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Thermal characterization of a cross-flow heat exchanger with a new flow arrangement

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A B S T R A C T

The objective of the present paper is to thermally characterize a cross-flow heat exchanger featuring a new cross-flow arrangement, which may find application in contemporary refrigeration and automobile industries. The new flow arrangement is peculiar in the sense that it possesses two fluid circuits extending in the form of two tube rows, each with two tube lines. To assess the heat exchanger performance, it is compared against that for the standard two-pass counter-cross-flow arrangement. The two-part comparison is based on the thermal effectiveness and the heat exchanger efficiency for several combinations of the heat capacity rate ratio, C^*, and the number of transfer units, *NTU*. In addition, a third comparison is made in terms of the so-called "heat exchanger reversibility norm" (*HERN*) through the influence of various parameters such as the inlet temperature ratio, τ, and the heat capacity rate ratio, C^*, for several fixed *NTU* values. The proposed new flow arrangement delivers higher thermal effectiveness and higher heat exchanger efficiency, resulting in lesser entropy generation over a wide range of C^* and *NTU* values. These metrics are quantified with respect to the arrangement widely used in refrigeration industry due to its high effectiveness, namely, the standard two-pass counter-cross-flow heat exchanger. The new flow arrangement seems to be a promising avenue in situations where cross-flow heat exchangers for single-phase fluid have to be used in refrigeration units.

1. Introduction

The main aim of the present paper is to thermally characterize a new flow arrangement of the family of cross-flow heat exchangers, which may find application in the refrigeration and automobile industries. This new configuration was recently proposed by Cabezas-Gómez et al. [1] relying on the general guidelines developed in Guo et al. [2]. Those authors examined the implications and applicability of the so-called "uniformity principle" of the temperature difference field (*TDF*) in connection to the heat exchangers effectiveness. They tested several flow arrangements and demonstrated both theoretically and experimentally that a heat exchanger with a more uniform *TDF* distribution holds higher effectiveness values. According to [2], the thermal effectiveness of a cross-flow heat exchanger could be upgraded by tuning a more uniform *TDF*. This aspect can be handled in two different ways, either redistributing the heat transfer area or rearranging the connections between tubes. In the present study, certain arguments related to the latter procedure have been exploited as the basis for the development of a tube-side flow distribution. At the end, the thermal performance of the heat exchanger could show vital signs of enhancement. It must be recognized that the idea behind the uniform *TDF* is closely related to other norms that characterize the performance of the heat exchanger resting on the 2nd Law of Thermodynamics, such as the so-called *HERN* (Heat Exchanger Reversibility Norm), developed by Sekuliç [3]. It turns out that the flow arrangement under scrutiny also brings forth higher heat exchanger efficiency, a concept recently introduced by Fakheri [4,5].

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Nomenclature

A	outer total heat transfer area, m^2
$AMTD$	arithmetic mean temperature difference, K
C	heat capacity rate, W/K
C^*	heat capacity rate ratio, C_{min}/C_{max}, dimensionless
Fa	fin analogy number, dimensionless
$HERN$	heat exchanger reversibility norm, dimensionless
NTU	number of transfer units, UA/C_{min}, dimensionless
q	heat transfer rate, W
T	temperature, K
$\overline{T}$	hot fluid average temperature, K
$\bar{t}$	cold fluid average temperature, K
TDF	temperature difference field, K
U	overall heat transfer coefficient, $W/(m^2K)$
Y_s	quality of energy transformation, dimensionless

Greek symbols

ϵ	conventional heat exchanger effectiveness, q/q_{max}, dimensionless
Γ	dimensionless parameter in Eq. (1)
η	heat exchanger efficiency, dimensionless
τ	inlet temperatures ratio, dimensionless

Subscripts

c	cold fluid side of heat exchanger
h	hot fluid side of heat exchanger
i	inlet condition
max	maximum
min	minimum
o	outlet condition
opt	optimum

Superscripts

e	element

The flow arrangement under study consists of two fluid circuits displayed in two tube rows, each with two tube lines (refer to the forthcoming Fig. 1), which is similar to that proposed in [2]. Those authors considered an arrangement with four tube rows, four fluid circuits and two passes, leading to an in-tube fluid with some degree of un-mixing. In the present arrangement the in-tube fluid is mixed at each pass in each circuit (see Fig. 1a of Section 3). This flow arrangement is new to the best of the authors' knowledge. In order to evaluate the new heat exchanger performance, a comparison has been made against the standard two-pass counter-cross-flow arrangement, which is widely used in the refrigeration industry due to its high effectiveness. In the present study, the comparison has been performed regarding these trio of thermal parameters: (1) the thermal effectiveness, ϵ; (2) the heat exchanger efficiency, η; and (3) *HERN*. The first two parameters, ϵ and η, were compared for several values of the heat capacity rate ratio, C^*, and the number of transfer units, *NTU*. The direct effect upon the *HERN* values has been determined for such parameters as the inlet temperature ratio, τ, and the heat capacity rate ratio C^*, for several *NTU* values, based on the assumption of negligible fluid friction and mixing contributions. In reference to cross-flow heat exchangers with both fluids unmixed, the influence of mixing occurs at the entrance and outlet headers of both fluids, a process that might downgrade the magnitude of the reversibility norm due to inherent mixing irreversibilities [6]. It is worth emphasizing that in the present analysis, issues pertinent to temperature difference have been contemplated in details in Section 3.1; while issues regarding the mixing effects are discussed heuristically in Section 3.2, respectively.

The thermal effectiveness, the heat exchanger efficiency and the entropy generation in both cross-flow heat exchanger configurations have been computed by applying a recently developed computational procedure described in Navarro and Cabezas-Gómez [7] and Cabezas-Gómez et al. [8]. In the latter reference, a comprehensive study of the cross-flow heat exchanger flow arrangement proposed by Guo et al. [2], (refer to Fig. 9(d) of this reference) was performed. In addition, Cabezas-Gómez et al. [8] have shown that the thermal effectiveness of the complex flow arrangement is higher than that of a four-pass counter-cross-flow heat exchanger with higher values of C^* and *NTU*. However, this result is not suitable to the lower range of C^* and *NTU*, because in this range an opposite trend has been found. A graphical

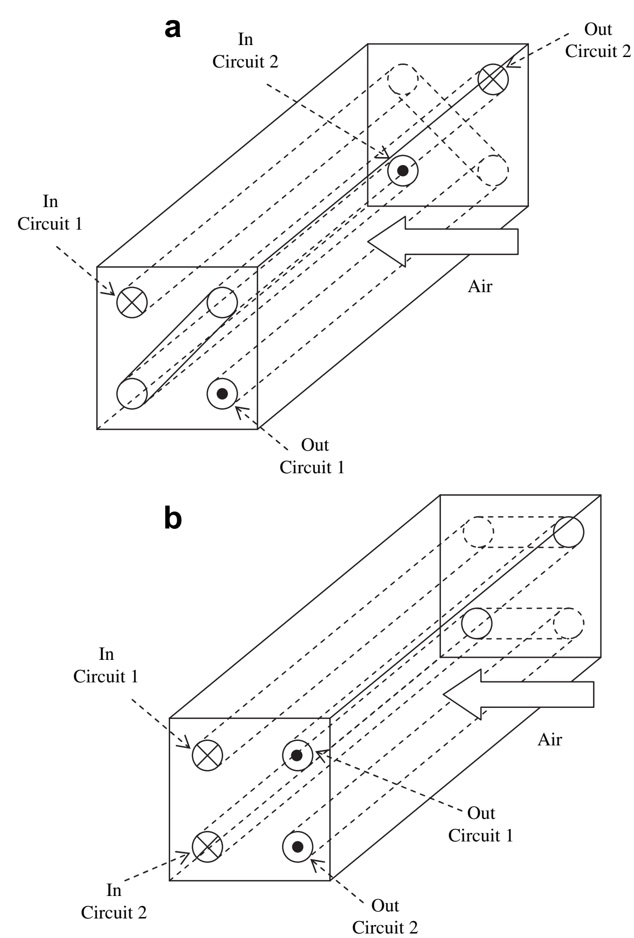

Fig. 1. Schematic representation of heat exchangers: (a) new heat exchanger configuration, (b) standard two-pass counter-cross-flow. (the × and ● symbols indicate fluid streams into or out of the paper sheet, respectively).

representation of performance data for the new flow arrangement due to Guo et al. [2] was also provided.

2. Theory

The computation of the important parameters, thermal effectiveness, heat exchanger efficiency and entropy generation, employs the approach delineated in [7,8]. The procedure consists of a sequence of three basic steps: (i) division of the heat exchanger in a finite number of three-dimensional control volumes or elements, each element being itself a mixed-unmixed cross-flow heat exchanger; (ii) solution of the set of governing equations for each element; and (iii) step-by-step solution of the governing equations of subsequent elements along each circuit of the in-tube fluid.

In the above referenced computational procedure a typical element consists of a cross-flow heat exchanger with the in-tube fluid mixed and the external fluid unmixed (In Fig. 1 the external fins are not shown for figure simplification). Generally, the in-tube fluid is considered as the hot fluid for analysis purposes. The number of elements along the in-tube fluid circuit must be sufficiently large so that the element size remains small. This in turn, causes that the external fluid flow rate of the element will be very small when compared to that of the in-tube fluid. As a consequence, the thermal capacity rate of the external fluid, C, defined as the specific heat times the mass flow rate, will be significantly smaller than that of the in-tube fluid. This statement is equivalent to writing $C_c^e \ll C_h^e$, where the subscripts "h" and "c" stand for hot and cold, and the superscript "e" being the element in question. Since the elements are chosen to be small, the mixed-unmixed cross-flow heat exchangers fulfilling the above condition for the interrelation between C_c^e and C_h^e, along with their thermal effectiveness, Γ^e, can be approximated by the following expression:

$$\Gamma^e = \frac{\Delta T_c^e}{\left(T_h^e - T_{c,i}^e\right)} = 1 - \exp^{-\frac{(UA)^e}{C_c^e}} \tag{1}$$

where T_h^e is the average in-tube fluid temperature in the element given by:

$$T_h^e = 0.5\left(T_{h,i}^e + T_{h,o}^e\right) \tag{2}$$

In equation (1) U represents the overall heat transfer coefficient, A denotes the outer total heat transfer area, and ΔT_c^e stands for the average cold fluid temperature difference in the element.

The outlet temperatures of the in-tube and external fluids are obtained by combining equations (1) and (2) with those equations related to the energy balances in each fluid. This operation results in the following set of equations:

$$T_{c,o}^e = \frac{B + 2(1-\Gamma^e)}{2+B} T_{c,i}^e + \frac{2\Gamma^e}{2+B} T_{h,i}^e \tag{3}$$

$$T_{h,o}^e = \frac{2-B}{2+B} T_{h,i}^e + \frac{2B}{2+B} T_{c,i}^e \tag{4}$$

where $B = C_c^e \Gamma^e / C_h^e$. The solution of the governing equations for the set of elements that make up for the whole heat exchanger can be obtained by implementing a marching procedure, following the in-tube fluid circuitry. The final step in the calculation procedure is the determination of the thermal effectiveness of the heat exchanger from the evaluated exit temperatures of both fluids. Details about the computational procedure and the derivation of the set of equations can be found in [7,8].

According to Sekuliç [3], the entropy generation number defined by Bejan [9], scaled by the maximum entropy generation number has the physical meaning of an irreversibility norm. The resulting dimensionless entropy generation, $1 - Y_s$, is given by the following expression:

$$1 - Y_s = \frac{C^*\ln\left[1 - \epsilon(1-\tau^{-1})\right] + \ln\left[1 - C^*\epsilon(1-\tau)\right]}{C^*\ln\frac{C^*\tau + 1}{\left(C^*+1\right)\tau} + \ln\frac{C^*\tau+1}{C^*+1}} \tag{5}$$

This equation was derived taking into account the irreversibility caused by a finite temperature difference only. According to equation (5), the quality of the heat transfer process in a heat exchanger depends on three parameters (see [3]): (1) the inlet temperatures ratio, (2) the heat capacity rate ratio, and (3) the heat exchanger effectiveness. Considering that the heat exchanger effectiveness is a function of the heat capacity rate ratio, C^*, the number of transfer units, *NTU*, and the fluids flow arrangement, it turns out that the quality of energy transformation, named *HERN*, can be written as the following general functional form ([3]):

$$HERN = Y_s \sim f(\tau, C^*, NTU, \text{flow arrangement}) \tag{6}$$

Equation (5) is used in the present paper to compute the dimensionless entropy generation from the computational procedure results.

Recently Fakheri [4,5], following the footsteps of the fin efficiency concept, introduced the heat exchanger efficiency, η, defined as the ratio of the actual heat transfer rate, q, to the optimum rate of heat transfer, q_{opt}, that is,

$$\eta = \frac{q}{q_{opt}} = \frac{q}{UA(\overline{T} - \overline{t})} \tag{7}$$

The optimum heat transfer rate, q_{opt}, amounts to the product of *UA* times the arithmetic mean temperature difference, *AMTD*, which is the difference between the average temperatures of the hot and cold fluids, respectively, $\overline{T}$ and $\overline{t}$. Fakheri [4] introduced the heat exchanger efficiency based on the fact that the actual heat transfer rate in a heat exchanger is always lower than the optimum heat transfer rate which takes place only in an ideal balanced counter-flow heat exchanger. Thus, the heat exchanger efficiency can be computed from Equation (7) for any cross-flow heat exchanger flow arrangement. In the present study the heat exchanger efficiency has been determined from the computed numerical results. The heat exchanger efficiency has been introduced in the present study as an evaluation parameter considering that it is a new concept that provides an effective way for the analysis and the design of heat exchangers and heat exchangers networks [4].

3. Results and discussion

The new flow arrangement is illustrated in Fig. 1(a). The standard two-pass counter-cross-flow arrangement to be used as the comparison counterpart appears in Fig. 1(b). In both arrangements, air has been considered as the external fluid. While not shown in Fig. 1, the presence of fins on the external side is assumed in such a way to consider the external fluid as unmixed. Thus, the procedure introduced in Section 2 can be applied in the present analysis. As previously noted, the new arrangement contains two fluid circuits that extend over two tube rows, each with two tube lines, similar to the four tube rows examined by Guo et al. [2]. The second line of tubes in the two-pass counter-cross-flow arrangement in Fig. 1(b), have the same thermal "history" of the first line tubes, thus not affecting the thermal efficiency of the heat exchanger.

3.1. Thermal effectiveness, efficiency and entropy generation

Shown in Fig. 2 are representative plots of the thermal effectiveness versus *NTU* for the two configurations in Fig. 1 associated with several values of C^*. These plots cover data in the following ranges of C^* and *NTU*: $0 \le C^* \le 1.0$; and $0 < NTU \le 20$. A quick glance over the plots in Fig. 2 allows one to conclude that the proposed flow arrangement clearly delivers higher thermal effectiveness over the whole range of *NTU* and C^* tested. As expected, it can also be noted that the thermal effectiveness difference between both arrangements increases with *NTU*. Considering that the thermal effectiveness of standard two-pass counter-cross-flow arrangement is higher than that of typical two-pass cross-flow heat exchangers, it is apparent that the proposed arrangement upgrades the thermal performance of the two rows cross-flow heat exchangers.

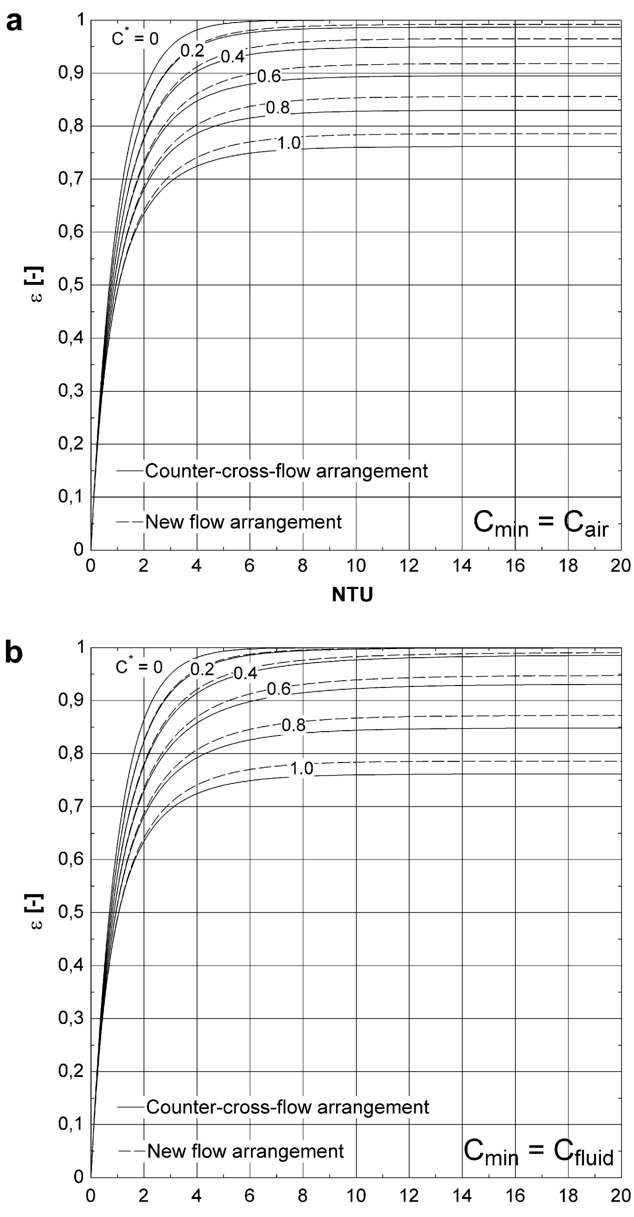

Fig. 2. Effectiveness-*NTU* plots for the two configurations in Fig. 1. Two cases: (a) $C_{\min} = C_{\text{air}}$ and (b) $C_{\min} = C_{\text{fluid}}$.

Entropy generation from Equation (5) has also been obtained for the same ranges of C^* and *NTU* as that for the thermal effectiveness, and inlet temperatures ratio, τ, equal to 0.1, 0.5, and 0.9. The dimensionless entropy generation for both flow arrangements is overlaid in the plots of Fig. 3 for a specific inlet temperatures ratio of 0.5. It can be noted that entropy generation reaches a maximum at *NTU* values in the range between 1 and 2. Beyond the maximum entropy generation corresponding to higher values of *NTU*, the present flow arrangement clearly exhibits better performance, since the heat exchanger reversibility norm values are lower than those of its counterpart (lower entropy generation). In the lower *NTU* range, an opposite trend has been observed, that is, the reversibility norm of the cross-counter-flow arrangement is better than the proposed arrangement, though differences are of lesser order than those corresponding to the *NTU* higher range. Although not shown in this paper, results for other values of inlet temperatures ratio are similar to the ones displayed in Fig. 3. It is clearly observable in the plots of Fig. 3 that the trends are similar to those encountered in the study by Sekuliç [3]. At this point, it must be stressed that in a limited range of *NTU* values, the reversibility norm

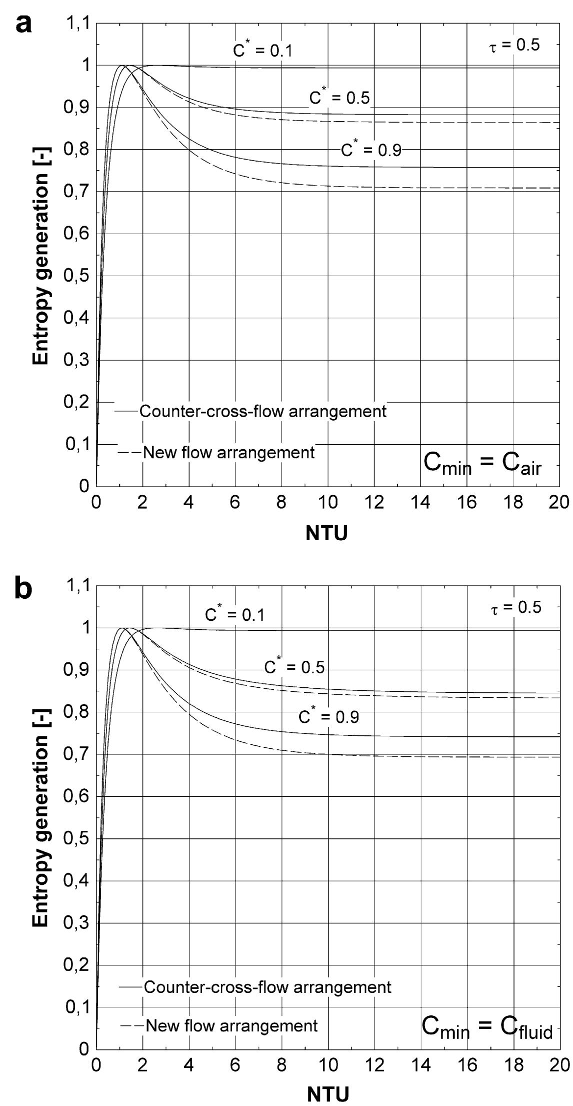

Fig. 3. Values of the dimensionless entropy generation as a function of *NTU* for the two configurations in Fig. 1 considering $C^* = 0.1$, 0.5 and 0.9. (a) $C_{\min} = C_{\text{air}}$ and (b) $C_{\min} = C_{\text{fluid}}$.

of the proposed arrangement is lower than that of the cross-counter-flow arrangement. Needless to say that satisfactory consistency is evident. This in turn corresponds to a higher thermal effectiveness over the whole range of *NTU* values.

Heat exchanger efficiency data are overlaid in Figs. 4 and 5 for a couple of values of C^*, 0.5 and 0.9, over a wide range of *NTU* values: $0 \leq NTU \leq 20$. The plots in both figures lead to the conclusion that the proposed flow arrangement is more efficient over the whole range of *NTU* values. It is apparent that for the pair of C^* values, the differences in heat exchanger efficiency for the two arrangements increase with *NTU* up to a maximum (near $NTU = 12$). Thereafter, the differences decrease for higher *NTU* values ($NTU > 12$). As noted in Figs. 4 and 5, the heat exchanger efficiency always decreases with *NTU*. Fakheri [4] argues that this trend is closely related to that of a fin, in which case a non-dimensional fin number, *Fa*, is defined. In fact, in the limit, the efficiency of a hypothetical infinitely long fin is equal to zero, even though it still transfers a finite amount of heat. Similarly, according to Fakheri [4], the efficiency of an infinitely large heat exchanger is equal to zero, even though it transfers a finite amount of heat.

According to Fakheri [10], the efficiency, η, can also be combined with the thermal effectiveness, ϵ, as follows:

$$\eta = \frac{1}{NTU} \frac{1}{\dfrac{1}{\epsilon} - \dfrac{(1 + C^*)}{2}} \tag{8}$$

It can be noted that for the same values of *NTU* and C^*, high values of the thermal effectiveness are related to higher values of heat exchanger efficiency. This tendency is relevant because it justifies the observed behavior in Figs. 4 and 5. Moreover, it proves that the proposed arrangement has a great potential for upgrading the thermal performance of two-row cross-flow heat exchangers.

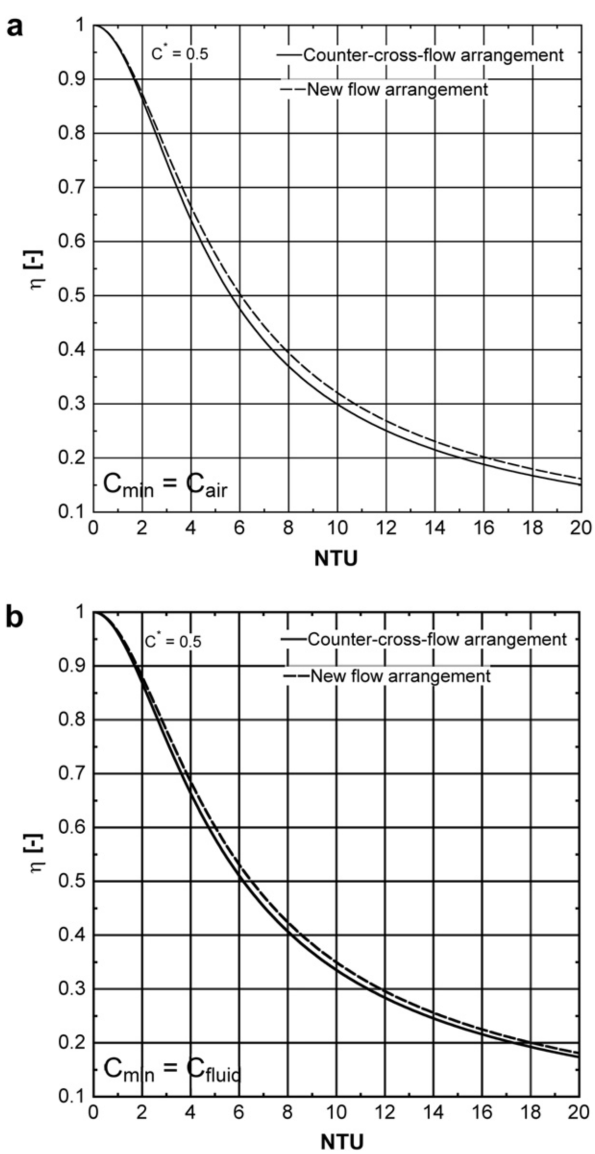

Fig. 4. Heat exchanger efficiency as a function of *NTU* for the two configurations in Fig. 1 for a given $C^* = 0.5$. Two cases: (a) $C_{min} = C_{air}$ and (b) $C_{min} = C_{fluid}$.

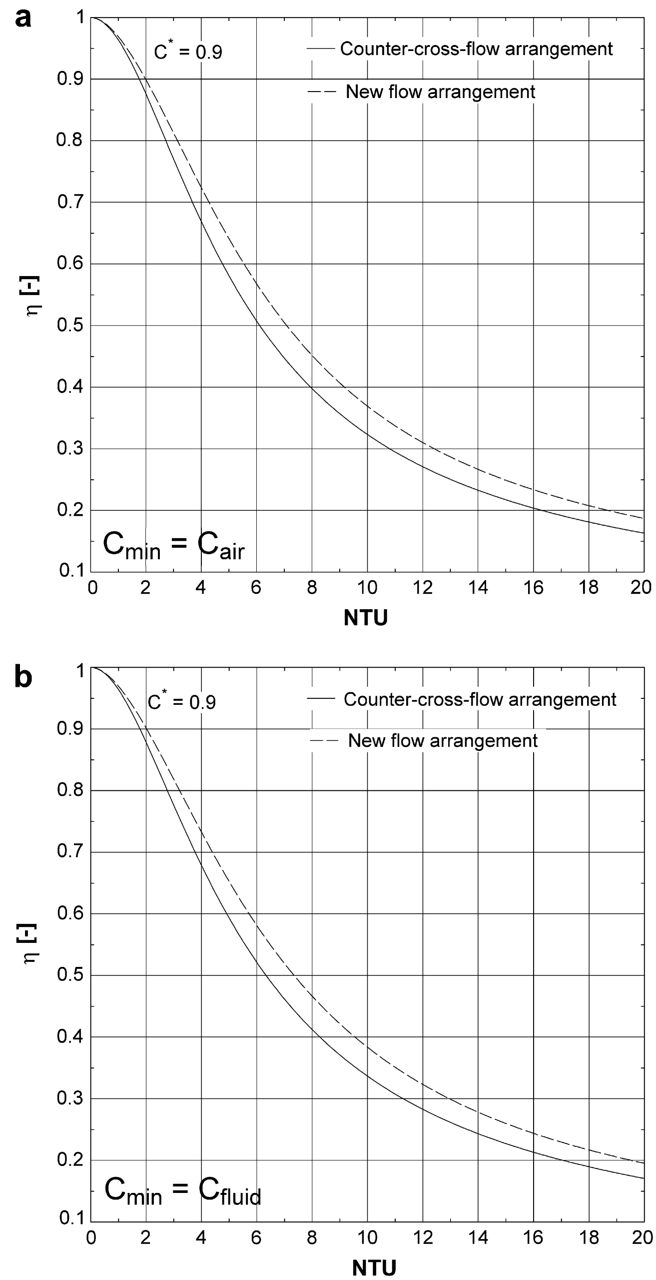

Fig. 5. Heat exchanger efficiency as a function of *NTU* for the two configurations in Fig. 1 for a given $C^* = 0.9$. Two cases: (a) $C_{min} = C_{air}$ and (b) $C_{min} = C_{fluid}$.

The thermal behavior of the preceding paragraph could be explained through an in-depth analysis of the proposed flow arrangement configuration. In both flow arrangements of Fig. 1, the in-tube fluid flows in overall cross-counter-flow direction in relation to the external fluid when the in-tube fluid passes from the first to the second row, respectively. Interestingly, the newly proposed flow arrangement introduces a new aspect. In each row both in-tube fluid circuits flow in counter-current with respect to each other, leading to a lesser locally mean element temperature differences along the in-tube fluid circuits, as well as in the whole heat exchanger. This feature of the proposed configuration can be used as a guideline in the development of other configurations seeking to optimize the thermal performance of heat exchangers by considering only the influence of flow arrangement on the heat transfer process.

3.2. Heuristic approach of the mixing effect

In order to analyze flow arrangement mixing effects on the heat exchanger effectiveness, a qualitative study based on a heuristic approach recently proposed by Shah and Sekuliç [6] has been performed. It must be stressed that mixing effects along the heat exchanger do not contribute to differences in thermal performance of the flow configurations of Fig. 1 since both streams are non-mixed (each circuit do not mix with each other, and the external fluid is unmixed). Mixing effects of the exit streams of both fluids do affect the performance due to the non-uniformity of the temperature distribution.

Considering the flow arrangements displayed in Fig. 1, the mixing of the in-tube fluid in each circuit is identical in both arrangements. The same occurs as far as the mixing of the external fluid in individual rows is concerned. However, local temperature differences in both configurations are different due to the overall flow arrangement, what results in the aforementioned effectiveness differences between them.

The arrangements of Fig. 1 are both cross-counter-flow. The difference between the two is that in the proposed arrangement, Fig. 1(a), there is a kind of compensation in order to make the temperature difference between both fluids lesser and more uniformly distributed than in the classical cross-counter-flow of Fig. 1(b); and, as a result, best thermally efficient. The following is a plausible explanation for this statement based on the external fluid path along the heat exchanger. The upper right region of the external fluid (air) initially meets the exit region of one of the in-tube circuit what implies in a low temperature difference between both streams. Then the external fluid meets the end of the first in-tube row. The lower right region of the external fluid follows an opposite trend. Initially meets a higher temperature difference than the upper region and then a higher temperature difference what results in an exit temperature close to the upper region air. The left region of the external fluid follows similar trend, but with inverted heat transfer characteristics between the upper and lower region with respect to the right area. One can thus conclude that the exit temperature distribution of the external fluid is rather uniform lessening mixture effects. In addition, the temperature difference between the fluids in the proposed arrangement is lesser and more uniform over the heat transfer area than the classical cross-counter-flow arrangement. In fact, after the first row in the configuration of Fig. 1(b), the temperature of the external fluid (colder one) increases from left to right, since the rate of heat transfer in the left region is lower than in the right due to the lower temperature difference between both fluids. In its passage through the second row, the fluid of the external left region meets the in-tube fluid entrance, and, as a result, the rate of heat transfer is higher (temperature difference is higher) than in the right region. Finally, the exit stream of the external fluid must present a rather uniform temperature distribution, similar to the proposed arrangement. The exit temperature of the in-tube circuits is very close to each other in both arrangements not being the cause of a major mixing effect.

4. Conclusions

A new flow arrangement for a cross-flow heat exchanger has been conceived. The rationale behind its formulation is to seek a better thermal effectiveness than the two-row cross-counter-flow arrangement over the whole *NTU* range. Also, to seek a better reversibility norm for the range of *NTU* values higher than 2. As a direct result, the proposed flow arrangement can be considered as a potential substitute for typical arrangements currently used in single-phase cross-flow heat exchangers in various refrigeration applications. Further investigation including pressure drop and a more detailed study of mixing effects influence on the above three thermal parameters analyzed is needed to completely confirm the thermo-hydraulic efficacy of the proposed flow distribution.

The present results confirm the adequacy of the "uniformity principle" recommended by Guo et al. [2], as well as the potential use of the efficiency definition developed by Fakheri [4] for the analysis and the design of heat exchangers. The utility of the computational approach is highlighted, allowing the simulation of flow arrangements when the in-tube fluid circuits flow through the heat exchanger in opposite directions in each in-tube pass or heat exchanger row.

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