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An alternate algorithm for the analysis of multistream plate fin heat exchangers

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Abstract

A new algorithm has been developed for the analysis of multistream heat exchangers. The numerical technique involves partitioning of the exchanger in both axial and normal directions. Conservation equations written for each segment are solved using an iterative procedure. In the axial direction, the exchanger is successively partitioned to 2^k segments, the final value of k being determined by the point where further partitioning has only marginal effect. In the normal direction, the exchanger is divided into a stack of overlapping two-stream exchangers interacting through their common streams. The algorithm has been tested against published results and good agreement has been observed.

Keywords: Heat exchanger; Plate fin; Multistream; Successive partitioning; Area splitting

1. Introduction

Plate fin heat exchangers are basically multistream devices capable of handling two or more fluid streams in a single unit. They offer high degrees of compactness and effectiveness. These features lead to small terminal temperature difference with a small overall size of the exchanger and flexibility in stream arrangement. Depending on the process requirement, intermediate entry and exit of the streams can also be provided.

The design of a two-stream heat exchanger is rather straight forward. Analytical formulae are given in all text and reference books on heat exchangers [1–9] for design of two-stream units. Non-uniform heat transfer coefficient, property variations and the presence of secondary effects like longitudinal conduction and axial dispersion makes the analysis somewhat complex; but can be easily tackled using numerical approach [10,11]. On the other hand the design and simulation of multistream plate fin heat exchangers are markedly different from those of two-fluid exchangers. Features like bypass heat transfer [12] or cross-over in temperature [13], common in multistream heat exchangers, have no equivalent in two-stream units.

In the simplest form a multi stream handles three different streams of fluids. Sorlie [14] developed a design theory for three-fluid heat exchangers of the concentric-tube and plate fin types, in which the intermediate and cold streams were thermally insulated. He derived closed form solutions for the temperatures of all the streams by solving a set of three first order linear ordinary differential equations and defined an expression for the overall effectiveness. Some of the theoretical results were compared with experiments and excellent agreement was obtained. Aulds and Barron [15] extended the work of Sorlie by analysing the case in which all three streams were in thermal communication, which is relevant to many three-fluid heat exchangers used in cryogenic systems.

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Nomenclature

A	heat transfer area (m^2)
A_w	primary (wall) heat transfer area (m^2)
A_f	secondary (fin) surface area (m^2)
C_i	heat capacity rate of fluid i (W/K)
C_p	specific heat (J/kg K)
h	local heat transfer coefficient (W/m^2 K)
k, k'	dimensionless discrete location
L	heat exchanger length (m)
m	number of segments i.e. 2^j
$\dot{m}$	mass flow rate (kg/s)
n	total number of fluid streams
Q	amount of heat transfer (W)
T	separating wall temperature (K)
t	fluid temperature (K)
U	overall heat transfer coefficient (W/m^2 K)
x, y, z	length coordinates

Greek symbols

α	fraction of area defined in Eq. (21)
Φ_P	additional temperature term for co-current exchanger defined in Eq. (25) (K)
Φ_C	additional temperature term for counter current exchanger defined in Eq. (31) (K)
η_f	fin temperature effectiveness
Ψ_P	dimensionless parameter for co-current exchanger defined in Eq. (26)
Ψ_C	dimensionless parameter for counter current exchanger defined in Eq. (32)

Subscripts and superscripts

f	fin
i	index of fluid streams
j	stage of partitioning
w	separating wall

Sekulic and Shah [16] have presented an extensive review of literature on the three-fluid heat exchangers. They have stated that the existing analytical solutions were valid only for a particular design and/or for a particular flow arrangement, and are not generally suited for use in general purpose computer codes. Lack of a unified design methodology motivated Sekulic and Shah [16] to derive a comprehensive theory for design of parallel flow three-stream heat exchangers. They outlined a step-by-step procedure for the solution of rating and sizing problems. This analysis, however, indicates that the extension of usual effectiveness – NTU concept to three-stream units increases the complexity of the analysis to a significant degree.

Pioneering work on multi-channel, parallel flow heat exchangers have been done by Kao [17] and Wolf [18]. They showed that in the absence of the effect of axial conduction through the separating wall, the basic equations describing the process of heat transfer in a multi-channel heat exchanger are a set of linear differential equations involving the temperatures of the fluids and the separating walls. A similar approach has been adopted by Zaleski [19] to analyse multi-channel heat exchangers with unconnected channels, particularly lamella-type, plate-type, and helix-type units.

In general, the assumption of constant physical properties is a poor one, particularly in case of cryogenic heat exchangers where the change in temperature between the two ends is quite high. Chato et al. [20] have suggested dividing the heat exchanger into a large number of smaller sections over which physical properties remain approximately constant. They could express the temperatures at one end of an exchanger section in terms of those of the other end through the use of a "temperature transfer matrix". An "overall transfer matrix" for the whole exchanger was formed by multiplying all these individual sectional matrices and was used to predict the performance of the entire exchanger.

Bentwich [21] considered an idealised steady-state model of a counter-current heat exchanger in the absence of axial conduction and property variation assuming a parabolic temperature profiles between consecutive walls. Finite difference solution of the resulting equation set was obtained in terms of inlet condition and heat flux across the separating wall. The author proposed to define efficiency of the exchanger by a method of normalising the heat flux across every separating wall.

Integration of the governing differential equations of a multistream heat exchanger often leads to divergence of solution [12], particularly when two or more streams flow in each direction. This divergence is not a mathematical or numerical artifice, but is intrinsic to the solution process [12]. To overcome this difficulty, Haseler [12] suggested a novel solution methodology termed as constant wall temperature assumption. Based on this the temperatures of all the separating walls are assumed to be equal at any cross section normal to the flow direction. She derived a "bypass fin efficiency" which takes into account the transverse conduction through the fins. This method suffers from the serious drawback that it precludes the effect of layer-stacking pattern and transverse conduction because of the pre-assumed "constant wall temperature" approximation. Layer-stacking pattern is a crucial aspect of the design exercise for maximising heat transfer between several cold and hot streams. Moreover, the method, though mathematically stable, can lead to significant under-design of the heat exchanger for some streams, because it artificially introduces a heat flow between the plates in the transverse direction.

Prasad and Gurukul [22,23], in their formulation of the differential method for design of plate fin heat exchangers, applied the same simplifying idealization. Existence of zero temperature gradient in the fins near the channel centreline proposed by Chato et al. [20] was also used by Prasad [24] to study the layer-stacking pattern in the multistream heat exchanger.

Paffenbarger [25] worked out a generalized method for the analysis of counterflow plate fin heat exchanger using a commercial package of finite element. The effects of axial conduction and variable physical properties were considered in the analysis. However, heat exchange with the environment was ignored.

Prasad [24] analysed the mechanism of heat transfer through the fins connecting two walls at different temperatures. His model considered conduction through the fins along with convective heat transfer to the fluid to determine the temperature profile over the fin and calculate the net heat transfer to the separating walls. The effect of transverse conduction was thus taken care of by this formalism. In a later improvement Prasad [26] showed that the assumption of zero temperature gradient over the fins near the channel centreline made [20,24] could result in considerable error. He suggested a method of correction to this based on the mechanism of local heat transfer and developed a computational algorithm [27] using this new formalism.

Unlike Haseler [12], Prasad and his co-workers [24,26,27] have employed the modified shooting method to solve the governing equations. Their method could suppress the tendency of divergence as it corrected the guess temperature at each segment. Their method is, however, sensitive to the initial guesses and under certain circumstances wrong initial guess of the temperature profile can lead to divergence.

Luo et al. [28] have developed an analytical model of a multistream exchanger with constant physical properties. In a separate paper [29], the authors have proposed a more generalised analytical solution for predicting the thermal performance of multistream heat exchangers and their networks. This model is also applicable to other types of one-dimensional heat exchangers such as shell and tube and plate heat exchangers.

"Pinch Technology" is a method usually adopted for the analysis of heat exchanger networks. Polley and Picon-Nunez [30] and Picon-Nunez et al. [31,32] extended this technique for multistream plate fin heat exchangers based on the use of temperature vs. enthalpy diagrams or "composite curves". Their design objective, aimed at maximum utilization of allowable pressure drop, ensures homogeneous heat load in all the channels and produces equal number of hot and cold channels.

Wang et al. [33] presented a new methodology for design of multistream plate fin heat exchangers through optimisation of heat exchanger networks. Pinguad et al. [34] and Luo et al. [35] have also carried out steady-state and dynamic simulation of plate fin heat exchangers.

In multistream, multi-channel plate fin heat exchangers, the estimation of fin heat transfer is rather a complex task. In some cases the two layers on either side of a given stream may both be at a higher or a lower temperature than the middle one. In such cases, a maximum or a minimum in the temperature profile exists in the central stream. One can then assume that an *adiabatic plane* exists somewhere within central layer, through the points of maximum or minimum temperature.

In other cases a temperature maximum or minimum does not exist inside the central stream and an adiabatic plane cannot be constructed. A hypothetical adiabatic plane may be assumed to exist outside the central stream and its location can be computed by analytically extending the temperature profile [26,27]. Different authors have accounted for this phenomenon in different ways. While deriving the governing heat transfer equations, the temperature profile over the fin is taken into account either through a "bypass fin efficiency" [12,25], or by directly evaluating conduction heat transfer through the fins [26,28].

Though some work have been done to analyse the performance of the multi stream plate fin heat exchangers, the techniques have not been standardized as in the case of two-fluid heat exchanger. There is a scope of developing new algorithms, which can take care of stacking pattern without leading to a divergence of the solution. Based on the above observations, an alternative methodology for analysis has been developed in the present work. Two key concepts have been used in the present work. The multistream heat exchanger has been conceived as a combination of a number of overlapping two-stream heat exchangers. This needs apportioning the heat exchanger area between different streams. This has been achieved by "*Area Splitting Method*". Next, the heat exchanger has been progressively subdivided in the axial direction by "*Successive Partitioning Method*" to improve the accuracy of prediction. It may be noted that the concept of "*area splitting*" is new in case of multistream heat exchangers. Partitioning the heat exchanger into small control volumes in the axial direction is not new, but the method of successive partitioning adopted in the present algorithm is different from the earlier works. The present algorithm shows good agreement with published theoretical results and experimental data.

2. Basic design approach

The basic features of a multistream heat exchanger can be understood taking an example of a three-fluid heat

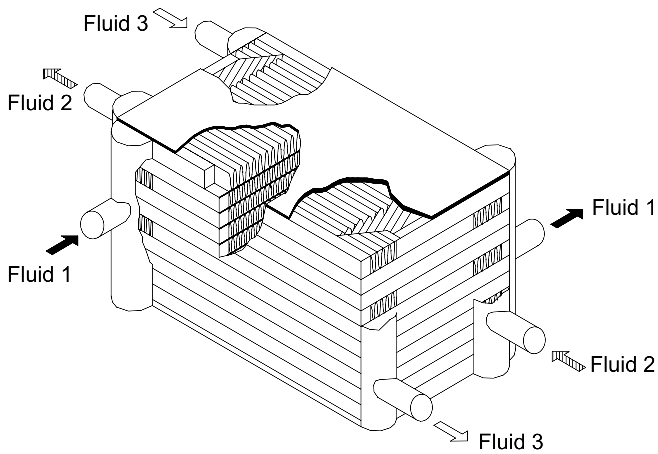

Fig. 1. Multistream heat exchanger with three-fluid streams.

exchanger as shown in Fig. 1. The first and the third streams of this exchanger enter from one end, while the second one enters from the other.

The multistream plate fin heat exchanger can be imagined as a combination of several overlapping two-stream units (sub-exchangers) stacked in a pile. The interaction between the sub-exchangers takes place through their common streams and boundaries. For example, a three-stream exchanger shown in Fig. 2 can be considered as a combination of two overlapping sub-exchangers, each carrying two streams.

The co-current sub-exchanger 1 consists of streams 1 and 2, while streams 2 and 3 constitute a counter-current exchanger (sub-exchanger 2), the middle stream (stream 2) being common to both the sub-exchangers. Q_i representing the heat flow rate across the ith plate. If the cap sheets are insulated Q_0 and Q_n are equal to zero. Q_i is considered positive when the flow direction is from stream i to stream $(i+1)$. In sub-exchanger 1, the fluid in channel-1 (with flow rate $\dot{m}_1$) exchanges heat with the fluid in channel-2 (with flow rate $\dot{m}_2$) across plate-1. Fluid stream 1 also receives an additional amount of heat flow Q_0 ($=0$ for insulated cap sheet), and fluid stream-2 delivers an additional heat stream Q_2 across plate-2. The participating surfaces are the primary surfaces of plate-1 ($A_{\mathrm{w},1}$ assumed equal on both sides) and the secondary (fin) surfaces of the two channels $A_{\mathrm{f},1}$ and $A_{\mathrm{f},2}$. Similarly in the sub-exchanger 2, the fluid in channel-2 (with flow rate $\dot{m}_2$) exchanges heat with the fluid in channel-3 (with flow rate $\dot{m}_3$) across plate-2. The participating surfaces of sub-exchanger 2 are the primary surfaces of plate-2 ($A_{\mathrm{w},2}$) and the secondary surfaces $A_{\mathrm{f},2}$ and $A_{\mathrm{f},3}$.

Thus the secondary surface $A_{\mathrm{f},2}$ also participate in heat exchange simultaneously in two sub-exchangers. While considering sub-exchanger 1, we can assume that a part of the secondary heat transfer area $\alpha_2 A_{\mathrm{f},2}$ participates in the internal heat exchange between the streams 1 and 2, while a fraction $(1-\alpha_2)A_{\mathrm{f},2}$ is responsible for the external heat transfer Q_2 out of the sub-exchanger-2. It is assumed that the participating heat transfer area is proportional to the heat flow rate.

Mathematically,

$$\frac{A_{\mathrm{w},1}+\alpha_2\eta_{\mathrm{f},2}A_{\mathrm{f},2}}{A_{\mathrm{w},2}+(1-\alpha_2)\eta_{\mathrm{f},2}A_{\mathrm{f},2}} = -\left(\frac{Q_1}{Q_2}\right) \tag{1}$$

In other words,

$$\alpha_2 = \left(\frac{Q_1(A_{\mathrm{w},2}+\eta_{\mathrm{f},2}A_{\mathrm{f},2})+Q_2A_{\mathrm{w},1}}{(Q_1-Q_2)\eta_{\mathrm{f},2}A_{\mathrm{f},2}}\right) \tag{2}$$

In a rectangular geometry all primary surfaces have the same configuration and thus the same area A_{w}. Setting $A_{\mathrm{w},1}=A_{\mathrm{w},2}=A_{\mathrm{w}}$, we can rewrite Eq. (2) as

$$\alpha_2 = \left(\frac{Q_1(A_{\mathrm{w}}+\eta_{\mathrm{f},2}A_{\mathrm{f},2})+Q_2A_{\mathrm{w}}}{(Q_1-Q_2)\eta_{\mathrm{f},2}A_{\mathrm{f},2}}\right) \tag{3}$$

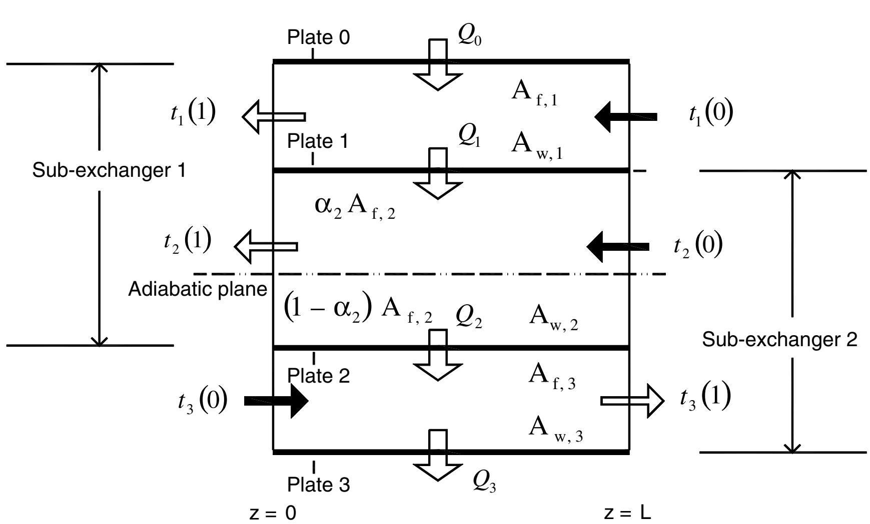

Fig. 2. Splitting of a three-stream heat exchanger into two sub-exchangers.

If stream-2 has a real adiabatic plane, α_2 will lie between 0 and 1. Otherwise, it will be a number less than 0 or greater than 1.

It may be noted that the assumption on which Eq. (1) is based is an intuitive one. Out of several other assumptions tried in this study, this one works the best. One should use caution while using this criterion as it may lead to a very high value of α_2 in some cases. However, as this assumption is made only to initiate the iteration process it does not bias the final solution.

In the schematic of the three-stream heat exchanger shown in Fig. 2, the adiabatic plane cuts across channel 2, with a fraction of secondary area α_2 above it (closer to Channel 1) and $(1-\alpha_2)$ below, close to Channel 3. The streams are identified by subscripts 1, 2 and 3. The symbol t (lower case) stands for fluid temperatures while T (upper case) stands for the plate temperature. The digits (0) and (1) within parentheses stand for the *inlet* and *exit* conditions respectively. For streams 1 and 2, the inlet is at $z = L$, while for stream 3 the inlet is at $z = 0$, the exit for all the streams being at the respective opposite ends. The plates are assumed to be infinitely conducting, so that an entire plate is considered to be at a single temperature T. This assumption may appear unrealistic for a complete exchanger, or even for a large segment of it; but for a sufficiently small segment of a heat exchanger, this assumption is logical and forms the foundation of the present design approach.

The following additional assumptions are made to keep the analysis simple and tractable:

1. Steady-state conditions exist throughout the heat exchanger.
2. There is no temperature variation across the width of the channel, i.e. normal to the plane of the paper.
3. The inlet conditions are known for all the streams *in a given section*.
4. All fluid streams are in single-phase and remain so throughout the exchanger.
5. There is no heat transfer to or from the surroundings (Q_0 and Q_n are equal to 0).
6. Axial conduction is absent both in the matrix and the fluid.

2.1. Analysis of a two-stream co-current exchanger with transverse heat addition

Fig. 3 shows the schematic of a two-stream co-current heat exchanger of length L. Two streams at temperature $t_1(0)$ and $t_2(0)$ enter at $z = L$ and leave with temperatures $t_1(1)$ and $t_2(1)$ respectively at $z = 0$.

Let us consider a small element of length dz at a distance z measured from the exit end opposite to the direction of flow. Assuming uniform distribution of heat flux over a small section of the heat exchanger length, the heat input to the element dz through the top plate becomes $\frac{Q_0}{L}\mathrm{d}z$,

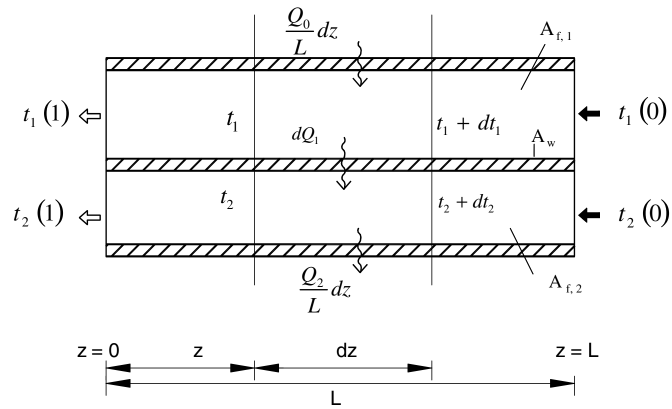

Fig. 3. A two-stream co-current heat exchanger with transverse heat addition.

and that through the bottom plate $\frac{Q_2}{L}\mathrm{d}z$. The rate of heat transfer dQ_1 through the wall separating the fluids stream 1 and 2 can be expressed in terms of Δt, the temperature difference between the two-fluid streams in the element dz.

$$\mathrm{d}Q_1 = \frac{UA\Delta t}{L}\mathrm{d}z \tag{4}$$

The overall heat transfer coefficient and area product (UA) for this two-stream exchanger is given by the equation

$$\frac{1}{UA} = \frac{1}{h_1(A_\mathrm{w} + \eta_{\mathrm{f},1}A_{\mathrm{f},1})} + \frac{1}{h_2(A_\mathrm{w} + \eta_{\mathrm{f},2}A_{\mathrm{f},2})} \tag{5}$$

where, A_w refers to the primary (wall) surface area (assumed to be the same in both sides of plate-1) and A_f the secondary surface area contributed by the fins (when we consider the sub-exchanger-1 of the three-stream exchanger shown in Fig. 2, the area $A_{\mathrm{f},2}$ is replaced by $\alpha_2 A_{\mathrm{f},2}$).

If the increment in temperature of fluid 1 within the element dz is dt_1, then

$$\mathrm{d}t_1 = \frac{\mathrm{d}Q_1}{C_1} - Q_0\frac{\mathrm{d}z}{L}\frac{1}{C_1} = \frac{\mathrm{d}Q_1}{C_1} - \frac{Q_0}{C_1}\frac{\mathrm{d}z}{L} \tag{6}$$

Similarly, for the second fluid stream the increment in temperature within the element dz can be written as

$$\mathrm{d}t_2 = \frac{Q_2}{C_2}\frac{\mathrm{d}z}{L} - \frac{\mathrm{d}Q_1}{C_2} \tag{7}$$

Combining Eqs. (6) and (7),

$$\mathrm{d}(t_1 - t_2) = \mathrm{d}t_1 - \mathrm{d}t_2 = \mathrm{d}(\Delta t) = \mathrm{d}Q_1\left(\frac{1}{C_1} + \frac{1}{C_2}\right) - \left(\frac{Q_0}{C_1} + \frac{Q_2}{C_2}\right)\frac{\mathrm{d}z}{L} \tag{8}$$

Substituting the expression for dQ_1 from Eq. (4) in Eq. (8), we get

$$\mathrm{d}(\Delta t) = \frac{UA\Delta t}{L}\left(\frac{1}{C_1} + \frac{1}{C_2}\right)\mathrm{d}z - \left(\frac{Q_0}{C_1} + \frac{Q_2}{C_2}\right)\frac{\mathrm{d}z}{L} \tag{9}$$

Integrating Eq. (9) over the limits $z = 0$ to $z = L$,

$$\int_1^0 \frac{\mathrm{d}\Delta t}{UA\Delta t\left(\frac{1}{C_1}+\frac{1}{C_2}\right)-\left(\frac{Q_0}{C_1}+\frac{Q_2}{C_2}\right)} = \frac{1}{L}\int_0^L \mathrm{d}z$$

$$\Rightarrow \int_1^0 \frac{\mathrm{d}\Delta t}{\Delta T - \Phi_{P,1}} = \frac{UA}{L}\left(\frac{1}{C_1}+\frac{1}{C_2}\right)\int_0^L \mathrm{d}z$$

$$\Rightarrow \left(\frac{t_1(1)-t_2(1)-\Phi_{P,1}}{t_1(0)-t_2(0)-\Phi_{P,1}}\right) = \mathrm{e}^{-UA\left(\frac{1}{C_1}+\frac{1}{C_2}\right)} \tag{10}$$

where

$$\Phi_{P,1} = \frac{\left(\frac{Q_0}{C_1}+\frac{Q_2}{C_2}\right)}{UA\left(\frac{1}{C_1}+\frac{1}{C_2}\right)}, \tag{11}$$

and the integration limits 1 and 0 refer to the exit and inlet states respectively. Applying energy balance over each of the fluid streams 1 and 2, we can express the outlet temperature $t_1(1)$ purely in terms of inlet temperatures

$$t_1(1) = \frac{\Phi_{P,1}+\Psi_{P,1}\{t_1(0)-t_2(0)-\Phi_{P,1}\}}{\left(1+\frac{C_1}{C_2}\right)} + \frac{C_2t_2(0)+C_1t_1(0)+Q_0-Q_2}{\left(1+\frac{C_1}{C_2}\right)C_2} \tag{12}$$

with

$$\Psi_{P,1} = \mathrm{e}^{-UA\left(\frac{1}{C_1}+\frac{1}{C_2}\right)} \tag{13}$$

Substituting the expression for $t_1(1)$ from Eq. (12) we find $t_2(1)$ as

$$t_2(1) = \frac{1}{C_2}[C_1t_1(0)+C_2t_2(0)+Q_0-Q_2] - \left(\frac{C_1}{C_2}\right)\frac{\Phi_{P,1}+\Psi_{P,1}\{t_1(0)-t_2(0)-\Phi_{P,1}\}}{\left(1+\frac{C_1}{C_2}\right)} - \left(\frac{C_1}{C_2}\right)\frac{C_2t_2(0)+C_1t_1(0)+Q_0-Q_2}{\left(1+\frac{C_1}{C_2}\right)C_2} \tag{14}$$

2.2. Analysis of two-stream counter-current exchanger with transverse heat addition

Fig. 4 shows the schematic of a two-stream *counter-current* heat exchanger with heat addition over the top and bottom faces. Deriving the governing equations for the counter-current arrangement in a manner similar to that followed for co-current exchanger, we get

$$\mathrm{d}(\Delta t) = \frac{UA\Delta t}{L}\left(\frac{1}{C_2}-\frac{1}{C_3}\right)\mathrm{d}z - \left(\frac{Q_1}{C_2}-\frac{Q_3}{C_3}\right)\frac{\mathrm{d}z}{L} \tag{15}$$

Integrating Eq. (15) over the limits $z = 0$ and $z = L$ using the method of integrating factor,

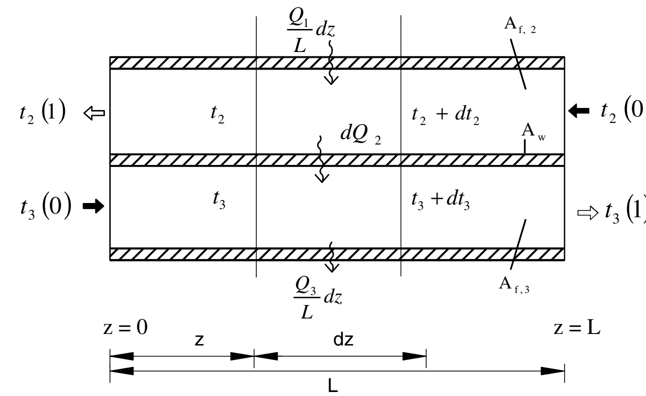

Fig. 4. Two-stream counter-current heat exchanger with transverse heat addition.

$$\left(\frac{t_2(1)-t_3(0)-\Phi_{C,1}}{t_2(0)-t_3(1)-\Phi_{C,1}}\right) = \mathrm{e}^{-UA\left(\frac{1}{C_2}-\frac{1}{C_3}\right)} \tag{16}$$

where

$$\Phi_{C,1} = \frac{\left(\frac{Q_1}{C_2}-\frac{Q_3}{C_3}\right)}{UA\left(\frac{1}{C_2}-\frac{1}{C_3}\right)} \tag{17}$$

The energy balance equation for the counter-current exchanger in Fig. 4 may be used to find out the exit temperatures of these streams:

$$t_2(1) = \frac{\{t_3(0)+\Phi_{C,1}+\Psi_{C,1}(t_2(0)-\Phi_{C,1})\}}{1-\left(\frac{C_2}{C_3}\right)\Psi_{C,1}} - \frac{\Psi_{C,1}\{C_2t_2(0)+C_3t_3(0)+Q_1-Q_3\}}{\left(1-\frac{C_2}{C_3}\Psi_{C,1}\right)C_3} \tag{18}$$

with

$$\Psi_{C,1} = \mathrm{e}^{-UA\left(\frac{1}{C_2}-\frac{1}{C_3}\right)} \tag{19}$$

Substituting $t_2(1)$ from Eq. (18) one can find $t_3(1)$ as

$$t_3(1) = \frac{1}{C_3}[C_2t_2(0)+C_3t_3(0)+Q_1-Q_3] - \left(\frac{C_2}{C_3}\right)\frac{\{t_3(0)+\Phi_{C,1}+\Psi_{C,1}(t_2(0)-\Phi_{C,1})\}}{1-\left(\frac{C_2}{C_3}\right)\Psi_{C,1}} + \left(\frac{C_2}{C_3}\right)\frac{\Psi_{C,1}\{C_2t_2(0)+C_3t_3(0)+Q_1-Q_3\}}{\left(1-\frac{C_2}{C_3}\Psi_{C,1}\right)C_3} \tag{20}$$

2.3. Analysis of the three-stream heat exchanger

With this analytical background, it is possible to adopt an iterative scheme to model the complete three-stream exchanger (Fig. 2) to determine the heat transfer rates through the separating plates and the outlet temperatures of all the fluid streams. Assigning an arbitrary non-zero value to Q_2, the exit fluid temperatures of sub-exchanger-

1 and the rate of heat transfer through the separating plate Q_1 can be determined using Eqs. (11)–(14). This value of Q_1 is used to estimate α_2 and subsequently to calculate the fluid outlet temperatures of sub-exchanger-2 using Eqs. (17)–(20). Modelling of sub-exchanger-2 yields a new value for Q_2. This modified value of Q_2 is now employed to model sub-exchanger-1 and compute Q_1, $t_1(1)$ and $t_2(1)$. The process is repeated till the values of Q_1 and Q_2 in two consecutive iterations lie within a specified tolerance.

This approach of splitting the area of an intermediate stream between two sub-exchangers can be extended to designing of multistream plate fin heat exchangers which is discussed in the next section. We call it the "*area splitting*" method.

3. Area splitting method for multistream exchanger (normal direction)

When the *area splitting* approach discussed in earlier section is extended to a multistream plate fin heat exchanger having n fluid streams, the multistream unit is considered to be a combination of $(n-1)$ overlapping two-stream sub-exchangers (Fig. 5).

While constructing the two-stream sub-exchangers, the mass flow rates in each channel are kept intact, while the heat transfer areas are divided between the two sides. For example, in the ith channel, $\alpha_i A_{\mathrm{f},i}$ is the part of the secondary area in thermal communication with $(i-1)$th plate, while the remaining fraction $(1-\alpha_i)A_{\mathrm{f},i}$ communicates with the ith plate. Following the logic adopted in Section 2 for calculation of α_2 in case of a three-stream exchanger, α_i for the multistream unit can be expressed as

$$\alpha_i = \left(\frac{Q_{i-1}(A_\mathrm{w} + \eta_{\mathrm{f},i}A_{\mathrm{f},i}) + Q_i A_\mathrm{w}}{(Q_{i-1} - Q_i)A_\mathrm{f}}\right) \tag{21}$$

The primary area A_w is considered equal for all the plates. Extending the concepts discussed earlier the relationship for ith (parallel) and $(i+1)$th (counter current) can be developed.

For the ith exchanger, using the appropriate formulae for co-current configuration, we get

$$t_{i,j}(1) = \frac{\Phi_{P,i} + \Psi_{P,i}\{t_i(0) - t_{i+1}(0) - \Phi_{P,i}\}}{\left(1 + \frac{C_i}{C_{i+1}}\right)} + \frac{C_{i+1}t_{i+1}(0) + C_i t_i(0) + Q_{i-1} - Q_{i+1}}{\left(1 + \frac{C_i}{C_{i+1}}\right)C_{i+1}} \tag{22}$$

$$t_{i+1}(1) = \frac{C_{i+1}t_{i+1}(0) + C_i t_i(0) + Q_{i-1} - Q_{i+1}}{C_{i+1}} - \left(\frac{C_i}{C_{i+1}}\right)\frac{\Phi_{P,i} + \Psi_{P,i}\{t_i(0) - t_{i+1}(0) - \Phi_{P,i}\}}{\left(1 + \frac{C_i}{C_{i+1}}\right)} - \left(\frac{C_i}{C_{i+1}}\right)\frac{C_{i+1}t_{i+1}(0) + C_i t_i(0) + Q_{i-1} - Q_{i+1}}{\left(1 + \frac{C_i}{C_{i+1}}\right)C_{i+1}} \tag{23}$$

$$Q_i = C_i\{t_{i,j}(0) - t_{i,j}(1)\} + Q_{i-1} \tag{24}$$

where

$$\Phi_{P,i} = \frac{\left(\frac{Q_{i-1}}{C_i} + \frac{Q_{i+1}}{C_{i+1}}\right)}{U\{A_\mathrm{w} + (1-\alpha_i)A_{\mathrm{f},i}\}\left(\frac{1}{C_i} + \frac{1}{C_{i+1}}\right)} \tag{25}$$

$$\Psi_{P,i} = \mathrm{e}^{-U\{A_\mathrm{w} + (1-\alpha_i)A_{\mathrm{f},i}\}\left(\frac{1}{C_i} + \frac{1}{C_{i+1}}\right)} \tag{26}$$

$$\frac{1}{U\{A_\mathrm{w} + (1-\alpha_i)A_{\mathrm{f},i}\}} = \frac{1}{h_i\{A_\mathrm{w} + (1-\alpha_i)A_{\mathrm{f},i}\}} + \frac{1}{h_{i+1}\{A_\mathrm{w} + \alpha_{i+1}A_{\mathrm{f},i+1}\}} \tag{27}$$

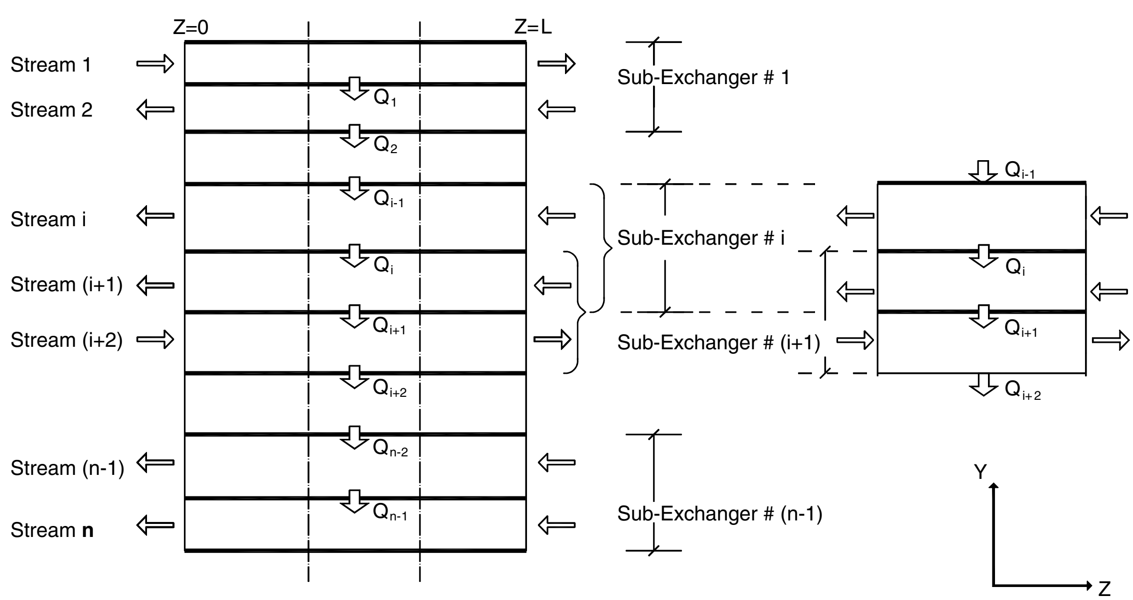

Fig. 5. A multistream plate fin heat exchanger seen as a stack of overlapping two-stream sub-exchangers.

For the $(i+1)$th exchanger, using formula for countercurrent arrangement, one gets the following relationships:

$$t_{i+1}(1) = \frac{\{t_{i+2}(0) + \Phi_{C,i} + \Psi_{C,i}(t_{i+1}(0) - \Phi_{C,i})\}}{1 - \left(\frac{C_{i+1}}{C_{i+2}}\right)\Psi_{C,i}} - \frac{\Psi_{C,i}\{C_{i+1}t_{i+1}(0) + C_{i+2}t_{i+2}(0) + Q_i - Q_{i+2}\}}{\left(1 - \frac{C_{i+1}}{C_{i+2}}\Psi_{C,i}\right)C_{i+2}} \tag{28}$$

$$t_{i+2,}(1) = \frac{C_{i+1}t_{i+1}(0) + C_{i+2}t_{i+2}(0) + Q_i - Q_{i+2}}{C_{i+2}} - \left(\frac{C_{i+1}}{C_{i+2}}\right)\frac{\{t_{i+2}(0) + \Phi_{C,i} + \Psi_{C,i}(t_{i+1}(0) - \Phi_{C,i})\}}{1 - \left(\frac{C_{i+1}}{C_{i+2}}\right)\Psi_{C,i}} + \left(\frac{C_{i+1}}{C_{i+2}}\right)\frac{\Psi_{C,i}\{C_{i+1}t_{i+1}(0) + C_{i+2}t_{i+2}(0) + Q_i - Q_{i+2}\}}{\left(1 - \frac{C_{i+1}}{C_{i+2}\Psi_{C,i}}\right)C_{i+2}} \tag{29}$$

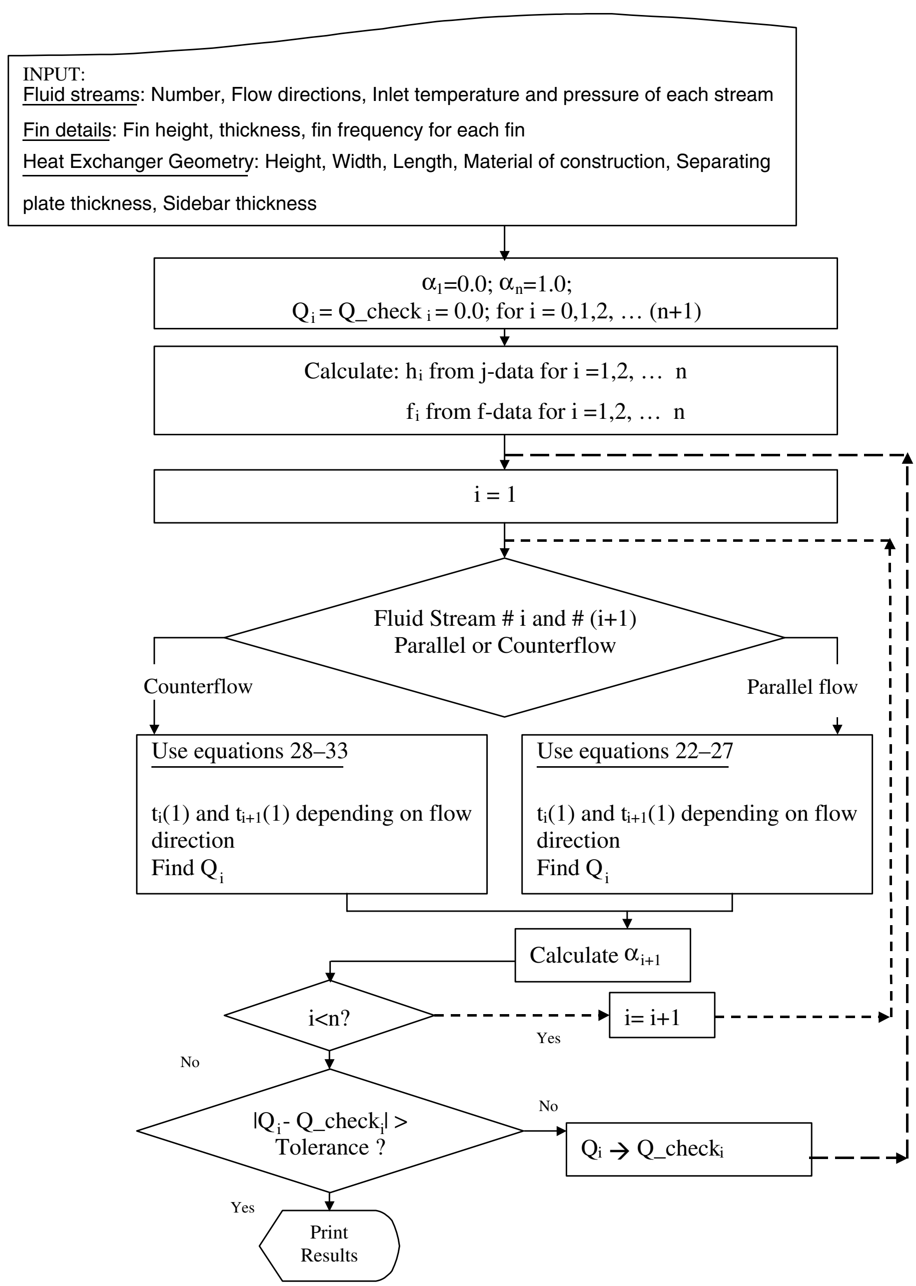

Fig. 6. Scheme to calculate outlet temperatures in a given section with known inlet condition of the fluid streams.

and

$$Q_{i+1} = C_{i+1}\{t_{i+1}(0) - t_{i+1,j}(1)\} + Q_i \tag{30}$$

where

$$\Phi_{C,i} = \frac{\left(\frac{Q_i}{C_{i+1}} - \frac{Q_{i+2}}{C_{i+2}}\right)}{U(A_\mathrm{w} + \alpha_{i+2}A_{\mathrm{f},i+2})\left(\frac{1}{C_{i+1}} - \frac{1}{C_{i+2}}\right)} \tag{31}$$

$$\Psi_{C,i} = \mathrm{e}^{-U(A_\mathrm{w}+\alpha_{i+2}A_{\mathrm{f},i+2})\left(\frac{1}{C_{i+1}}-\frac{1}{C_{i+2}}\right)} \tag{32}$$

and

$$\frac{1}{U(A_\mathrm{w} + \alpha_{i+2}A_{\mathrm{f},i+2})} = \frac{1}{h_{i+2}(A_\mathrm{w} + \alpha_{i+2}A_{\mathrm{f},i+2})} + \frac{1}{h_{i+1}\{A_\mathrm{w} + (1-\alpha_{i+1})A_{\mathrm{f},i+1}\}} \tag{33}$$

The temperature $t_{(i+1)}(1)$ computed from the two sub-exchangers must be equal when the solution finally converges.

Starting with a guess value for Q_2, each sub-exchanger is modelled in a sequential manner. Heat flow rates and fluid exit temperatures are computed using Eqs. (22)–(27) for co-current and Eqs. (28)–(33) for counter-current configurations. The process is continued till all heat flow rates and exit temperatures converge within pre-set tolerance limits. Fig. 6 gives a schematic flow diagram of the solution process for modelling of an n-stream heat exchanger of small overall length.

4. Successive partitioning method for multistream exchanger (axial direction)

The *area splitting* method is based on the assumption that every separating plate is at a single uniform temperature. This assumption is valid only for an exchanger of sufficiently small length. If we split a heat exchanger into a large number of small segments, each individual segment will qualify to be designed by the *area splitting* method. Some of the existing design suggests dividing the exchanger into arbitrary number of small segments at the beginning [24,26,27]. However, such partitioning may result in divergence of solution if the initial guess of the temperature profile is a poor one. For instance, the heat exchanger of example II in Ref. [25] does not yield results when Prasad's algorithm [27] is employed and the exchanger is divided into 20 or more parts. Fig. 7 shows how the predicted fluid exit temperature diverges with succeeding iterations. Consideration of a large number of segments from the beginning may also be computationally expensive.

In our approach, the heat exchanger is not partitioned into a large number of segments in one operation. Instead, the exchanger is initially divided only into two segments. The governing heat transfer and energy conservation equations are solved for each of the two segments yielding a new set of fluid outlet temperatures. The process of evaluating exit temperatures in each segment is repeated till all temperatures converge within predefined tolerance limits. Each segment of the exchanger is then further divided into two parts. Such division of the exchanger into successively smaller parts in multiples of 2 continues till fluid discharge

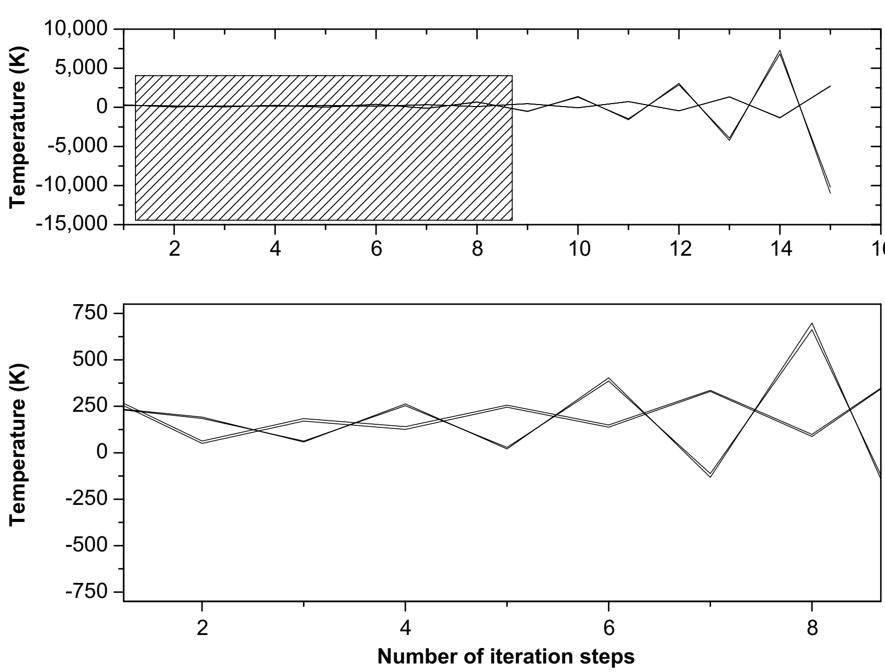

Fig. 7. Predicted outlet temperature shown as a function of iteration number in solution of example II of Ref. [25] using Prasad's [27] algorithm.

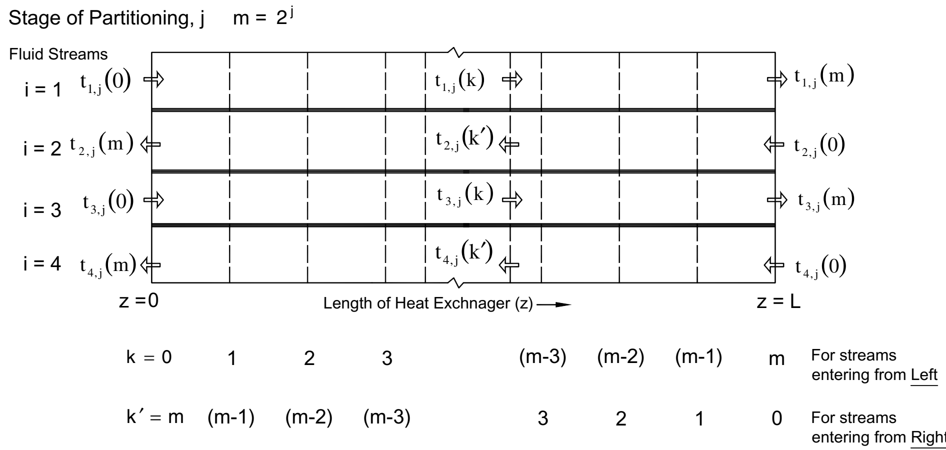

Fig. 8. The numbering scheme for partitioning of multistream heat exchange.

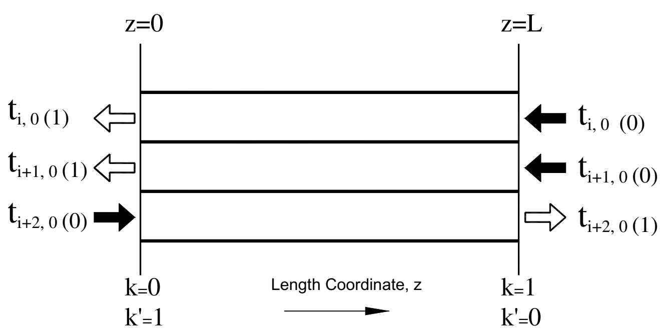

Fig. 9. Inlet and outlet temperatures for partition stage ($j = 0$).

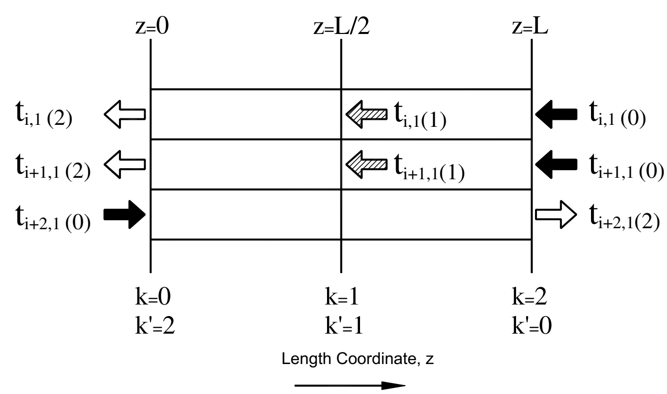

Fig. 10. Partitioning of the exchanger into two parts ($j = 1$).

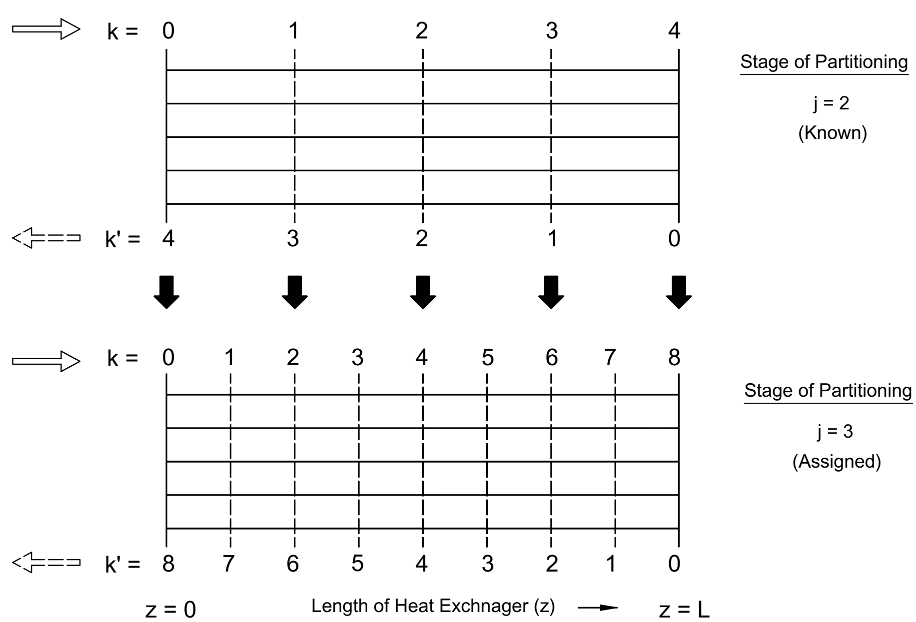

Fig. 11. Assignment of serial numbers to stations on partitioning. The example shows the case for $j = 3$. Values for $j = 2$ are known at the start of this stage of partitioning.

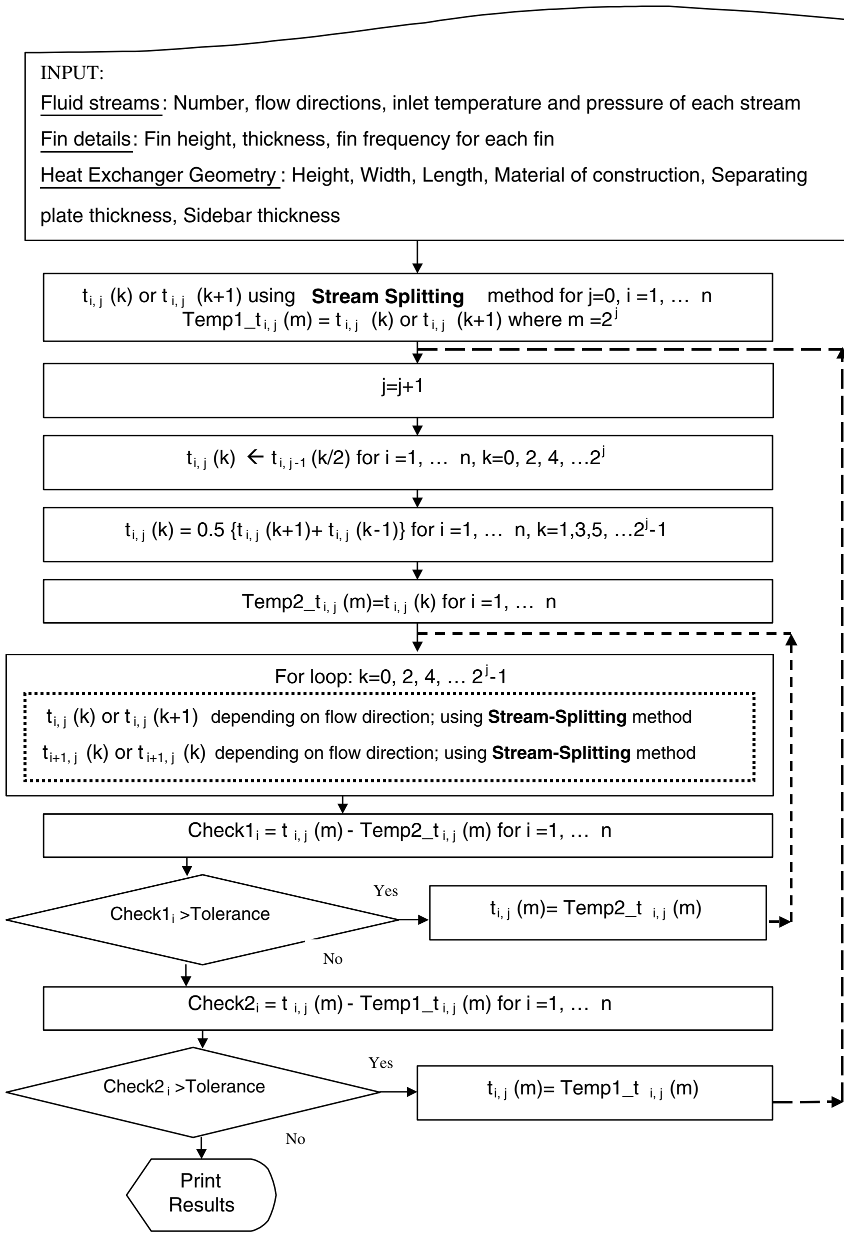

Fig. 12. Flow chart for successive partitioning of heat exchanger.

temperatures over two consecutive levels of partitioning agree within an acceptable difference. We call this division procedure the "*successive partitioning*" method. This technique helps avoiding divergence of solution, which may occur in the conventional one time partitioning technique proposed by Prasad and co-workers [23,24,26,27].

The fluid temperature, in its generalised form, is represented by the symbol $t_{i,j}(k)$ for streams flowing in the positive z-direction. Reverse flowing streams are denoted by $t_{i,j}(k')$ as shown in Fig. 8. The subscript i refers to the stream number as before. The subscript j has been introduced to distinguish fluid temperatures between stages of partitioning, $j = 0$ referring to the original (unpartitioned) exchanger. The number of segments m increases with the partitioning stage j as

$$m = 2^j \tag{34}$$

At any particular location, k is related to k' by the expression $k' = (2^j - k)$. According to this nomenclature, the inlet temperatures of the fluid streams irrespective of their entry from any end of the exchanger, always become $t_{i,j}(0)$ and the exit temperature $t_{i,j}(m)$. The separating plate temperatures, having no direction properties, are denoted by $T_{i,j}(k)$ with single parameter k in the parenthesis.

4.1. The complete exchanger ($j = 0$)

Considering the entire heat exchanger as a single segment, the *area splitting method* is employed to determine the unknown exit temperatures of all fluid streams. The temperatures are denoted as shown in Fig. 9.

4.2. Step 1 ($j = 1$)

In step ($j = 1$), the exchanger is divided into 2 segments, thus generating 3 points along the length: $z = 0$ ($k = 0, k' = 2$), $z = L/2$ ($k = 1, k' = 1$) and $z = L$ ($k = 2, k' = 0$). The temperature of all streams at $z = 0$ ($k = 0, k' = 2$) and $z = L$ ($k = 2, k' = 0$) are known from step 0; but those at $z = L/2$ ($k = 1, k' = 1$) are still unknown.

In order to be able to apply the *area splitting* method to the first segment (between locations $z = 0$ and $z = L/2$), we need the temperature of all reverse-flowing streams at $z = L/2$ ($k = 1, k' = 1$). These are guessed by linear interpolation between the corresponding temperatures at $z = 0$ ($k = 0, k' = 2$) and at $z = L$ ($k = 2, k' = 0$). These temperatures have been represented by the hatched arrows in Fig. 10. Once all the inlet temperatures are known (or guessed), the area splitting method is employed to compute all outlet temperatures, including those at $z = L/2$ ($k = 1, k' = 1$), which serve as the inlet stream to the second segment. The modelling of the second segment, in turn, yields the temperatures of the reverse flowing streams at $z = L/2$ ($k = 1, k' = 1$,) which replace the initial guesses (hatched arrows in Fig. 10).

The process is repeated, alternating between segments 1 and 2 till convergence is achieved for all temperatures within specified tolerance.

4.3. The general step j

If the exit temperatures computed in Step 1 ($j = 1$) do not agree with those computed in step 0 ($j = 0$), the exchanger is partitioned further, doubling the number of sections in each division. Thus at the completion of the $(j - 1)$th step, the exchanger has been divided into 2^{j-1} sections and all temperatures $[t_{i,(j-1)}(k)$ or $t_{i,(j-1)}(k')$, $k = 0$ to 2^{j-1}, $k' = 2^j - k$, $i = 1$ to n; $T_{i,\ j-1}(k)$, $k = 0$ to $2^{j-1} - 1$, $i = 0$ to $n]$ are known at appropriate level of accuracy. In the jth step, the exchanger is partitioned further creating 2^j segments. The serial number of each physical station changes and the new temperatures are copied from the older ones through the following transformation:

$$t_{i,j}(k) = t_{i,j-1}(k/2) \quad \text{for } k = 0, 2, \ldots 2^j \text{ and } i = 1 \text{ to } n$$
$$t_{i,j}(k') = t_{i,j-1}(k'/2) \quad \text{for } k' = 0, 2, \ldots 2^j \text{ and } i = 1 \text{ to } n \tag{35}$$

The temperatures at $k = 1, 3, \ldots, 2^j - 1$ are unknown. These temperatures for the fluid streams moving from right to left are estimated as arithmetic mean of the nearest known temperatures, i.e.

$$t_{i,j}(k') = [t_{i,j}(k' - 1) + t_{i,j}(k' + 1)]/2$$
$$\text{for } k' = 1, 3, 5, \ldots 2^j - 1 \text{ and } i = 1 \text{ to } n \tag{36}$$

The "area splitting" algorithm is first applied to the first segment $k = 0 \to k = 1$ ($k' = 2^j \to k' = 2^j - 1$) to generate the outgoing stream temperatures at $k = 1$ ($k' = 2^j - 1$, $z = L/2^j$). The calculated temperatures of the streams flowing from left to right at $k = 1$ ($k' = 2^j - 1$) are used to solve the equation set for the second segment $k = 1 \to k = 2$ ($k' = 2^j - 1 \to k' = 2^j - 2$). The process is repeated till the last segment $k = 2^j - 1 \to k = 2^j$ ($k' = 1 \to k' = 0$) is reached. The computation process is repeated, starting from the first segment ($k = 0 \to k = 1$ or $k' = 2^j \to k' = 2^j - 1$) and ending at the last ($k = 2^j - 1 \to k = 2^j$ or $k' = 1 \to k' = 0$), till the stream exit temperatures at $k = 0$ ($k' = 2^j$) and at $k = 2^j$ ($k' = 0$) converge within prescribed tolerance limits. An example of the partitioning process in step $j = 3$ is shown schematically in Fig. 11.

The algorithm of the successive partitioning method is shown with the help of a flow chart in Fig. 12. A computer program has been written in C++ to execute the successive partitioning and stream splitting techniques for design of multistream plate fin heat exchangers. Input and output of data are in a user-friendly format and the program

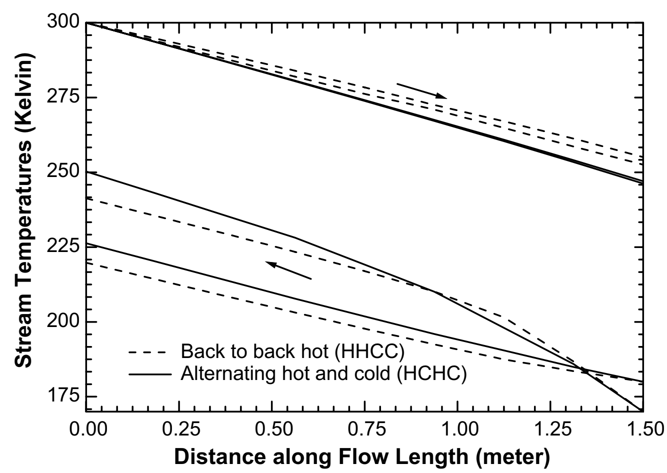

Fig. 13. Temperature profiles in a 4-stream plate fin heat exchanger (Example I in Ref. [25]) simulated using present algorithm.

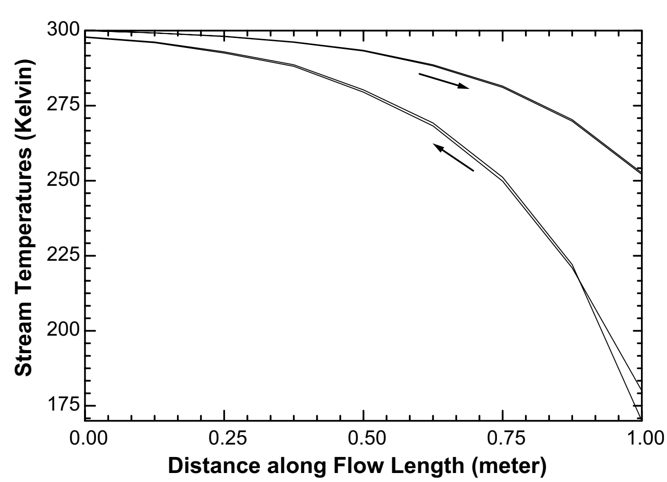

Fig. 14. Temperature profiles in a 4-stream plate fin exchanger (Example II in Ref. [25]) simulated using present algorithm.

can be used for professional design of plate fin heat exchangers.

5. Validation of the design approach

To check the validity of the proposed model, several multistream plate fin heat exchangers analysed in published literature [25] have been modelled using the present algorithm. Fig. 13 shows the performance of a four-stream plate fin heat exchanger with different layer-stacking patterns. Results for one of the stacking patterns (HCHC) is presented with solid line, while those for another (HHCC) are shown with dotted line. The change in performance on varying the stacking pattern is clearly seen. The fluid temperature profiles are in good agreement with the results of Paffenbarger [25], except for the fluid stream with entry temperature of 170 K in "back-to-back hot" configuration. This deviation is caused due to the extremely low mass flow rate of that fluid stream. Performance prediction of another four-stream plate fin heat exchanger is shown in Fig. 14. Results exactly match with the reported data.

The proposed method has also been verified by using the experimental results obtained by Luo et al. [28] on a multistream plate fin heat exchanger. A good agreement between the results predicted by the present algorithm and those reported by the authors can be seen from the graphs (Figs. 15–17).

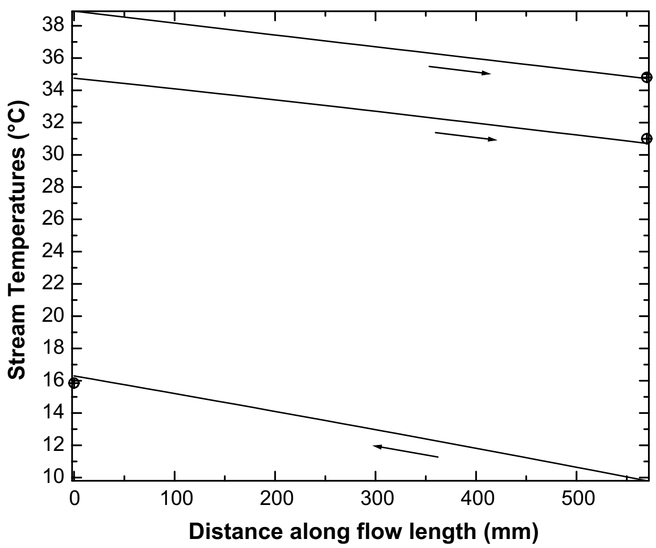

Fig. 15. Temperature profile of 3-stream exchanger (Experiment I in Ref. [28]) ((—) calculated temperature profile; (⊕) experimental observations of Luo et al. [28]).

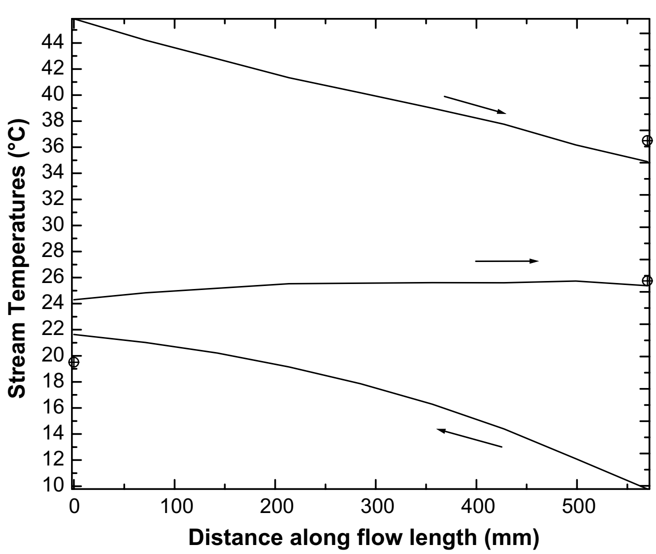

Fig. 16. Temperature profile of 3-stream exchanger (Experiment II in Ref. [28]) ((—) calculated temperature profile; (⊕) experimental observations of Luo et al. [28]).

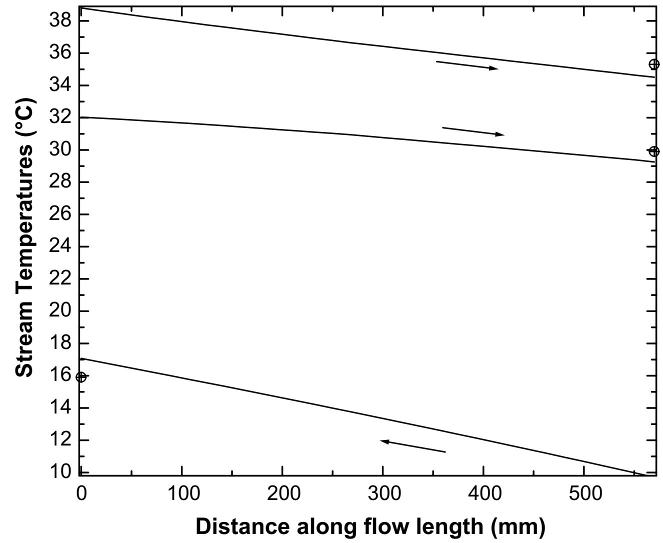

Fig. 17. Temperature profile of 3-stream exchanger (Experiment III in Ref. [28]) ((—) calculated temperature profile; (⊕) experimental observations of Luo et al. [28]).

6. Conclusion

In summary, it may be said that the *successive partitioning* and *area splitting* methods provide a practical algorithm for design of multistream plate fin heat exchangers. The concept of *area splitting* helps to analyse the multistream heat exchanger as a stack of two-stream heat exchangers interacting with each other. This technique gives definite advantage in the analysis of multistream heat exchangers. The logic of *successive partitioning* is also unique and provides a safe guard against divergence. The algorithm based on these two formalism is simple and the program is fast. The results agree with published theoretical prediction and experimentally observed data.

While the boundary thermal resistance and geometrical effects are adequately taken care of by the algorithm, it still does not consider several secondary sources of irreversibility such as axial conduction, flow maldistribution, temperature dependent fluid properties and heat exchange with the surroundings. In high effectiveness heat exchangers, particularly those used in cryogenic applications, these secondary irreversibilities may play a controlling role. Incorporation of these effects into the design program constitutes the subject of a separate paper under preparation. Further, though the present algorithm has been employed for plate fin heat exchangers the basic design principle can be used for any multistream heat exchanges.

References

[1] W.M. Kays, A.L. London, Compact Heat Exchangers, third ed., McGraw-Hill, New York, 1984.

[2] F.P. Incropera, D.P. DeWitt, Fundamentals of Heat Transfer, second ed., John Wiley, New York, 1985.

[3] R.K. Shah, P. Dusan, D.P. Sekulic, Fundamentals of Heat Exchanger Design, John Wiley & Sons, 2003.

[4] R.K. Shah, Compact Heat Exchangers & Enhancement Technology, Begell House Pub, 1999.

[5] R.K. Shah, Heat exchanger basic design methods, in: S. Kakac, R.K. Shah, A.B. Bergles (Eds.), Low Reynolds Number Flow Heat Exchangers, Hemisphere, New York, 1983, pp. 21–72.

[6] A.L. London, Compact heat exchangers – design methodology, in: S. Kakac, R.K. Shah, A.B. Bergles (Eds.), Low Reynolds Number Flow Heat Exchangers, Hemisphere, New York, 1983, pp. 21–72.

[7] R.K. Shah, Compact heat exchanger design procedures, in: S. Kakac, A.E. Bergles, F. Mayinger (Eds.), Heat Exchangers, Thermal-Hydraulic Fundamentals and Design, Hemisphere Publication, 1980, pp. 495–536.

[8] E.M. Smith, Plate fin surface optimisation using direct-sizing, Adv. Enhanced Heat Transfer, ASME 4 (2000) 105–115.

[9] A.P. Frass, Heat Exchanger Design, second ed., Wiley-Interscience, 1989.

[10] P.G. Kroeger, Performance deterioration in high effectiveness heat exchangers due to axial heat conduction effects, Adv. Cryogen. Eng. 12 (1967) 363–372.

[11] K. Chowdhury, S. Sarangi, Performance of cryogenic heat exchangers with heat leak from the surroundings, Adv. Cryogen. Eng. 29 (1984) 273–280.

[12] L. Haseler, Performance calculation methods for multistream plate fin heat exchangers, in: J. Taborek, G.F. Hewitt, N. Afgan (Eds.), Heat Exchangers – Theory and Practice, Hemisphere Publishing, New Work, 1983, pp. 495–506.

[13] Y.N. Fan, How to design plate fin heat exchangers, Hydrocarbon Process. 45 (11) (1966) 211–217.

[14] T. Sorlie, Three fluid heat exchanger design theory, counter and parallel flow, Technical Report 54, Department of Mechanical Engineering, Stanford University Stanford, 1962.

[15] D.D. Aulds, R.F. Barron, Three-fluid heat exchanger effectiveness, Int. J. Heat Mass Transfer 10 (1967) 1457–1462.

[16] D.P. Sekulic, R.K. Shah, Thermal design theory of three-fluid heat exchanger, Adv. Heat Exchanger 26 (1995) 219–327.

[17] S. Kao, A systematic design approach for a multistream exchanger with interconnected wall, ASME Paper 61-WA-255, 1961.

[18] J. Wolf, General solution of the equations of parallel flow multichannel heat exchangers, Int. J. Heat Mass Transfer (1964) 901–919.

[19] T. Zaleski, A general mathematical model of parallel flow, multichannel heat exchangers and analysis of its properties, Chem. Eng. Sci. 39 (7/8) (1984) 1251–1260.

[20] J.C. Chato, R.J. Lverman, J.M. Shah, Analyses of parallel flow multistream heat exchangers, Int. J. Heat Mass Transfer 14 (1971) 1691–1703.

[21] M. Bentwich, Multistream countercurrent heat exchangers, Trans. ASME (1973) 458–463.

[22] B.S.V. Prasad, S.M.K.A. Gurukul, Differential method for sizing multistream plate fin heat exchangers, Cryogenics 27 (1987) 257–262.

[23] B.S.V. Prasad, S.M.K.A. Gurukul, Differential methods for the performance prediction of multistream plate fin heat exchangers, J. Heat Transfer 114 (1992) 41–49.

[24] B.S.V. Prasad, The performance prediction of multistream plate fin heat exchangers based on stacking pattern, Heat Transfer Eng. 12 (1991) 58–70.

[25] J. Paffenbarger, General computer analysis of multistream plate fin heat exchangers, in: R.K. Shah, A.D. Kraus, D. Metzger (Eds.), Compact Heat Exchangers – A Festschrift for A.L. London, Hemisphere Publishing, New York, 1990, pp. 727–746.

[26] B.S.V. Prasad, Fin efficiency and mechanisms of heat exchange through fins in multistream plate fin heat exchangers: formulation, Int. J. Heat Mass Transfer 39 (1996) 419–428.

[27] B.S.V. Prasad, Fin efficiency and mechanisms of heat exchange through fins in multistream plate fin heat exchangers: Development and application of a rating algorithm, Int. J. Heat Mass Transfer 40 (1997) 4279–4288.

[28] X. Luo, K. Li, M. Li, Prediction of the thermal performance of multistream plate fin heat exchangers, Int. J. Heat Exchangers 2 (2001) 47–60.

[29] X. Luo, M. Li, W. Roetzel, A general solution for one dimensional multistream heat exchangers and their networks, Int. J. Heat Mass Transfer 45 (2002) 2695–2705.

[30] G.T. Polley, M. Picon Nunez, Understanding Multistream Heat Exchanger Design, February, 2001. Available from: <www.pinchtechnolgy.com>.

[31] G.T. Polley, M. Picon Nunez, Methodology for the design of multistream plate-fin heat exchangers, in: B. Sunden, P.J. Heggs (Eds.), Recent Advances in Analysis of Heat Transfer for Fin Type Surfaces, WIT Press, 2000, pp. 251–276.

[32] M. Picon Nunez, G.T. Polley, M. Medina-Flores, Thermal design of multistream heat exchangers, Appl. Thermal Eng. 22 (2002) 1643–1660.

[33] L. Wang, B. Sunden, Design methodology for multistream plate fin heat exchangers in heat exchangers networks, Heat Transfer Eng. 22 (6) (2001) 3–11.

[34] H. Pingaud, J.M. Le Lann, B. Koehret, Steady-state and dynamic simulation of plate fin heat exchangers, Comput. Chem. Eng. 13 (4–5) (1989) 577–585.

[35] X. Luo, X. Guan, M. Li, W. Roetzel, Dynamic behaviour of one-dimensional flow multistream heat exchangers and their networks, Int. J. Heat Mass Transfer 46 (2003) 705–715.