seems to be more suitable and will simplify a number of heat exchanger design problems. The efficiency G is defined as a ratio of the real heat flow exchanged (Φ) and the maximum heat flow exchanged (Φ_{max}) [27].

$$G = \frac{\Phi}{\Phi_{max}} \tag{6}$$

The heat flow ratio (R) and the number of transfer units (NTU) are expressed in the following equations:

$$R = \frac{(QC_p)_{min}}{(QC_p)_{max}} \tag{7}$$

with QC_p is the heat capacity flow rate.

$$NTU = \frac{KA}{(QC_p)_{min}} \tag{8}$$

3- Mathematical modeling of thermal hydraulic behavior for CCFSTHE

In this work, the following assumptions are taken into account: The global heat transfer coefficient (K) and the thermo-physical properties (density, specific heat) vary along the heat exchanger, the cold fluid circulating on the tube side, and the hot fluid cooled in the same way as the cold fluid is heated, there is no loss of heat outside the exchanger and the transfer occurs without phase change.

The heat flow of the hot fluid (Φ^h) is expressed using the following equation:

$$\Phi^h = -q^h c_p^h (T^h - T_i^h) \tag{9}$$

The heat flow of the cold fluid (Φ^c), is written as follows:

$$\Phi^c = q^c c_p^c (T_f^c - T^c) \tag{10}$$

where q^h and q^c are the mass flow rates, T^h and T_i^h are the temperatures of the hot fluid at the studied local point and the inlet, respectively. T_f^c and T^c are the temperatures of the cold fluid at the outlet and the local studied point, and c_p^h and c_p^c are the hot and cold fluids specific heat capacities.

The local form of the heat transfer equation is given by:

$$-q^h c_p^h (T^h - T_i^h) = q^c c_p^c (T_f^c - T^c) = ks\Delta T \tag{11}$$

s is the element surface corresponding to the ΔT.

By differentiating and developing the equation above, we find out that:

$$k\,s = \frac{1}{m}(\theta - 1) \tag{12}$$

with $\theta = \frac{T_i^h - T_f^c}{T^h - T^c}$ and $m = \frac{1}{q^h c_p^h} - \frac{1}{q^c c_p^c}$

By extrapolating Eq. (12) to the total size of the heat exchanger, we obtain:

$$MKA = \Theta - 1 \tag{13}$$

with $\Theta = \frac{T_i^h - T_f^c}{T_f^h - T_i^c}$ and $M = \frac{1}{Q^h C_p^h} - \frac{1}{Q^c C_p^c}$

Θ is the hot and cold fluids temperature differences ratio between the initial state and the final state.

The term above M is expressed as function of the cold and hot fluids heat capacity flow rate (QC_p). It can also be expressed as a function of the thermal and hydraulic parameters and the compactness of the heat exchanger, as follow:

$$M = \frac{4}{D_{eq}\lambda R_e^h P_r^h} - \frac{4}{D_i \lambda R_e^c P_r^c} \tag{14}$$

with λ the thermal conductivity of the tube material, D_i is the inner diameter of tubes, $R_e^h = \rho_h V_h D_{eq}/\mu_h$, $P_r^h = C_p^h \mu_h/\lambda$ are Reynolds and Prandtl numbers of hot fluid, $R_e^C = \rho_C V_C D_i/\mu_C$, $P_r^C = C_p^c \mu_C/\lambda$ are Reynolds and Prandtl numbers of cold fluid, V is the fluid velocity, ρ is the density, μ is the fluid viscosity and D_{eq} is the equivalent diameter which can be written as a square layout of the following form:

$$D_{eq} = \frac{4(P_t^2 - \frac{\pi D_e^2}{4})}{\pi D_e} \tag{15}$$

For a triangular layout, D_{eq} is given by:

$$D_{eq} = \frac{4(\frac{\sqrt{3}\,P_t^2}{4} - \frac{\pi D_e^2}{8})}{\frac{\pi D_e}{2}} \tag{16}$$

P_t : is the pitch, D_e : is the outside tube diameter.

The dimensionless quantity MKA regroups the various thermal and hydraulic parameters required in analysis and the design of the heat exchanger. As given in eq. 13, this quantity has been simplified and expressed only as a function of Θ (the ratio of the hot and cold fluids temperature differences between the initial and the final states).

Depending on the value of M, and according to our model, we can distinguish three possible cases of exchange:

<u>First case</u>: M is positive ($M > 0: Q^h C_p^h < Q^c C_p^c$)

The Θ, G, R and NTU parameters can be written as follows:

$$G = \frac{T_i^h - T_f^h}{T_i^h - T_i^c} \quad ; R = \frac{Q^h C_p^h}{Q^c C_p^c} = \frac{T_f^c - T_i^c}{T_i^h - T_f^h} < 1 \quad , NTU = \frac{KA}{Q^h C_p^h} \text{ and } \quad 1 < \Theta = \frac{T_i^h - T_f^c}{T_f^h - T_i^c} \leq 2$$

where the hot fluid cooling rate is higher than the cold fluid heating rate (Fig. 3). The temperature difference between hot and cold fluids diminishes along the heat exchanger (T_i^h evolves towards T_i^c (Fig. 3)), i.e., the cooling heat capacity flow rate of hot fluid is greater than that of the heating of cold fluid.

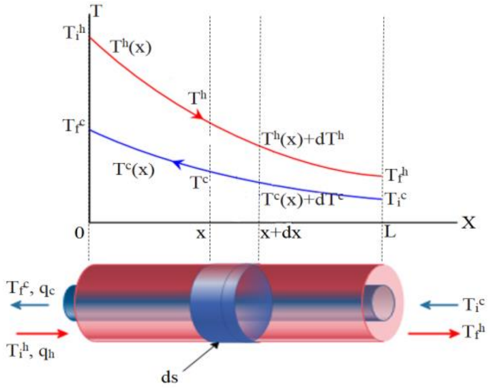

Figure 3. Temperature evolution along the CCFSTHE for M > 0

By writing Θ as a function of R and G, equation (13) becomes:

$$MKA = G\,\frac{1-R}{1-G} \tag{17}$$

with $$\Theta = \frac{1-RG}{1-G} \tag{18}$$

In the first case (M > 0), the variation of the dimensionless parameter MKA with the heat capacity ratio R and the efficiency G is presented in Fig. 4.

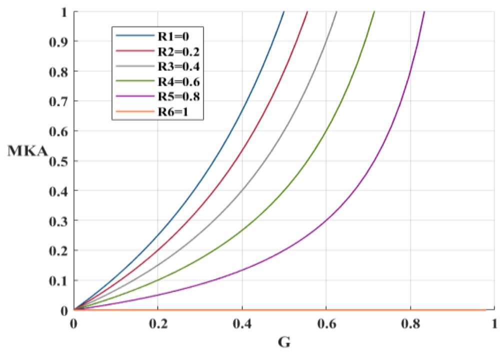

Figure 4. Variation of MKA as a function of G for different values of R

In this case, hot fluid governs the process of the heat transfer. MKA and Θ act in a non-linear form versus G and R (Fig. 4). This nonlinear variation is not important as the temperature difference between hot and cold fluid decreases along the exchanger. The calculation of Θ enables us to deduce MKA. We note that: $1 < \Theta < 2$ and $0 < MKA < 1$.

The following expression of NTU can be inferred from the MKA expression:

$$NTU = \frac{G}{1-G} \tag{19}$$

The NTU depends only on G and its calculation can be performed without introducing any correlation.

It is interest to note that:

$$\frac{MKA}{NTU} = 1 - R \tag{20}$$

<u>Second case</u> : M is negative ($M < 0 : Q^h C_p^h > Q^c C_p^c$)

The parameters Θ, G, R and NTU can be written as follows:

$$G = \frac{T_f^c - T_i^c}{T_i^h - T_i^c} \quad ; \quad R = \frac{Q^c C_p^c}{Q^h C_p^h} = \frac{T_i^h - T_f^h}{T_f^c - T_i^c} < 1 \;,\; NTU = \frac{KA}{Q^c C_p^c} \text{ and } 0 < \Theta = \frac{T_i^h - T_f^c}{T_f^h - T_i^c} \leq 1$$

where the cold fluid heating rate is higher than the hot fluid cooling rate (Fig. 5).

The difference of temperature between the hot and cold fluids rises along the heat exchanger (T_i^c evolves towards T_i^h (Fig. 5)), i.e., the cooling heat capacity flow rate of hot fluid is smaller than that of the heating of cold fluid.

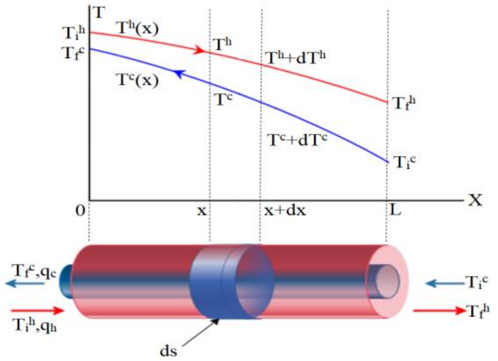

Figure 5. Temperature evolution along the CCFSTHE in the case for M < 0

By writing Θ as a function of R and G, Eq. (13) becomes:

$$MKA = \frac{G - RG}{RG - 1} \tag{21}$$

where

$$\Theta = \frac{1 - G}{1 - RG} \tag{22}$$

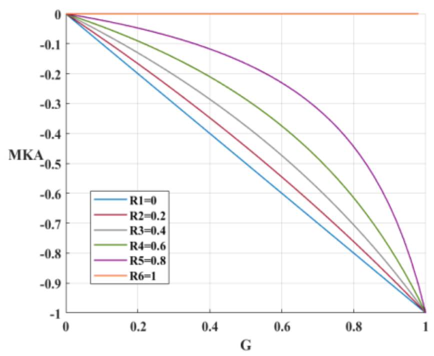

Figure 6: Variation of MKA as a function of G for different values of R

In this case, the process of the heat transfer is governed by the cold fluid. As mentioned in the first case, MKA and Θ show a similar tendency. They behave non-linearly with G and R (Fig. 6). Their non-linear variation is important when the temperature gradient is amplified along the heat exchanger. Thus, it is necessary to take into account this non-linear behavior of MKA and Θ during the thermal hydraulic performance analysis and the design of CCFSTHE. We note that : $0 < \Theta < 1$ and $-1 < MKA < 0$.

The NTU expression is given by this formula:

$$NTU = \frac{G}{1 - RG} \tag{23}$$

NTU is calculated without introducing any correlation.

We notice that $\frac{MKA}{NTU}$ ratio is given in term of R as follows:

$$\frac{MKA}{NTU} = R - 1 \tag{24}$$

<u>Third case</u>: M equal to zero ($M = 0 : Q^h C_p^h = Q^c C_p^c$)

The former parameters R, G and Θ take in this case the following values:

$$R = 1 \quad ; \qquad G = 1 \qquad ; \qquad \text{and} \qquad \Theta = \frac{T_i^h - T_f^c}{T_f^h - T_i^c} = 1$$

When M is approximately zero, the heat capacity rates of the two fluids and the temperature difference between hot and cold fluids are equal throughout the heat exchanger. In this case, the conventional methods are sufficient for the CCFSTHE design, because the effect temperature is not important.

4- Comparison between the current model of G and the model reported in literature as a function of NTU

In the heat exchanger, the conventional method fails to give an accurate performance analysis due to significant variations in thermal and hydraulic properties. The expressions of efficiency G extracted from the literature (Eq. 25) [27] and from our current models (Eqs. 26 and 27) are written in the following.

$$G = \frac{1-\exp(-NTU(1-R))}{1-R\times\exp(-NTU(1-R))} \tag{25}$$

$$G = \frac{NTU}{1+NTU} \qquad \text{for} \qquad M > 0 \tag{26}$$

$$G = \frac{NTU}{1+R\times NTU} \qquad \text{for} \qquad M < 0 \tag{27}$$

The comparison of our models and model available in literature are represented in Fig. 7 where the efficiency G as a function of NTU for different values of R.

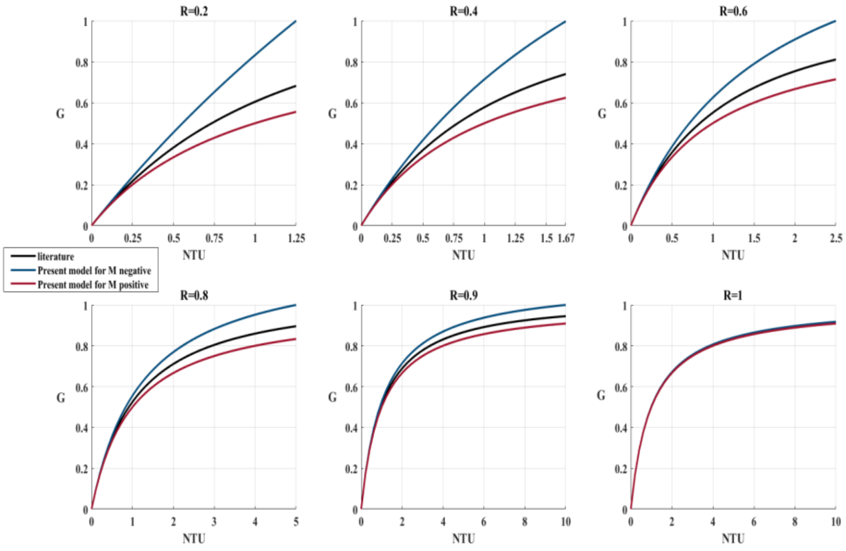

Figure 7. Eefficiency calculations by the present models versus the literature model in terms of NTU and R

G is represented in different corresponding NTU intervals. One can distinguish between two regimes depending to NTU values. The first regime is linear and is obtained for low NTUs (and also for low efficiency). In this regime, all the curves supplying G coincide and the type of the heat exchanger and fluids flow directions are no longer important. These results are in a good agreement with the literature [23, 24]. In the second regime, we clearly observe that the change of G has the same trend for all models. However, the figure 7 shows a significant deviation of our models data from literature. This deviation can be attributed to the non-linear dependence on temperature of thermo-physical properties and the overall coefficient of heat transfer. The efficiency G spreads out in the second case (M < 0) more than in the first (M > 0). One can also notice that the difference in G between our models and the literature model diminishes for higher R, where the heat exchange is favorable. Around $R \sim 1$, the three models provide again the same value of G, because the temperature effect becomes insignificant (Fig. 7). These results are also in a good agreement with the literature of the previous work theoretical and experimental [23, 24].

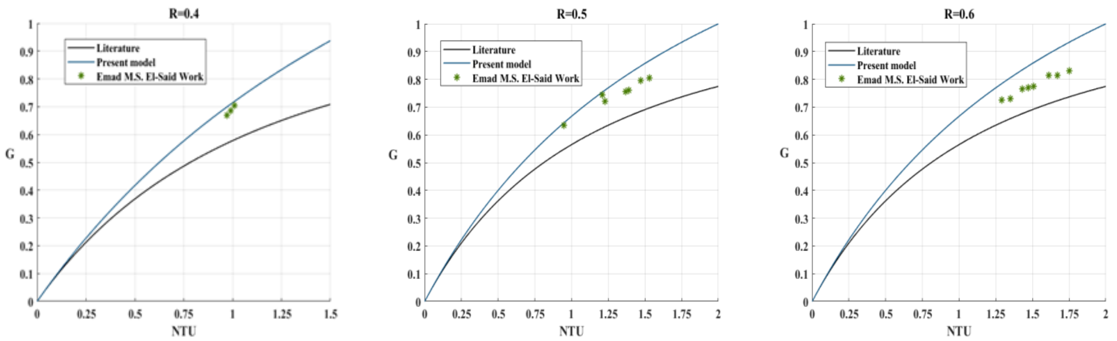

Figure 8. Comparison between our models, literature model and experiment [24]

Hereafter, we have accomplished the validation of our proposed procedure and results of efficiency G by comparing them to the experimental results realised by Emad M.S. et al. [24] (Fig. 8). In their experiments, they studied the effect of air injection on the thermal performance of STHE aiming at increasing the thermal performance for different air flow rates and to estimate optimal performance conditions.

Our expression of efficiency G for M < 0, depicted in figure 8, is roughly in agreement with the experiment of Emad M.S. et al [24]. This concordance is more obvious when R decreases (R less than 0.6) where the temperature effect becomes dominant. Hence, for low values of R, using either LMTD or NTU conventional methods may result in inaccurate performance of thermal hydraulic analysis and design of CCFSTHE.

For R values close to 1, our results based on our model agree with those obtained by Magazoni, F. C., et al [23], particularly for M < 0 and R equal 0,7 and 1.

5- Comparison between present models of G and the model reported in literature as a function of MKA

Cause, MKA take into account all thermo-physical properties as mentioned above, we can express G as a function of MKA and the Eqs. (25, 26 and 27) will be as follows:

$$G = \frac{1-\exp(-|\mathrm{MKA}|)}{1-R\times\exp(-|\mathrm{MKA}|)} \tag{28}$$

$$M > 0 : G = \frac{|MKA|}{|MKA|+1-R} \tag{29}$$

$$M < 0 : G = \frac{-|\mathrm{MKA}|}{-R\times|\mathrm{MKA}|-1+R} \tag{30}$$

Note that equation (28) can be deduced from Eqs. (20), (21) and (25), Eqs. (29) and (30) can be derived from Eqs. (17) and (21). These expressions are plotted in figure 9.

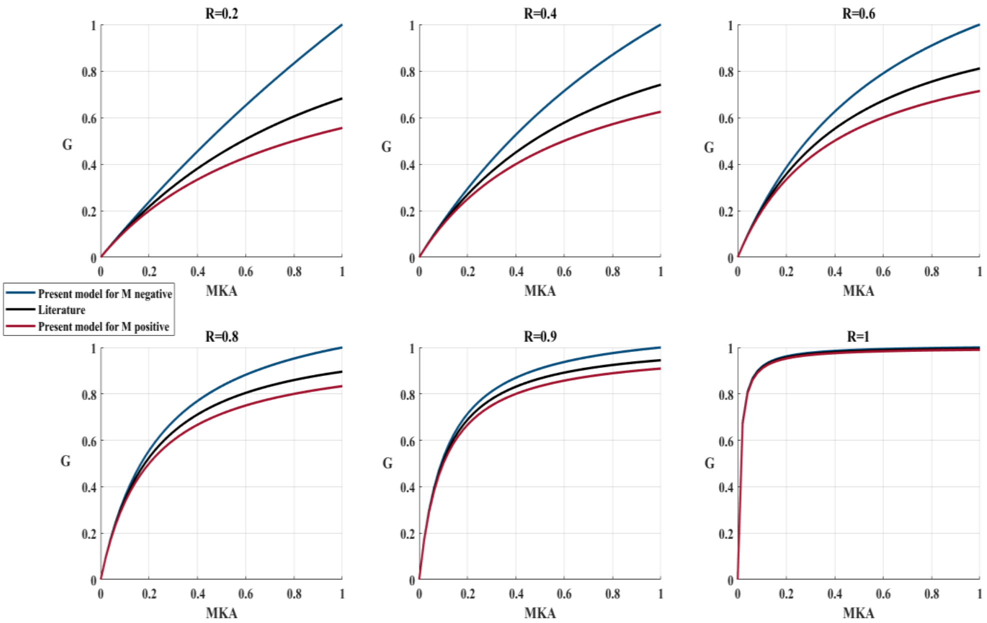

Figure 9. G calculations as function of $|MKA|$ for different R values

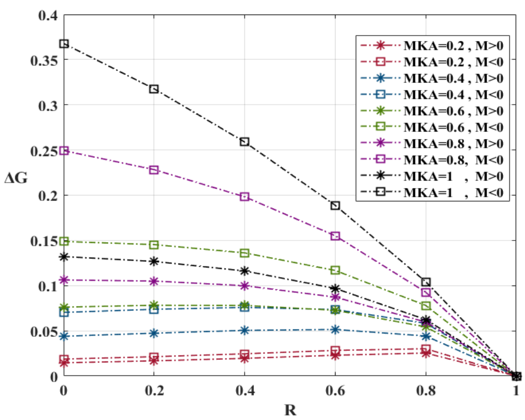

Figure 10. $|\Delta G|$ predictions versus R for different MKA values

The approach developed in the present study for CCFSTHE is based on the determination of MKA and NTU, without impinging the correction factor F.

In order to visualize the change in the magnitude of G, we have illustrated in Fig. 10 $\Delta G = G$ (Our model) - G (standard model) as a function of R for different MKA values.

We can point out that by decreasing R ($M \sim 0$), ΔG becomes more pronounced for $M < 0$ than for $M > 0$. The reason for this behavior is linked, as stated above, to the change of the overall heat transfer coefficient and to the thermo-physical properties as a function of the circulating fluids temperature [19]. The difference ΔG cancels out when R approaches 1. One can conclude that assuming a constant thermo-physical property is no longer valid, especially when a strong change in the temperature gradient occurs along the heat exchanger.

6- Conclusion

As a conclusion, the established present model could be a very useful model in minimizing the estimations related to the calculation of the design parameters for a single pass 1-1 counter current flow shell and tube heat exchanger (CCFSTHE) type TEMA E. In this model a new dimensionless number (MKA) linked to the NTU only by heat flows ratio, without including the correlation coefficient F is established. The expression of the efficiency G obtained, for $M > 0$ and $M < 0$, are compared to those of the conventional method. These comparisons show that our models can reproduce results as reported in the literature. They also demonstrate the importance of temperature effect on thermo-physical properties and the overall transfer coefficient. Our results, based on the approach described above, show a good agreement with experimental results [24], particularly for $M < 0$. However, the results would not change if the hot-cold fluid in our assumptions (hot fluid circulates throughout the tubes) is reversed.

Additionally, the present work allows us to determine new temperature effectiveness and seems to be appropriate, for preliminary analysis of the performance of the thermal and hydraulic properties, and thus for designing CCFSTHE using the effectiveness - MKA method (G-MKA) and reducing the number of tests in the experiment.

The proposed model turns out to be quite complex when the hot fluid is not cooled in the same way as the cold fluid is heated.

Acknowledgment

This manuscript was subjected to a profound scrutiny performed by **Mr. Mohammed SAISSI**, An ELT teacher in Directorate of Safi-Morocco and a PhD student in department of English at the faculty of letters and human sciences in Mohammedia, University Hassan II Casablanca, Morocco. Accordingly, we'd like to thank him for his fruitful contribution to release this paper.

Nomenclature

A : Total heat exchanger area, [m^2]

c_p : Specific heat corresponding to s, [$J.\ kg^{-1}.\ K^{-1}$]

C_p : Specific heat at the outlet of the exchanger, [$J.\ kg^{-1}.\ K^{-1}$]

CCFSTHE : Counter-current flow shell and tube heat exchanger
D : Tube diameter, [m]
D_e : Shell-side hydraulic diameter, [m]
F : Correction factor
G : Efficiency ($=\Phi/\Phi_{max}$)
H : Coefficient of convection, [$W.m^{-2}.K^{-1}$]
k : Local heat transfer coefficient, [$W.m^{-2}.K^{-1}$]
K : Overall heat transfer coefficient, [$W.m^{-2}.K^{-1}$]
L : Tube length, [m]
LMTD: Logarithmic Mean Temperature Difference
n : Number of tubes
NTU : Number of Transfer Units ($=KA/(QC_p)_{min}$)
P : Pitch
Pr : Prandtl number ($=C_p\mu/\lambda$)
q : Local mass flow rate
Q : Total mass flow rate
r : Tube radius
R : Resistance or heat flow ratio ($= (QC_p)_{min}/(QC_p)_{max}$)
Re : Reynolds number ($=\rho VD/\mu$)
s : Surface of a tube section corresponding to ΔT, [m^2]
S : Total area of a tube, [m^2]
STHE : Shell and tube Heat Exchanger
T : Temperature, [K]
TEMA : Tubular Exchanger Manufacturers Association
Δt : Cold side temperature difference, [K]
ΔT : Hot side temperature difference, [K]
ΔT_{lm} : The logarithmic mean temperature difference, [K]
V : velocity, [m/s]

Greek symbols

λ : Thermal conductivity, [W/m.K]
Φ : Heat flow [W]
Θ : The hot and cold fluids temperature differences ratio between the initial state and the final state
ρ : Density, [Kg/m^3]
μ : Fluid viscosity, [$Pa\ s^{-1}$]

Subscripts

e : External
max : Maximum
min : Minimum
i : Inlet, Internal, indoor or initial
f : Final or fouling
eq : Equivalent
o : Outlet
t : Transversal

Superscripts

h : Hot fluid
c : Cold fluid

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Submitted:	10.07.2021
Revised:	17.10.2021
Accepted:	30.10.2021