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 to the heat flow ratio, R. The results based on our models show a pronounced deviation compared to the model reported in the literature (number of transfer units 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 $0 \leq R \leq 1$.

Key words: *heat transfer, single pass 1-1, CCFSTHE, thermal design, modelling, effectiveness, MKA method*

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 [1-3]. 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 CCFSTHE TEMA *E*-type is largely used in various industrial fields such as petro-chemical industry, electrical power production, food preservation, manufacturing indus-

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try, 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 [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 increases 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 be included 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]. Taking into account the temperature effect, the integration of methods leads to better results when compared to other approaches such as Colburn [21] and Roetzel and 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 comparing of our results with both those of the conventional method and the experimental results.

Principal equations governing the design of STHE

Heat exchangers are devices commonly used in a wide range of applications. The most common design of STHE is TEMA *E*-type, 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].

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

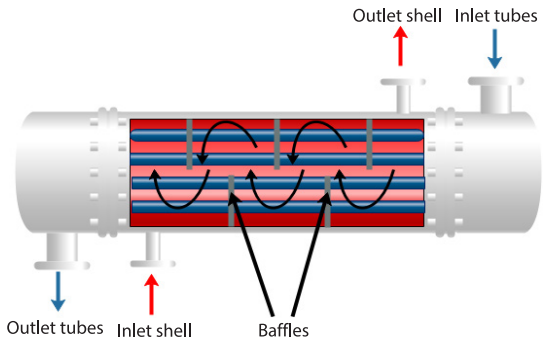

Figure 1. One-pass (1-1) CCFSTHE TEMA *E*-type with baffling

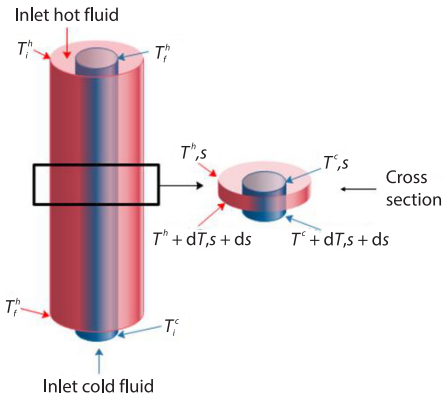

Figure 2. Temperature distribution of fluids in a single tube heat exchanger

The heat exchange calculation is based on [5]:

$$\Phi = KA\,\Delta T_{\mathrm{lm}} \tag{1}$$

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:

$$A = \int_{T_{\mathrm{i}}^{\mathrm{c}}}^{T_{\mathrm{f}}^{\mathrm{c}}} \frac{q^{\mathrm{c}} c_p^{\mathrm{c}}}{k\Delta T}\,\mathrm{d}T^{\mathrm{c}} \tag{2}$$

where q^{c}, c_p^{c}, and T^{c} are, respectively, local mass-flow, specific heat and temperature of cold fluid, corresponding to the rectangular image shown in fig. 2 and k – the local heat transfer coefficient.

The overall 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:

$$K = \frac{1}{\dfrac{A}{A_{\mathrm{i}}h_{\mathrm{i}}} + \dfrac{A}{A_{\mathrm{i}}}R_{\mathrm{f,i}} + \dfrac{A}{2\pi L\lambda}\ln\dfrac{r_{\mathrm{e}}}{r_{\mathrm{i}}} + \dfrac{A}{A_{\mathrm{e}}}R_{\mathrm{f,e}} + \dfrac{A}{A_{\mathrm{e}}h_{\mathrm{e}}}} \tag{3}$$

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:

$$\Delta T_{\mathrm{lm}} = \frac{\Delta T_{\mathrm{i}} - \Delta T_{\mathrm{o}}}{\ln\dfrac{\Delta T_{\mathrm{i}}}{\Delta T_{\mathrm{o}}}} \tag{4}$$

with

$$\Delta T_i = T_{\mathrm{i}}^{\mathrm{h}} - T_{\mathrm{f}}^{\mathrm{c}} \quad \text{and} \quad \Delta T_{\mathrm{o}} = T_{\mathrm{f}}^{\mathrm{h}} - T_{\mathrm{i}}^{\mathrm{c}}$$

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 form:

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

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, $\Phi_{\max}$ [27]:

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

The heat flow ratio, R, and the NTU are expressed:

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

with QC_p is the heat capacity flow rate:

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

Mathematical modelling of thermal hydraulic behavior for CCFSTHE

In this work, the following assumptions are taken into account: The global 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 are 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:

$$\Phi^{\mathrm{h}} = -\, q^{\mathrm{h}} c_p^{\mathrm{h}} \left(T^{\mathrm{h}} - T_{\mathrm{i}}^{\mathrm{h}}\right) \tag{9}$$

The heat flow of the cold fluid, Φ^{c}, is written:

$$\Phi^{\mathrm{c}} = q^{\mathrm{c}} c_p^{\mathrm{c}} \left(T_{\mathrm{f}}^{\mathrm{c}} - T^{\mathrm{c}}\right) \tag{10}$$

where q^{h} and q_{c} are the mass-flow rates, T^{h} and $T_{\mathrm{i}}^{\mathrm{h}}$ are the temperatures of the hot fluid at the studied local point and the inlet, respectively. The $T_{\mathrm{f}}^{\mathrm{c}}$ 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:

$$-q^{\mathrm{h}} c_p^{\mathrm{h}} \left(T^{\mathrm{h}} - T_{\mathrm{i}}^{\mathrm{h}}\right) = q^{\mathrm{c}} c_p^{\mathrm{c}} \left(T_{\mathrm{f}}^{\mathrm{c}} - T^{\mathrm{c}}\right) = ks\Delta T \tag{11}$$

where s is the element surface corresponding to the ΔT.

By differentiating and developing previous equation, we find out:

$$ks = \frac{1}{m}(\theta - 1) \tag{12}$$

with

$$\theta = \frac{T_{\mathrm{i}}^{\mathrm{h}} - T_{\mathrm{f}}^{\mathrm{c}}}{T^{\mathrm{h}} - T^{\mathrm{c}}} \text{ and } m = \frac{1}{q^{\mathrm{h}} c_p^{\mathrm{h}}} - \frac{1}{q^{\mathrm{c}} c_p^{\mathrm{c}}}$$

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

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

with

$$\Theta = \frac{T_{\mathrm{i}}^{\mathrm{h}} - T_{\mathrm{f}}^{\mathrm{c}}}{T_{\mathrm{f}}^{\mathrm{h}} - \mathrm{T}_{\mathrm{i}}^{\mathrm{c}}} \text{ and } M = \frac{1}{Q^{\mathrm{h}} C_p^{\mathrm{h}}} - \frac{1}{Q^{\mathrm{c}} C_p^{\mathrm{c}}}$$

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

The term M is expressed as a 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:

$$M = \frac{4}{D_{\mathrm{eq}} \lambda \, \mathrm{Re}^{\mathrm{h}} \, \mathrm{Pr}^{\mathrm{h}}} - \frac{4}{D_{\mathrm{i}} \lambda \, \mathrm{Re}^{\mathrm{c}} \, \mathrm{Pr}^{\mathrm{c}}} \tag{14}$$

where λ is the thermal conductivity of the tube material, D_{i} – the inner diameter of tubes, $\mathrm{Re}^{\mathrm{h}} = \rho_{\mathrm{h}} V_{\mathrm{h}} D_{\mathrm{eq}}/\mu_{\mathrm{h}}$, $\mathrm{Pr}^{\mathrm{h}} = C_p^{\mathrm{h}} \mu_{\mathrm{h}}/\lambda$ are Reynolds and Prandtl numbers of hot fluid, $\mathrm{Re}^{\mathrm{c}} = \rho_{\mathrm{c}} V_{\mathrm{c}} D\mathrm{i}/\mu_{\mathrm{c}}$, $\mathrm{Pr}^{\mathrm{c}} = C_p^{\mathrm{c}} \mu_{\mathrm{c}}/\lambda$ are Reynolds and Prandtl numbers of cold fluid, V – the fluid vlocity, ρ – the density, μ – the fluid viscosity, and D_{eq} – the equivalent diameter which can be written as a square lay-out of the form:

$$D_{\mathrm{eq}} = \frac{4\left(P_{\mathrm{t}}^2 - \frac{\pi D_{\mathrm{e}}^2}{4}\right)}{\pi D_{\mathrm{e}}} \tag{15}$$

For a triangular lay-out, D_{eq} is given:

$$D_{\mathrm{eq}} = \frac{4\left(\frac{\sqrt{3}\, P_{\mathrm{t}}^2}{4} - \frac{\pi D_{\mathrm{e}}^2}{8}\right)}{\frac{\pi D_{\mathrm{e}}}{2}} \tag{16}$$

where P_{t} is the pitch and D_{e} – the outside tube diameter.

The dimensionless quantity MKA regroups the various thermal and hydraulic parameters required in the 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:

Case 1. The M is positive ($M > 0$: $Q^{\mathrm{h}}C_p^{\mathrm{h}} < Q^{\mathrm{c}}C_p^{\mathrm{c}}$)

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

$$G = \frac{T_i^{\mathrm{h}} - \mathrm{T}_{\mathrm{f}}^{\mathrm{h}}}{T_i^{\mathrm{h}} - \mathrm{T}_{\mathrm{i}}^{\mathrm{c}}},\ R = \frac{Q^{\mathrm{h}}C_p^{\mathrm{h}}}{Q^{\mathrm{c}}C_p^{\mathrm{c}}} = \frac{T_{\mathrm{f}}^{\mathrm{c}} - T_{\mathrm{i}}^{\mathrm{c}}}{T_i^{\mathrm{h}} - T_{\mathrm{f}}^{\mathrm{h}}} < 1,\ NTU = \frac{KA}{Q^{\mathrm{h}}C_p^{\mathrm{h}}} \text{ and } 1 < \Theta = \frac{T_{\mathrm{i}}^{\mathrm{h}} - T_{\mathrm{f}}^{\mathrm{c}}}{T_{\mathrm{f}}^{\mathrm{h}} - T_{\mathrm{i}}^{\mathrm{c}}} \le 2$$

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, $T_{\mathrm{i}}^{\mathrm{h}}$ evolves towards $T_{\mathrm{i}}^{\mathrm{c}}$, fig. 3, *i.e.*, the cooling heat capacity flow rate of hot fluid is greater than that of the heating of cold fluid.

By writing Θ as a function of R and G, eq. (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 R and the efficiency G is presented in fig. 4.

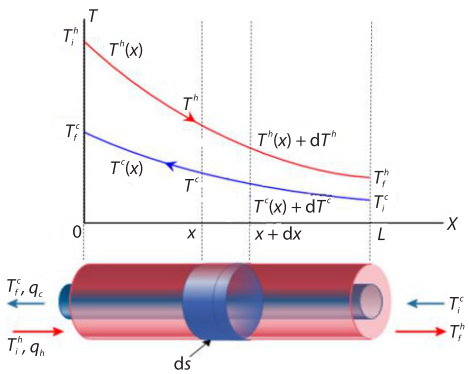

Figure 3. Temperature evolution along the CCFSTHE for $M > 0$

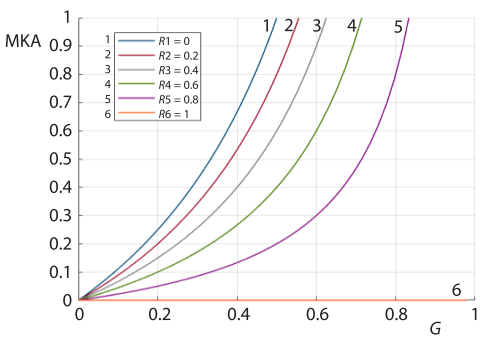

Figure 4. Variation of MKA as a function of G for different values of R

In this case, hot fluid governs the process of the heat transfer. The MKA and Θ act in a non-linear form *vs.* G and R, fig. 4. This non-linear 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 interested to note that:

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

Case 2. The M is negative ($M < 0$: $Q^{\mathrm{h}}C_p^{\mathrm{h}} > Q^{\mathrm{c}}C_p^{\mathrm{c}}$)

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

$$G = \frac{T_{\mathrm{f}}^{\mathrm{c}} - T_{\mathrm{i}}^{\mathrm{c}}}{T_{\mathrm{i}}^{\mathrm{h}} - T_{\mathrm{i}}^{\mathrm{c}}}, \quad R = \frac{Q^{\mathrm{c}}C_p^{\mathrm{c}}}{Q^{\mathrm{h}}C_p^{\mathrm{h}}} = \frac{T_{\mathrm{i}}^{h} - T_{\mathrm{f}}^{\mathrm{h}}}{T_{\mathrm{f}}^{\mathrm{c}} - T_{\mathrm{i}}^{\mathrm{c}}} < 1, \quad NTU = \frac{KA}{Q^{\mathrm{c}}C_p^{\mathrm{c}}} \quad \text{and} \quad 0 < \Theta = \frac{T_{\mathrm{i}}^{\mathrm{h}} - T_{\mathrm{f}}^{\mathrm{c}}}{T_{\mathrm{f}}^{\mathrm{h}} - T_{\mathrm{i}}^{\mathrm{c}}} \le 1$$

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, $T_{\mathrm{i}}^{\mathrm{c}}$ evolves towards $T_{\mathrm{i}}^{\mathrm{h}}$, fig. 5, *i.e.*, the cooling heat capacity flow rate of hot fluid is smaller than that of the heating of cold fluid.

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

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

where

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

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 $-1 < MKA < 0$.

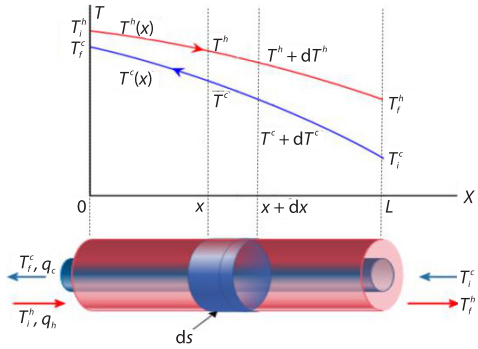

Figure 5. Temperature evolution along the CCFSTHE in the case for $M < 0$

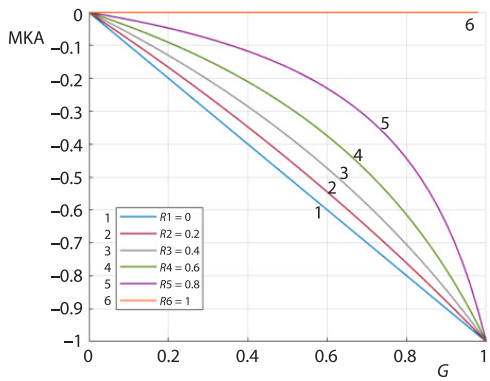

Figure 6. Variation of *MKA* as a function of *G* for different values of *R*

The *NTU* expression is given:

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

where NTU is calculated without introducing any correlation.

We notice that MKA/NTU ratio is given in term of R:

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

Case 3. The M equal to zero ($M = 0$: $Q^{\mathrm{h}}C_p^{\mathrm{h}} = Q^{\mathrm{c}}C_p^{\mathrm{c}}$)

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

$$R = 1, \quad G = 1, \quad \Theta = \frac{T_{\mathrm{i}}^{\mathrm{h}} - T_{\mathrm{f}}^{\mathrm{c}}}{T_{\mathrm{f}}^{\mathrm{h}} - T_{\mathrm{i}}^{\mathrm{c}}} = 1$$

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.

Comparison between the current model of *G* and the model reported in the 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:

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

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

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

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

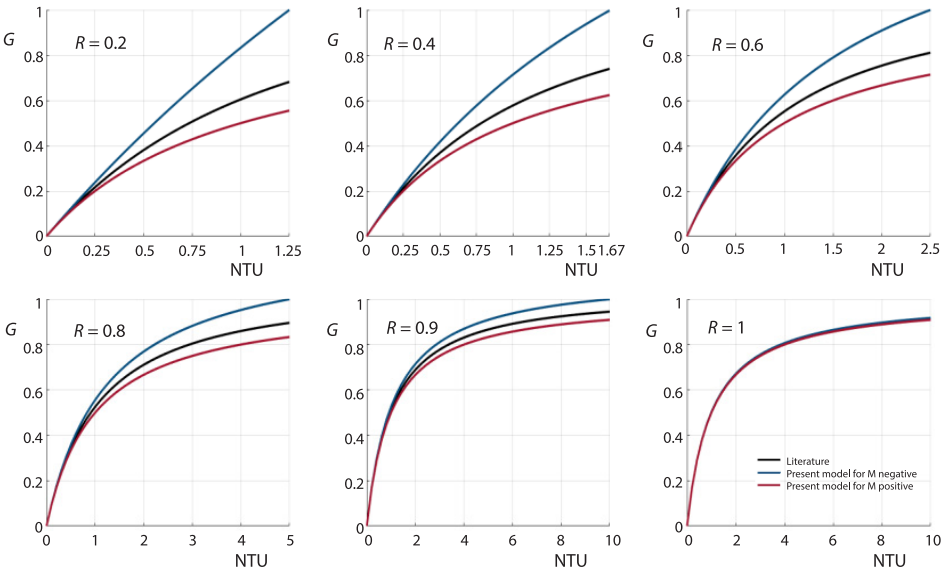

Figure 7. Eefficiency calculations by the present models *vs.* the literature model in terms of NTU and *R*

The 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 NTU (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, fig. 7 shows a significant de-

viation of our models data from the literature. This deviation can be attributed to the non-linear dependence on the 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 theoretical and experimental work [23, 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 *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.

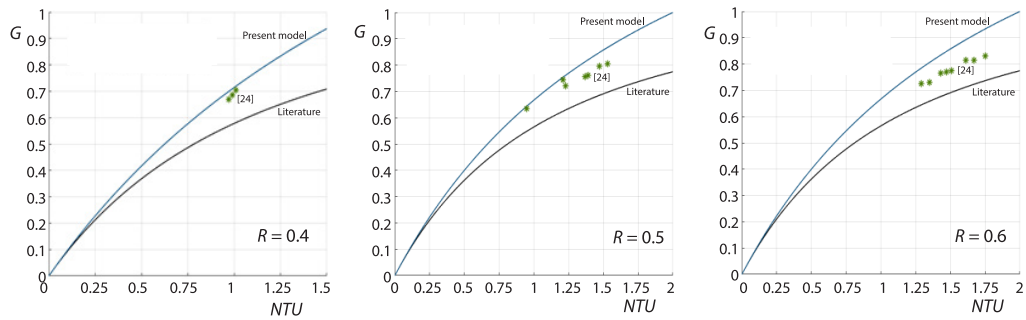

Figure 8. Comparison between our models, the literature model and the experiment [24]

Our expression of efficiency G for $M < 0$, depicted in fig. 8, is roughly in agreement with the experiment of Emad *et al*. [24]. This concordance is more obvious when R decreases (R less than 0.6) and 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 *et al*. [23], particularly for $M < 0$ and R equal to 0.7 and 1.

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

Seeing that MKA takes into account all aforementioned thermo-physical properties, we can express G as a function of MKA and the eqs. (25)-(27) will be:

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

$$G = \frac{\left|MKA\right|}{\left|MKA\right|+1-R} \quad \text{for} \quad M > 0 \tag{29}$$

$$G = \frac{-\left|MKA\right|}{-R\times\left|MKA\right|-1+R} \quad \text{for} \quad M < 0 \tag{30}$$

Note that eq. (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 fig. 9.

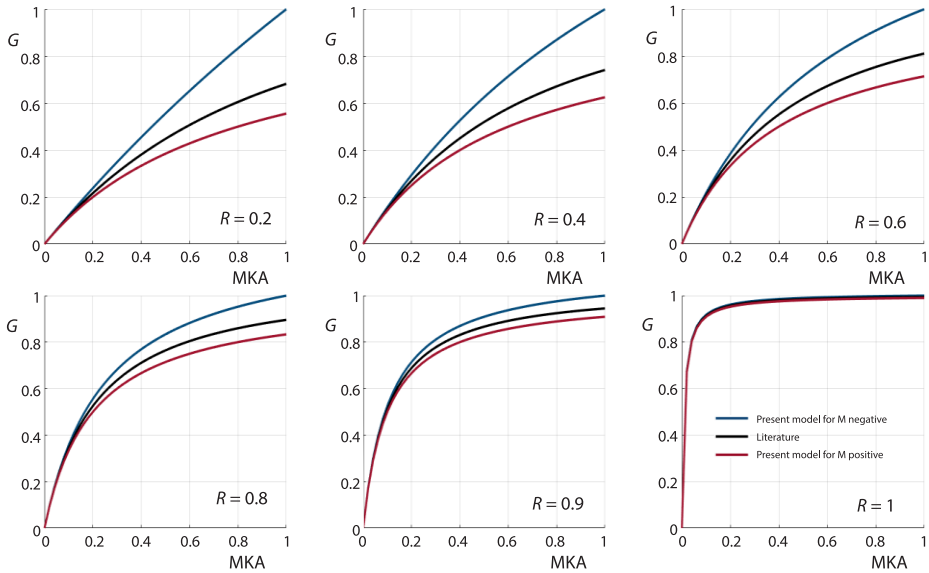

Figure 9. The G calculations as function of |*MKA*| for different *R* values

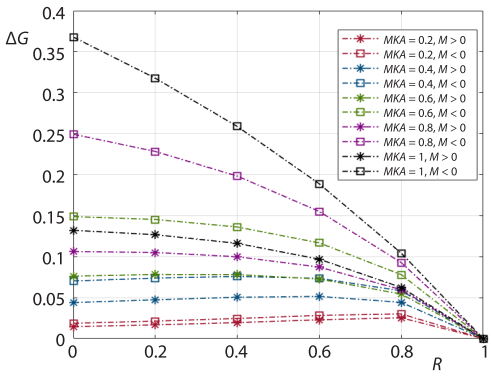

Figure 10. The | Δ*G*| predictions *vs. R* for different MKA values

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 previously, 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.

Conclusions

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 CCFSTHE TEMA *E*-type. 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 previously, 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.

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Nomenclature

A – total heat exchanger area, [m^2]
C_p – specific heat at the outlet of the exchanger, [Jkg^{-1}K^{-1}]
c_p – specific heat corresponding to s, [Jkg^{-1}K^{-1}]
D – tube diameter, [m]
D_e – shell-side hydraulic diameter, [m]
F – correction factor
G – efficiency ($= \Phi/\Phi_{max}$)
h – coefficient of convection, [Wm^{-2}K^{-1}]
K – overall heat transfer coefficient, [Wm^{-2}K^{-1}]
k – local heat transfer coefficient, [Wm^{-2}K^{-1}]
L – tube length, [m]
NTU – number of transfer units [$= KA/QC_p)_{min}$]
P – pitch
Pr – Prandtl number ($= C_p/\mu\lambda$)
Q – total mass-flow rate
q – local mass-flow rate
R – resistance or heat flow ratio [$= QC_p)_{min}/ QC_p)_{max}$]
Re – Reynolds number ($= \rho VD/\mu$)
r – tube radius
s – surface of a tube section corresponding to ΔT, [m^2]
T – temperature, [K]
ΔT – hot side temperature difference, [K]
ΔT_{lm} – logarithmic mean temperature difference, [K]
V – velocity, [ms^{-1}]

Greek symbols

Θ – hot and cold fluids temperature differences ratio between the initial state and the final state
λ – thermal conductivity, [Wm^{-1}K^{-1}]
μ – fluid viscosity, [Pas^{-1}]
ρ – density, [Kgm^{-3}]
Φ – heat flow [W]

Subscripts

e – external
eq – equivalent
i – inlet, internal, indoor or initial
f – final or fouling
max – maximum
min – minimum
o – outlet
t – transversal

Superscripts

h – hot fluid
c – cold fluid

Acronyms

CCFSTHE – counter-current flow shell and tube heat exchanger
LMTD – logarithmic mean temperature difference
STHE – shell and tube heat exchanger
TEMA – tubular exchanger manufacturers association

References

[1] Yang, J., *et al.*, Optimization of Shell-and-Tube Heat Exchangers Conforming to TEMA Standards with Designs Motivated by Constructal Theory, *Energy Convers Manage*, *78* (2014), Feb., pp. 468-476
[2] Wang, Y., *et al*., Experimental Study on the Heat Transfer and Resistance Characteristics of Pin-Fin Tube, *Thermal Science*, *25* (2021), 1A, pp. 59-72
[3] Xinting, W., *et al.*, Numerical Analysis and Optimization Study on Shell-Side Performances of a Shell and Tube Heat Exchanger with Staggered Baffles, *International Journal of Heat and Mass Transfer*, *124* (2018) Sept., pp. 247-259
[4] Ponce-Ortega, J. M., *et al*. Use of Genetic Algorithms for the Optimal Design of Shell-and-Tube Heat Exchangers, *Appl. Therm. Eng*., *29* (2009), 2-3, pp. 203-211
[5] Markowski, M., *et al*., Identification of the Influence of Fouling on the Heat Recovery in a Network of Shell and Tube Heat Exchangers, *Appl Energy*, *102* (2013), Feb., pp. 755-764
[6] Taborek, J., Charts for Mean Temperature Difference in Industrial Heat Exchanger Configuration, in: *Schlünder, E.U*. (Editor-in Chief), Heat Exchanger Design Handbook, Hemisphere Publishing Corporation, New York, USA, 1.5.3-8, 1983
[7] Pal, E. Kumar, I., *et al.*, The CFD Simulations of Shell-Side Flow in a Shell-and-Tube Type Heat Exchanger with and Without Baffles, *Chem. Eng. Sci*., *143* (2016), Apr., pp. 314-340
[8] Manglik, R. M., Bergles, A. E., Swirl Flow Heat Transfer and Pressure Drop with Twisted-Tape Inserts, *Advances in Heat Transfer*, *36* (2003), pp. 183-266
[9] Thome, J. R., On Recent Advances in Modelling of Two-Phase Flow and Heat Transfer, *Heat Transfer Engineering*, *24* (2003), 6, pp. 46-59
[10] Saffarian, M. R., *et al*., Numerical Study of Shell and Tube Heat Exchanger with Different Cross-Section Tubes and Combined Tubes, *International Journal of Energy and Environmental Engineering*, *10* (2019), Feb., pp. 33-46
[11] Mellal, M., *et al*., Hydro-Thermal Shell-Side Performance Evaluation of a Shell and Tube Heat Exchanger under Different Baffle Arrangement and Orientation, *International Journal of Thermal Sciences*, *121* (2017), Nov., pp. 138-149
[12] Abdelkader, B. A., *et al*., The Effect of a Number of Baffles on the performance of Shell-and-Tube Heat Exchangers, *Heat Transfer Engineering*, *40* (2019), 1-2, pp. 39-52
[13] El Maakoul, A., *et al*., Numerical Comparison of Shell-Side Performance for Shell and Tube Heat Exchangers with Trefoil-Hole, Helical and Segmental Baffles, *Applied Thermal Engineering*, *109* (2016), Part A, pp. 175-185
[14] Allen, B., Gosselin, L., Optimal Geometry and Flow Arrangement for Minimizing the Cost of Shell-and-Tube Condensers, *Int. J. Energy Res*., *32* (2008), 10, pp. 958-969
[15] Sahin, A. S., *et al*., Design and Economic Optimization of Shell and Tube Heat Exchangers Using Artificial Bee Colony (ABC) Algorithm, *Energy Conversion and Management*, *52* (2011), 11, pp. 3356-3362
[16] Vahdat, A. A., Amidpour, M., Economic Optimization of Shell and Tube Heat Exchanger Based on Constructal Theory, *Energy*, *36* (2011), 2, pp. 1087-1096
[17] Mirzaei, M., *et al*., Multi-Objective Optimization of Shell-and-Tube Heat Exchanger by Constructal Theory, *Applied Thermal Engineering*, *125* (2017), Oct., pp. 9-19
[18] Aydin, A., *et al*., Optimization and CFD Analysis of a Shell-and-Tube Heat Exchanger with a Multi Segmental Baffle, *Thermal Science*, *26* (2022), 1A, pp. 1-12
[19] Periyasamy Chokkeyee, M. K., *et al*., Analysis on Thermal and Flow Behavior of Triple Concentric Tube Heat Exchanger Handling MWCNT-Water Nanofluids, *Thermal Science*, *24* (2020), 1B, pp. 487-494
[20] Elouardi, E., *et al*., A Novel Approch for Thermal Designing a Single Pass Counter Flow Shell and Tube Heat Exchanger, *International Journal of Mechanical and Production Engineering Research and Development (IJMPERD)*, *10* (2020), 3, 269-280
[21] Colburn, A. P., Mean Temperature Difference and Heat Transfer Coefficient in Liquid Heat Exchangers, *Industrial & Engineering Chemistry*, *25* (1933), 8, pp. 873-877
[22] Roetzel, W., Spang, B., Fundamentals of Heat Exchanger Design – Thermal Design of Heat Exchangers – Chapter C1, in: *VDI-GVC* (Ed.), VDI Heat Atlas, 2nd ED., Springer-Verlag Berlin Heidelberg, Dusseldorf, Germany, 2010, pp. 33-66
[23] Magazoni, F. C., *et al*., Thermal Performance of one-Pass Shell-and-Tube Heat Exchangers in Counter-Flow, *Brazilian Journal of Chemical Engineering*, *36* (2019), 2, pp. 869-883
[24] Emad, M. S., *et al*., Experimental Investigation of Air Injection Effect on the Performance of Horizontal Shell and Multi-Tube Heat Exchanger with Baffles, *Applied Thermal Engineering*, *134* (2018), Apr., pp. 238-247

[25] Mashhour, A. A., Safaei, M. R., Combination Effect of Baffle Arrangement and Hybrid Nanofluid on Thermal Performance of a Shell and Tube Heat Exchanger Using 3-D Homogeneous Mixture Model, *Mathematics*, *9* (2021), 8, 881
[26] Shah, R. K., Sekulic, D. P., Non-Uniform Overall Heat Transfer Coefficients in Conventional Heat Exchanger Design Theory – Revisited, *Journal of Heat Transfer ASME*, *120* (1998), 2, pp. 520-525
[27] Ramesh K. S., Dusan P. S., *Fundamentals of Heat Exchanger Design*, John Wiley and Sons. New York, USA, 2003

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