A NOVEL APPROACH FOR THERMAL DESIGNING A SINGLE PASS COUNTER FLOW SHELL AND TUBE HEAT EXCHANGER

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ABSTRACT

In the Counter Flow Shell and Tube heat Exchanger (CFSTHE), many difficulties are encountered starting from the design until commissioning. These difficulties are mainly related to calculations based on empirical models simplified by correlations. These models are validated on laboratory scale in which heat exchanger is small. Nevertheless, industrial up-scaling involves an increase of calculation mismatch and errors due to the non-linear variation of the thermo physical parameters along the heat exchanger.

In this work, we study the coupling between the following parameters: total heat transfer coefficient, exchange surface and thermo physical properties of hot and cold fluids. Most of the previous work assumed that these parameters are constants [1]. Here, we considered those parameters along the heat exchanger variables without introducing the correction factor (F) in the HAUSBRAND formula. This modelling leads to new mathematical relationships MKA (a dimensionless number) and NTU coupling classical quantities (K and A) and thermo physical properties, appropriate for computerized preliminary analysis and design procedures. Our results are confronted to others in literature, it shows an acceptable agreement.

KEYWORDS: CFSTHE, TEMA E, Design, Modeling, MKA & NTU

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NOMENCLATURE

A: Total heat exchanger area;

c_p : Specific heat corresponding to s;

C_p : Specific heat at the outlet of the exchanger;

D: Tube diameter;

D_c : Shell-side hydraulic diameter;

D_t : Cold side temperature difference;
 DT : Hot side temperature difference;
 F : The correction factor;
 G : The efficiency;
 H : Coefficient of convection;
 k : Local transfer coefficient;
 K : Global transfer coefficient;
 L : Tube length;
 n : Number of tubes;
 NTU : Number of Transfer Units;
 Pr : Prandtl number;
 q : Local mass flow rate;
 Q : Total mass flow rate;
 r : Tube radius;
 R : Ratio of heat flow or resistance;
 Re : Reynolds number;
 s : Surface of a tube section corresponding to ΔT ;
 S : Total area of a tube;
 $STHE$: Shell and tube Heat Exchanger;
 T : Temperature;
 $TEMA$: Tubular Exchanger Manufacturers Association;
 ΔT : The hot fluid and cold fluid temperature difference;
 ΔT_{lm} : The logarithmic mean temperature difference;
 λ : Thermal conductivity;
 Φ : Heat flow;

Subscripts

e : External;
 max : Maximum;
 min : Minimum

i: Internal, indoor or initial;

f: Final or fouling;

eq: Equivalent;

t: Transversal;

Superscripts

h: Hot fluid;

c: Cold fluid.

1. INTRODUCTION

Heat exchangers are widely used in industry, transportation and residential as well [1-3]. 90% of energy transfers are made of heat exchangers (HE) [4-6]. Nowadays, because of a relative increase of the energy cost, it is necessary to find out a maximum efficiency of an installation at low energy. HE is available under different configurations. Depending on their applications, process fluids and heat transfer mode, the fluids can flow in parallel, counter current or cross flow.

Tube and shell heat exchangers are among the most used thermal equipment in many fields, particularly in energy conversion systems; they count for more than 40% of the market share. The most common design of shell-and-tube heat exchanger (STHE) is the type TEMA E due to its design simplicity, robustness and the good performance exchanging power. It can be constructed in a variety of geometries to operate at pressures up to 400 bars at 800°C [1, 2, 7-9].

There are several formulas of NTU, F, R and of G for CFSTHE. This is due to the variation of the temperatures of the hot and cold fluids along the exchange, and therefore the thermo physical properties.

In spite of a big number of analytical and experimental works performed on HE they are either restricted to some aspects [10] or neglected the coupling between different involved parameters.

Experimentally, several studies have been carried out to study effect of one thermo physical property on transfer coefficient [1, 11 - 14]. Other studies have been developed taking into account the dependence of temperature, heat transfer coefficient and position of the heat exchanger [15, 16]. The overall outcome of these studies have led to mathematical correlations and approximated models. as Also, the thermo-physical properties vary non-linearly with the temperatures of the hot and cold fluids [17-22].

To this end, many researchers used objective functions and algorithms to optimize the CFSTHE and then minimize the total cost and a drop of pressure or to maximize the heat transfer [1, 19, 23 - 28]. Moreover, some researchers have optimized the shell and tube heat exchanger efficiency using many design parameters such as heat transfer area, tube length and tube diameter, tube layout, number of shell and passes, number of tubes, tube pitch, spacing ratio baffles and cutting ratio, shell and tube side pressure drop [20, 29 and 30]. This paper reveals the effects various parameters in the thermal designing of counter flow shell and tube heat exchanger.

In this work, we introduce a new dimensionless number coupling the thermo physical properties with the classical quantities (K and A) of the heat exchanger. This leads us to a novel approach to design of the ECSF.

2. HEAT EXCHANGER DESIGN FORMULATION

Shell and tube heat exchangers consist of a series of narrow tubes arranged inside larger tubes. They are easy to build and provide a good ratio between heat transfer and exchange surface. The metal shell and tubes are pressurized. They must withstand the specified design pressures during the intended lifetime of the equipment. Baffles are used to direct the shell fluid passing the tubes in such a way that heat transfer is enhanced. Turbulence is caused, higher heat transfer coefficients, and hence higher rates result [1, 2].

Shell and tube heat exchangers can be constructed with different configurations. They are defined by the Tubular Exchanger Manufacturers Association (TEMA). Figure 1 presents an example of a one pass - TEMA E shell and tube and five baffles heat exchanger.

Figure 1: One-Pass (1-1) TEMA E CFSTHE with Five Baffles.

Our study is restricted to investigate E type shell and tubes heat exchanger. A starting point is a single-tube heat exchanger (Figure 2).

Figure 2: Temperature Distribution of Fluids flowing along a Single Tube.

2.1. Principal Equations Governing the Design of Heat Exchangers

The calculation of the heat exchange is based on HAUSBRAND formula:

$$\Phi = KA \Delta T \quad (1)$$

It provides a relationship between heat flux (Φ), exchange surface (A), global transfer coefficient (K) and temperature difference ΔT .

2.1.1. Exchange Surface (A)

Generally, heat exchanger surface A is calculated using the following formula:

$$A = \int_{T_i^c}^{T_f^c} \frac{q^c c_p^c}{k \Delta T} dT^c \quad (2)$$

Where q^c , c_p^c , T^c and k are locale: mass flow, specific heat, temperature and transfer coefficient of cold fluid, respectively.

2.1.2. Global Transfer Coefficient

Global transfer coefficient K is calculated from the thermodynamic or transport considerations:

$$KA = \frac{1}{\frac{1}{A_i h_i} + \frac{R_{fi}}{A_i} + \frac{1}{2\pi L \lambda} \ln \frac{r_e}{r_i} + \frac{R_{fe}}{A_e} + \frac{1}{A_e h_e}} \quad (3)$$

Where h_i , h_e , R_i , R_e , r_i and r_e are the internal and external convective heat transfer, fouling coefficients and tube radius, respectively.

And λ is the thermal conductivity for tube wall.

The global transfer coefficient includes all thermal resistances of fluids, tube material nature and conductivities due to the boundary layers and the fouling considerations. These resistances depend on the hydrodynamic and thermal conditions of the circulating fluid, Therefore, the calculation of K would most likely reflect a many estimation.

To design heat exchangers different compromises are considered, namely:

- To increase the efficiency of heat exchangers a high surface area is desirable, but it results in a high costs.
- To increase the values of the heat exchange coefficients and thus reduce the surface, a low fluid flow sections are used, they also are increasing the pressure drop, though.

2.1.3. Temperature Difference

Temperature deviation represents the effective driving force for a flow in a heat exchanger.

The relationships established in the case of double tube heat exchangers cannot be used directly for shell and tube exchangers. However, the difference in temperature can be obtained by multiplying the logarithmic mean temperature difference (LMTD) relative to the two ends by a corrected factor (F).

$$\Delta T = F \Delta T_{lm} \quad (4)$$

$$\Phi = KAF \Delta T_{lm} \quad (5)$$

With:

$$\Delta T_{lm} = \frac{DT - Dt}{\ln \frac{DT}{Dt}} \quad (6)$$

$$DT = T_i^h - T_f^c \quad (7)$$

And

$$Dt = T_f^h - T_i^c \quad (8)$$

The correction factor is empirically determined at a mean temperature between the inlet and the outlet of the exchanger, which increases the calculation error. If the design indicates a coefficient less than 0.8, baffles are then added. This increases not only the difference in temperature but also the pressure loss. The diagrams of the correction factor depend on the exchanger type, its position, geometry and thermo physical properties of a circulating fluid. This factor is expressed as a function of the heat capacity ratio (R) and the efficiency (G).

2.2. Efficiency, Heat Capacity Ratio and Number of Transfer Units

The efficiency (G) of a shell and tube heat exchanger is a crucial parameter in any design. It is defined as a ratio of the real heat flow exchanged (Φ) and the maximum heat flow exchanged (Φ_{max}) by a heat exchanger [31]:

$$G = \frac{\Phi}{\Phi_{max}} \quad (9)$$

The heat capacity ratio (R) is expressed as:

$$R = \frac{(Qc_p)_{min}}{(Qc_p)_{max}} \quad (10)$$

The NTU is a dimensionless parameter often used in analysis and the design of heat exchangers. In addition, it is needed for find out the efficiency G. This parameter is expressed as:

$$NTU = \frac{KA}{(Qc_p)_{min}} \quad (11)$$

3. MATHEMATICAL MODELING

3.1. Modeling Assumptions

In this work, we took the following considerations: (a) the flow is counter-current, (b) the global heat transfer coefficient and the thermophysical properties vary along the heat exchanger, (c) there is no loss of heat outside, (d) the transfer into the exchanger occurs without phase change, (e) the hot fluid is cooled in the same way as the heating of the cold fluid and (f) the cold fluid flowing on the tube side.

3.2. Heat Flux Equations

The heat flux equation of the hot fluid is written as follow:

$$\Phi^h = -q^h c_p^h (T^h - T_i^h) \quad (12)$$

The heat flow equation of the cold fluid is written in this way:

$$\Phi^c = q^c c_p^c (T_f^c - T^c) \quad (13)$$

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

$$-q^h c_p^h (T^h - T_i^h) = q^c c_p^c (T_f^c - T^c) = ks\Delta T \quad (14)$$

By developing, the equations above we find out that:

$$ks = \frac{1}{m} (\omega - 1) \quad (15)$$

With

$$\omega = \frac{T_i^h - T_f^c}{T^h - T^c} \quad (16)$$

And

$$m = \frac{1}{q^h c_p^h} - \frac{1}{q^c c_p^c} \quad (17)$$

By extrapolating the model (15) to the totality of the exchanger, we obtain:

$$MKA = W - 1 \quad (18)$$

With

$$W = \frac{T_i^h - T_f^c}{T_f^h - T_i^c} \quad (19)$$

And

$$M = \frac{1}{Q^h c_p^h} - \frac{1}{Q^c c_p^c} \quad (20)$$

The dimensionless quantity MKA depends only on the ratio of the temperature difference of the initial and final state.

4. RESULTS AND DISCUSSIONS

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

4- 1. First case $M > 0$: $Q^h c_p^h < Q^c c_p^c$

So we write;

$$W = \frac{T_i^h - T_f^c}{T_f^h - T_i^c} > 1; \quad G = \frac{T_i^h - T_f^h}{T_i^h - T_i^c} \quad ; \quad R = \frac{Q^h c_p^h}{Q^c c_p^c} = \frac{T_f^c - T_i^c}{T_i^h - T_f^h} < 1 \quad \text{And} \quad NTU = \frac{KA}{Q^h c_p^h}$$

Where the hot fluid cooling rate is higher than the cold fluid heating rate: the difference of the temperature is reached by the hot fluid of T_i^h which evolves towards T_i^c , i.e., the heat capacity flow rate of the heated fluid is greater than that of the heating fluid.

Writing W as a function of R and G, the equation (18) becomes:

$$MKA = G \frac{1-R}{1-G} \quad (21)$$

With

$$W = \frac{1-RG}{1-G} \quad (22)$$

Figure 3: Variation of MKA as a Function of R for different values of G with $M > 0$.

Figure 4: Variation of W as a Function of R for different Values of G with $M > 0$.

Where is the discussion of all these figures in text body?

In this case, hot fluid is limiting, MKA and W vary linearly with R for different values of G.

The expression of number of transfer units is written as follows:

$$NTU = \frac{G}{1-G} \quad (23)$$

It depends only on G and its calculation is achieved without introducing any correlation.

We can also note that:

$$\frac{MKA}{NTU} = 1 - R \quad (24)$$

4- 2. Second case $M < 0: Q^h C_p^h > Q^c C_p^c$

So we write;

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

Where the cold fluid heats up more than the warm fluid cools: the cold fluid that experiences higher temperature difference and heat capacity flow rate. It also corresponds to the output temperature of the heated fluid and to the inlet temperature of the heating fluid. Furthermore, we notice that the difference in temperature between the side $s = 0$ is greater than in $s = S$ (Figure 6).

Writing W as a function of R and G , equation (18) becomes:

$$MKA = \frac{G - RG}{RG - 1} \quad (25)$$

$$\text{With } W = \frac{1-G}{1-RG} \quad (26)$$

Figure 5: Variation of MKA according to R for Different Values of G with $M < 0$.

Figure 6: Variation of W according to R for different Values of G with $M < 0$.

In this case, the cold fluid is limiting, MKA and W vary nonlinearly with R for different values of G.

The expression of number of transfer units is given by this formula:

$$NTU = \frac{G}{1 - RG} \quad (27)$$

It is calculated without introducing any correlation.

Note also:

$$\frac{MKA}{NTU} = R - 1 \quad (28)$$

4- 3. Third case: If M is Substantially Zero (Singular Point): $Q^h C_p^h = Q^c C_p^c$

So we write;

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

Where the heat capacity rates of the two fluids and the temperature difference between both sides are equal.

The hot and cold temperature profiles vary linearly. That is to say, the heat capacity flow of the heating fluid varies in the same manner as that of the heated fluid.

The dimensionless number MKA = 0 and the temperature difference remains uniform ($W = 1$). This is a rare case.

The validation of our model in literature shows a good agreement with experimental results [33].

5. CONCLUSIONS

In this work, we minimized the estimates related to the calculation of the design parameters of CFSTHE, by establishing a new dimensionless number (MKA) which is linked to the NTU only by the ratio of heat flows and the exchanger efficiency, without introducing the correlation coefficient (F) and with an improvement of the calculation hypotheses. To calculate MKA and NTU, simply calculate W which depends only on the inlet and outlet temperatures of the heat exchanger. This dimensionless numbers seems to be appropriate for a preliminary analysis to optimize the design of CFSTHE.

This way minimizes the correlation numbers and correction factors used in the calculation. It allows the analysis of the performance of the thermal and hydraulic behavior and the coupling between the overall heat transfer coefficient and the pressure drop. We introduced a novel approach to design a one-pass (1-1) TEMA E - CFSTHE using the NTU method. The obtained NTU expressions are compared with the formulas available in the literature, showing a very good agreement [33].

Our experimental work on phosphoric acid concentration is in progress to validate this current mathematical model.

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