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Fin efficiency and mechanisms of heat exchange through fins in multi-stream plate-fin heat exchangers: development and application of a rating algorithm

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Abstract—The basic heat balance equations for separating surface and passage in a multi-stream plate-fin heat exchanger are derived based on a formalism proposed earlier [Prasad, *International Journal of Heat and Mass Transfer* 1996, **39**(2), 419–428]. An algorithm is developed for rating heat exchangers based on the equations, and incorporated into an existing computer code called STACK [Prasad, *Heat Transfer Engineering*, 1991, **12**(4), 58–70]. The program is used to analyse some typical heat exchangers. It was found that though transverse conduction could be present in some and absent in some passages, it tended to play an increasingly important role as fin effectiveness increased. Of the three mechanisms of heat exchange identified, Mechanism 1 and Mechanism 3 dominated at low fin effectiveness, whereas Mechanism 2 tended to dominate at high fin effectiveness. It was found that the developed method reliably predicted heat transfer in special situations such as the presence of dummy passages, where other methods were known to fail. © 1997 Elsevier Science Ltd.

1. INTRODUCTION

Multi-stream plate-fin heat exchangers are used wherever high performance, low volume and weight, and multi-functionality are mandated. They are used in cryogenic gas processing as main feed pre-coolers, condensers, reboilers and liquid chillers, in the aerospace industry as oil and fuel coolers, radiators and charge coolers, and in the HVAC industry as air to air exchangers, condensers and evaporators. They are often designed for very high effectiveness (design NTU up to 100 for balanced gas to gas exchangers in cryogenic process application), and usually offer a compactness of up to 1 : 50 in terms of volume and weight for the same thermal duty when compared to more conventional heat transfer equipment. This compactness is made possible by surface area densities of up to 5000 m^2/m^3. Cryogenic heat exchangers have been designed to handle up to 14 hot and cold gas streams in a single compact core.

The effectiveness-NTU method has been successfully applied to the design of plate-fin exchangers handling two streams, since Kays and London [1–5]. Attempts have since been made to extend this method to handle multi-stream exchangers [6], but the difficulties in applying this and other similar methods to the design of multi-stream plate-fin heat exchangers have led to the investigation of alternative methods over the past few years [6–17]. The methods presented in the literature have ranged from purely sizing [10, 11, 14] to sizing/rating [7, 12] to purely rating [5, 6, 8, 9, 13–18]. The models used in presented rating methods have varied from the relatively simple constant wall temperature model [7, 13, 15] to more complex multi-stream [6, 15] to multi-passage [9, 12, 16, 18] models. It is known that a general multi-passage approach is the most realistic for the design of these units over other approaches (see discussion in [16, 17]), because of the effects of stacking and transverse conduction effects (i.e. transfer of heat between adjacent separating surfaces by conduction through fins). Kao [8] was one of the first to deal at length with the development of a general design theory for multi-stream plate-fin heat exchangers based on the multi-passage concept. His theory has been adapted or adopted by many others [6, 9, 12, 18]. These authors have considered transverse conduction to be a general feature of heat transfer in multi-stream plate-fin heat exchangers and, hence, present in all passages; they have used a general 'by-pass efficiency' term for it. Others have concluded that no transverse conduction could be present as most passages would have a temperature extremum in their fin temperature profiles approximately in the middle of the fin [6]. Prasad [16] has earlier used the 'half-fin-length' formalism (i.e. the assumption that half the fin length in a given passage is involved in transferring heat to the neighbouring passage on either side) for heat transfer calculation, and a linear conduction model for transverse conduction, which could be switched on and off at will, much as could be done with the bypass heat transfer in the earlier models. Primarily to resolve these issues, and

NOMENCLATURE

A_p	primary surface area of passage [m^2]
C	capacity rate of passage $= [m \cdot c_p]$ [W K^{-1}]
c_p	specific heat of stream in passage [J kg^{-1} K^{-1}]
h	heat transfer coefficient of stream [W m^{-2} K^{-1}]
k	thermal conductivity of fin [W m^{-1} K^{-1}]
l	fin length [m]
m	factor $= \sqrt{(2h/kt)}$ [m^{-1}]
m	mass flow rate of stream [kg s^{-1}]
n	number of passages in exchanger
N	number of section in exchanger
q	quantity of heat entering/leaving subscripted fin base (always positive when leaving fin base, and negative when entering fin base) [W m^{-1}]
r	ratio of temperature differentials $= \theta_B/\theta_A$ dimensionless at fin bases
t	fin thickness [m]
T	temperature [K]
x	distance from fin base A [m]
X	distance from fin base A of the point of extremum or point of null temperature differential, depending on case [m].

Greek symbol

ϕ	defined in Section 2, $= \sqrt{2hkt}/\sinh(ml)$.

Subscripts

A	value for the A section of fin, attached to fin base A
B	value for the B section of fin, attached to fin base B
in	entry value
out	exit value
i	passage number.

to generate a sufficiently general formalism for the design of multi-stream plate-fin heat exchangers, Prasad [17] made a detailed analysis of the heat transfer in a passage of a multi-stream plate-fin heat exchanger with an arbitrary arrangement of passages. The possible temperature profiles of a fin with fin bases at unequal temperatures appears as Fig. 1. The following three mechanisms of heat transfer were identified.

Mechanism 1

An extremum would be present in the fin temperature profile when both fin bases have either higher or lower temperatures than the stream in the passage. The extremum, when present within the physical boundaries of the fin, imposes an adiabatic barrier across which no heat transfer can take place (Fig. 1a). No transverse conduction is possible in this mechanism.

Mechanism 2

It is possible for the extremum to fall outside the fin proper when both fin bases have either higher or lower temperatures than the stream in the passage. In this case, there would be transverse conduction (Fig. 1b).

Mechanism 3

Transverse conduction would also be present in the case of the stream having a temperature intermediate to the temperature of the fin bases (Fig. 1c).

For all the above cases, the equations for the quantity of heat entering/leaving the fin bases are identical,

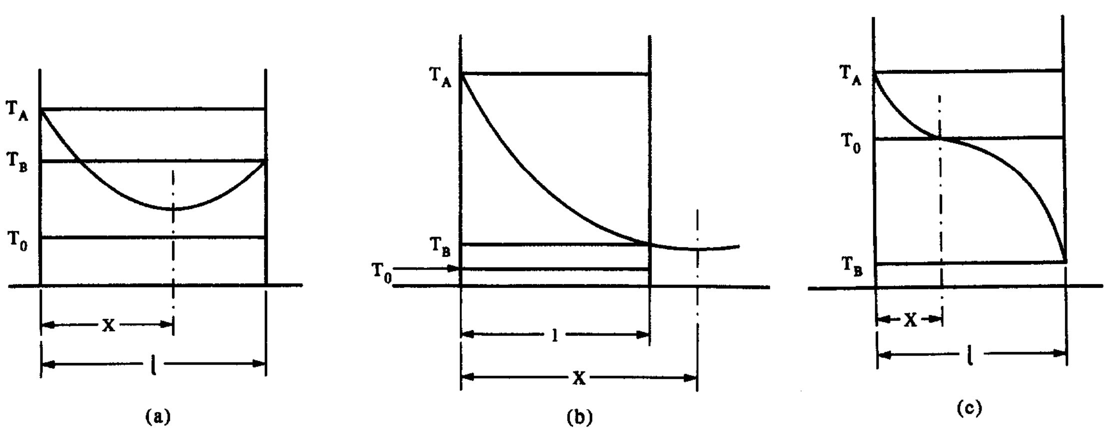

Fig. 1. Temperature profiles of a fin with bases at unequal temperatures: (a) temperature extremum in fin; (b) temperature extremum outside fin; (c) no temperature extremum.

and are given by equations (1) and (2) (see Prasad [17] for the detailed derivations):

$$q_A = \frac{\sqrt{2hkt}}{\sinh(ml)} \cdot (\theta_A \cosh(ml) - \theta_B) \quad (1)$$

$$q_B = \frac{\sqrt{2hkt}}{\sinh(ml)} \cdot (\theta_B \cosh(ml) - \theta_A) \quad (2)$$

where

$$\theta_A = (T - T_A) \quad (3)$$

$$\theta_B = (T - T_B) \quad (4)$$

and

$$T = (T_{in} + T_{out})/2. \quad (5)$$

Since the stream temperature in a passage of a multi-stream plate-fin heat exchanger could be above or below that of either wall, it is evident that transverse conduction would be present in some passages, and absent in others. More importantly, since the location of the temperature extremum determines the amount of fin surface devoted to heat transfer to the neighbouring passage on either side, rating methods based on the 'half-fin-length' idealization are likely to be in error in many situations.

Equations (1) and (2) above formed the basis for developing a general method for the rating of a multi-stream plate-fin heat exchanger, which automatically and realistically accounted for the effects of both transverse conduction and stacking arrangement. The method is based on a rigorous solution of the heat balance equations for each passage and separating surface. The present paper discusses the development of the method and its application in a computer program.

2. DEVELOPMENT OF THE HEAT BALANCE EQUATIONS FOR SEPARATING SURFACE AND PASSAGE

Figure 2 is a schematic of heat transfer at a cross-section of a multi-stream plate-fin heat exchanger. The passages and separating surfaces have been number from the top. The detail shows the stream and separating surface temperatures, and heat transfer between the ith passage and its neighbours. As outlined elsewhere [17], the following heat balances could be written for the ith separating surface and passage:

$$q_{p,i,i+1} + q_{A,i} + q_{p,i+1,i} + q_{B,i+1} = 0 \quad (6)$$

(for heat transfer across the ith surface)

$$q_{p,i,i+1} + q_{A,i} + q_{p,i,i-1} + q_{B,i} = C_i(T_{in,i} - T_{out,i}) \quad (7)$$

(for heat transfer in the ith passage) putting

$$\phi = \frac{\sqrt{2hkt}}{\sinh(ml)} \quad (8)$$

substituting relations (1)–(5) in equation (1) and rearranging

$$\begin{aligned} & T_{s,i-1}(\phi_i) + T_{out,i}\left(\frac{h_i A_{p,i}}{4} + \frac{\phi_i \cosh(m_i l_i)}{2} - \frac{\phi_i}{2}\right) \\ & + T_{s,i}\left(-\frac{h_i A_{p,i}}{2} - \frac{h_{i+1} A_{p,i+1}}{2} - \phi_i \cosh(m_i l_i) - \phi_{i+1}\cosh(m_{i+1} l_{i+1})\right) \\ & T_{out,i+1}\left(\frac{h_{i+1} A_{p,i+1}}{4} + \frac{\phi_{i+1}\cosh(m_{i+1} l_{i+1})}{2} - \frac{\phi_{i+1}}{2}\right) + T_{s,i+1}(\phi_{i+1}) \\ & = T_{in,i}\left(-\frac{h_i A_{p,i}}{4} - \frac{\phi_i \cosh(m_i l_i)}{2} + \frac{\phi_i}{2}\right) \\ & + T_{in,i+1}\left(-\frac{h_{i+1} A_{p,i+1}}{4} - \frac{\phi_{i+1}\cosh(m_{i+1} l_{i+1})}{2} + \frac{\phi_{i+1}}{2}\right). \end{aligned} \quad (9)$$

Similarly, equation (2) can be rearranged to

$$\begin{aligned} & T_{s,i-1}\left(-\frac{h_i A_{p,i}}{2} - \phi_i \cosh(m_i l_i) + \phi_i\right) \\ & + T_{out,i}\left(\frac{h_i A_{p,i}}{2} + \phi_i \cosh(m_i l_i) - \phi_i + C_i\right) \\ & + T_{s,i}\left(-\frac{h_i A_{p,i}}{2} - \phi_i \cosh(m_i l_i) + \phi_i\right) \\ & = T_{in,i}\left(-\frac{h_i A_{p,i}}{2} - \phi_i \cosh(m_i l_i) + \phi_i + C_i\right). \end{aligned} \quad (10)$$

q_B for the top passage and q_A for the bottom passage would be zero if the heat exchanger were considered adiabatic with the surroundings. Equations (9) and (10) are modified accordingly for these passages. Since a total of $(2n-1)$ of these equations can be written, involving as many unknowns (n exit temperatures and $n-1$ surface temperatures), a suitable matrix method can be used to solve the above set of equations.

3. IMPLEMENTATION OF A COMPUTER ALGORITHM FOR RATING

The implementation of the computer algorithm for rating multi-stream plate-fin heat exchangers based on equations (9)–(12) is very similar to that used for the computer program STACK (Fig. 3) described in detail elsewhere [16]. Essentially, the exchanger is divided into a number of longitudinal sections in each of which the temperature range is sufficiently small that the thermohydraulic properties of the streams and the physical properties of the exchanger material can be

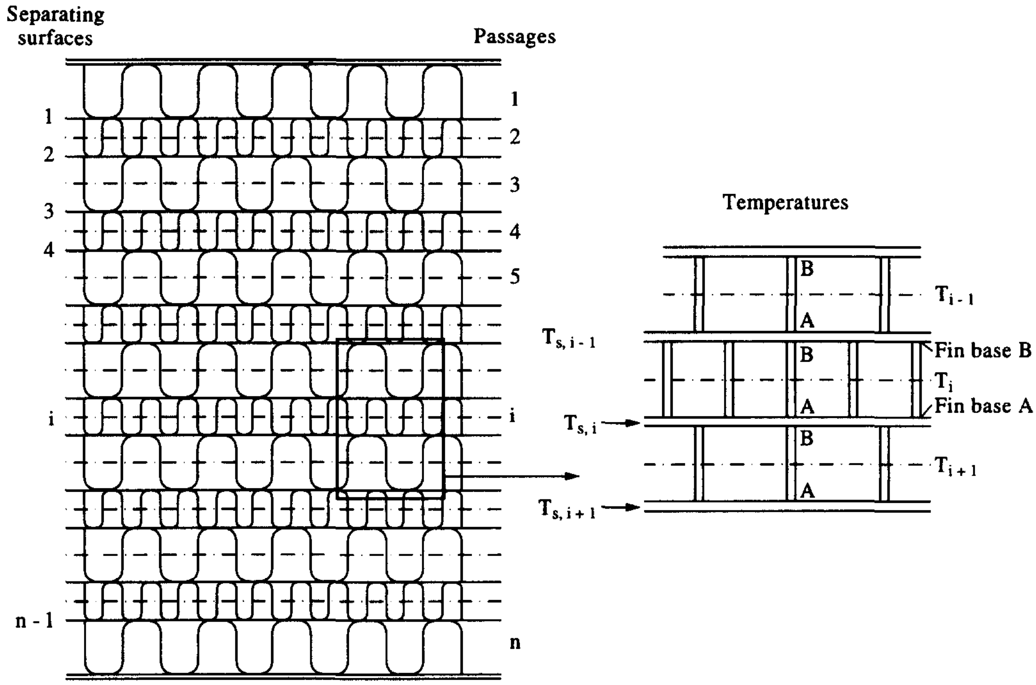

Fig. 2. Heat transfer in a multi-stream plate-fin heat exchanger.

considered constant. An iterative process starts with estimated inter-sectional temperatures for each passage. An error function based on the differences in inter-sectional temperatures and/or net heat exchanged between two successive iterations is defined, and the iterations are continued until the value of the error function falls below a predetermined tolerance i.e. until the variation of temperature profiles of streams in the exchanger between successive iterations is found to be insignificant.

It should be noted that the above method does not depend on guess values of exit temperatures at any stage, i.e. it is not a 'shooting' method [18]. The method works only with entry temperatures for all passages at all sections. The iteration can be started with any reasonably assumed temperature profiles for the streams, such as assuming that they continue at their entry temperatures. The method is robust and convergence is steady even for highly unbalanced exchangers, and exchangers with condensation/vapourization in some passages, in which some of the streams present a nearly constant temperature profile over a considerable part of their flow length in the exchanger.

It can be seen that the coefficient matrix defined by equations (9) and (10) is a non-symmetric, pentadiagonal matrix. Because of this, all the non-zero elements have to be stored, and general methods of solution have to be employed, unlike FE methods in structural analysis where usually symmetric matrices are encountered, and advantage can be taken of the symmetry in both the storage and solution of the equations.

The above method is to be classified as a combination of finite element and finite difference methods (a 'differential' or difference based method). The calculation of the sectional exit and surface temperatures described above, with the use of a closed-form solution obtained by the integration of the governing differential equation for the proper boundary conditions [17], constitutes a finite element method. The use of the basic equations in the overall solution of the entire exchanger in the longitudinal direction, which forms the iterated outer loop, is a finite difference method. This combination of methods possesses certain advantages over a general finite element scheme. In a general FEM solution, applied over the entire exchanger, the total number of unknowns would be $N(2n-1)$. Since general methods of solution (such as variants of the Gauss or Gauss–Jordan methods) have to be employed in this case, the total number of floating point operations is nearly proportional to the cube of the number of unknowns (see Forsythe and Moler [19] for a discussion of various methods).

Accordingly, the solution time for the temperature profiles would be proportional to the cube of the above number for each iteration. In addition, the numerical integration of the basic differential equations (implicit in the use of a general FEM solver) in each section adds a substantial computational overhead. As opposed to this, in the present method, the total unknowns are only $(2n-1)$ at any stage, and the solution time is proportional to $N(2n-1)^3$, per iteration. If the number of iterations required to reach a solution is considered to be of the same order of magnitude in both methods, then clearly the present method is faster by at least a factor of 'N^2'. The memory requirements are also smaller by the same factor

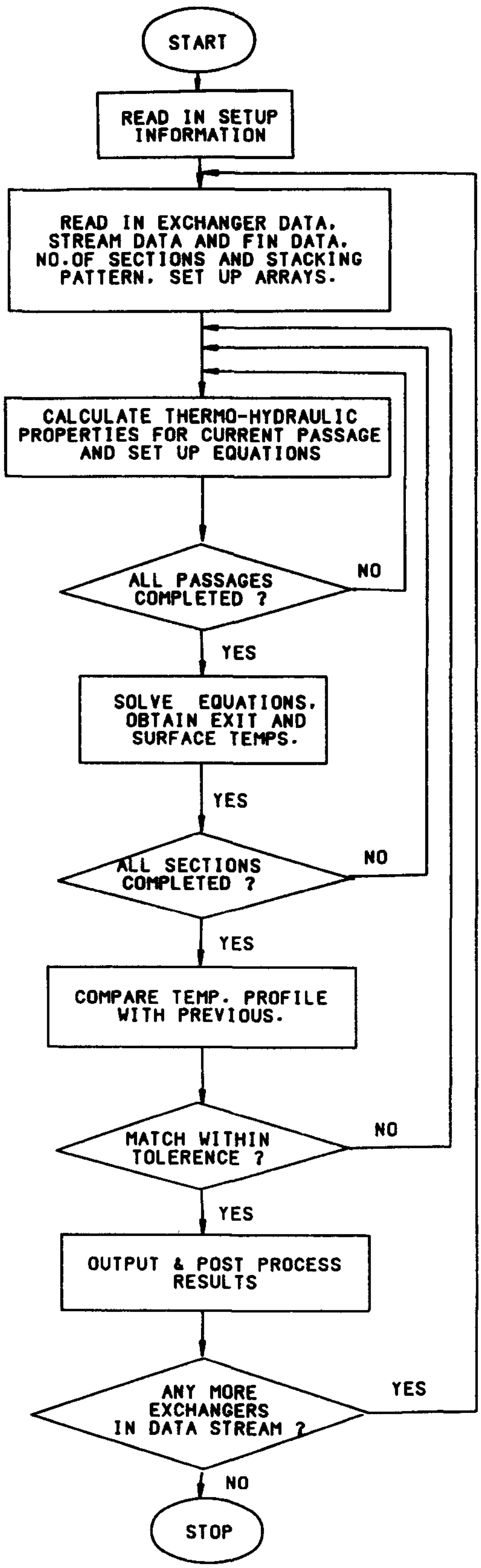

Fig. 3. Flowchart of computer program STACK.

as only $(2n-1)$ equations are required to be stored at any time (see Paffenbarger [18] for a discussion of memory and run time requirements with a general FEM solver). As a consequence of this, fewer floating point operations are needed in the solution of the equations, and round-off and truncation errors are lower. In fact, single precision storage of the variables has been found to be sufficient, and this further reduces the memory requirement in comparison with a general FEM solution.

The computer program STACK has been modified to set up and solve equations (9)–(12), while still retaining the code for using the 'half-fin-length' formalism for which it was originally developed [16]. The user has the option to use either formalism. The program is written in FORTRAN-77 and is currently implemented on both a MicroVAX II computer, as well as an IBM PC with a Pentium processor. A number of useful features, such as: (i) intermediate entry and exit and streams; (ii) partial drawal of streams at intermediate locations; (iii) interim fin changes; (iv) a more rigorous comparison of temperatures profiles in successive iterations for deciding convergence; (v) calculation of fin temperature profile, with location of temperature extremum, and the presence or absence of transverse conduction in each passage etc., have been incorporated. Algorithms for the calculation of secondary effects, such as flow maldistribution, longitudinal conduction and heat exchange with ambient are currently being developed. The program has extensive graphical facilities, with which the user can produce passage-wise plots of the surface and stream temperature profiles, on screen, printer as well as on a CALCOMP plotter. The PC version of STACK can also generate bitmap files of these graphs compatible with the Windows operating system. A Windows-hosted GUI front-end called STACKDEV has been developed for use with the PC version of STACK, which can be used to create data files, run STACK, view and print output and graphics files etc. STACKDEV also supports an extensive on-line help system.

Solution times depend primarily on the number of passages in the problem definition. Balanced exchangers are solved 4–5 times faster when compared to unbalanced exchangers. Balanced exchangers involving no complications (such as interm drawals, fin changes etc) rarely need to be divided into more than 20 sections; with others, more sections may be needed. Large problems involving balanced exchangers with upto 80 passages have been solved in less than a minute on the IBM PC with the Pentium processor running at 100 MHz.

Other computer programs similar to STACK in function and capability have been reported. Haseler [9, 20] described a computer program called MULE for rating multi-stream plate-fin heat exchangers, which has been available to members of the Heat Transfer and Fluid Flow Service (HTFS) for a number of years. Taylor and Starling [21] detailed the features of a program called PFeX, which can be used as a general design tool. Paffenbarger [18] described a program called MSE based on the COLSYS general FE modelling package, discussed the development of the algorithm, and reported a number of results. The original references should be reviewed for greater detail about these programs.

4. DISCUSSION OF RESULTS

An important aspect of the present formalism, as shown in ref. [16] as well as in the discussion above, is the presence of absence of transverse conduction between adjacent separating surfaces, depending on the temperature profile of the fin connecting them. To study this aspect, the main heat exchangers in some existing air separation plants have been analysed using STACK. The typical results for one such unit have been summarised in Table 1, Case 2. It can be seen that transverse conduction is present in 35–50% of the passages, and absent in the remaining. Note the large number of passages in which transverse conduction exists because the extremum falls outside the fin proper, i.e. Mechanism 2 is present. Also note that the situation keeps changing with location. It is evident that models which either completely neglect or assume transverse conduction in all passages could underpredict or overpredict the thermal performance, particularly when thermal conductivity of the fin material, and fin thickness are high, and significant imbalances in heat transfer rates exist [15]. Table 1 demonstrates the shift in the mechanism of heat exchange with increasing fin effectiveness (i.e. as $m \to 0$), simulated by increasing the thermal conductivity of the exchanger material. Note that as fin effectiveness increases, the less efficient Mechanisms 1 and 3 increasingly give way to Mechanism 2, as was surmised earlier [17]. The thermal performance of the exchanger for the above three cases is presented in Table 2, and Fig. 4 shows the surface temperature profiles of the exchanger for the cases. Note how the surface temperatures across the block even out as fin effectiveness increases, indicating increased transverse conduction [16, 17]. When the transverse conductance becomes infinite, a constant surface temperature results across the block [16]. The constant surface temperature formalism predicts the ideal performance in a multi-stream exchanger [16]. For the sake of comparison, the thermal performance predicted by the

Table 1. Mechanisms of heat exchange in multi-stream plate-fin heat exchanger

SNo.	Distance from hot end (m)	No. of passages with extremum, but no transverse conduction (Mechanism 1)	No. of passages with extremum, but no transverse conduction (Mechanism 2)	No. of passages with extremum, but no transverse conduction (Mechanism 3)	Total with transverse conduction
Case 1 : Thermal conductivity of exchanger material 1/10th normal value					
1.	0.34	46	6	11	17
2.	1.02	47	4	12	16
3.	1.70	51	1	11	12
4.	2.38	52	2	9	11
Case 2 : Thermal conductivity of exchanger material normal					
1.	0.34	37	22	4	26
2.	1.02	33	27	3	30
3.	1.70	34	23	6	29
4.	2.38	40	23	0	23
Case 3 : Thermal conductivity of exchanger material 10 times normal value					
1.	0.34	29	34	0	34
2.	1.02	27	36	0	36
3.	1.70	24	39	0	39
4.	2.38	23	40	0	40

Table 2. Thermal performance of multi-stream plate-fin heat exchanger predicted by various formalisms

Stream (1)	Entry temp (K) (2)	Predicted exit temps (K) for each formalism: Without trans. cond. (3)	Half-fin-length With trans. cond. (k) (4)	Present (k/10) (5)	Present (k) (6)	Present ($k \times 10$) (7)	Constant surface temp (8)
A	203	101.6	100.7	103.1	100.5	99.8	100.2
B	97	198.9	199.4	197.6	199.5	200.1	200.0
C	93.5	194.0	196.0	191.7	196.4	198.9	199.7
D	105	199.1	200.2	197.4	200.4	200.8	200.0
E	96	199.5	200.6	197.6	200.7	200.7	200.0

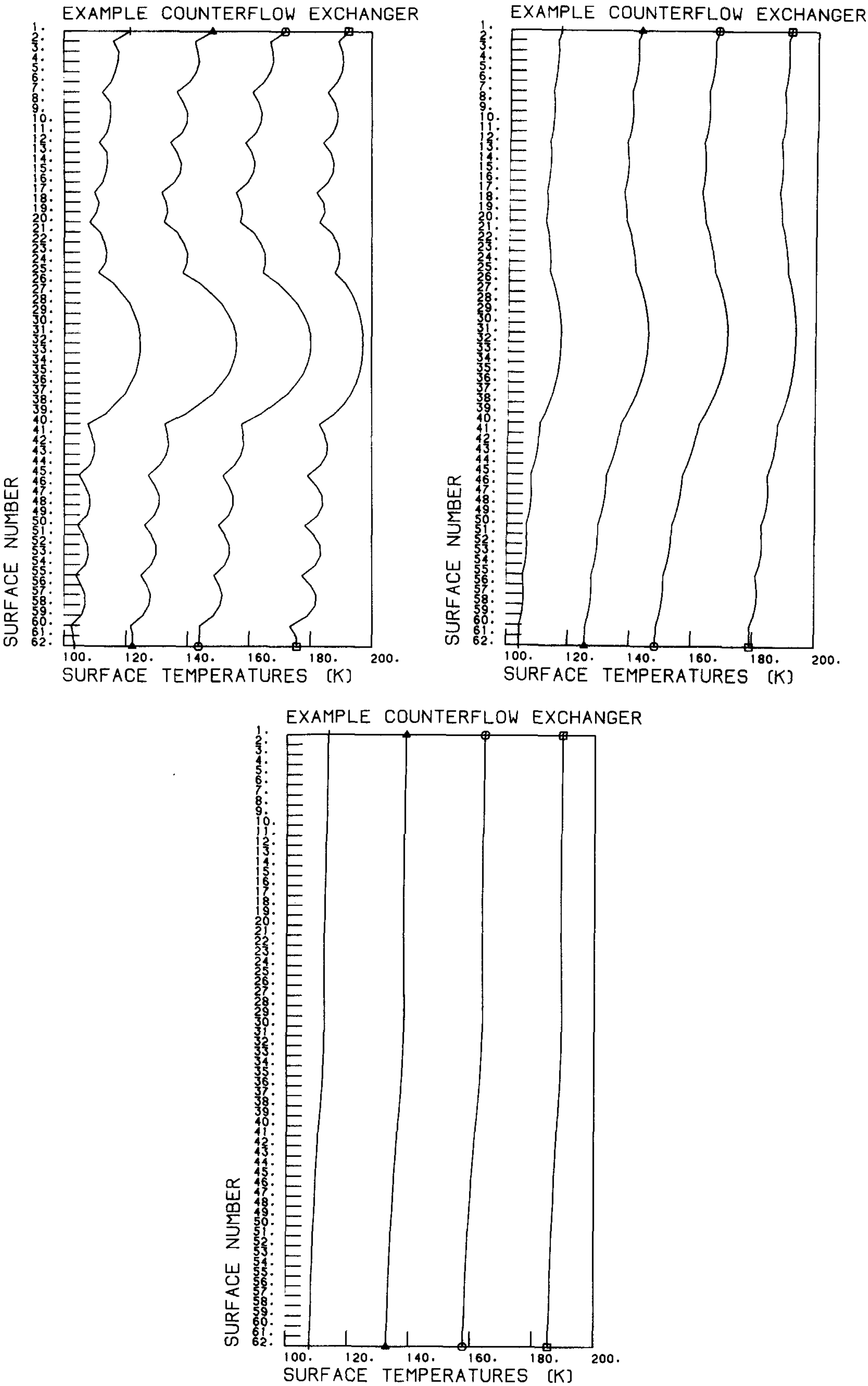

Fig. 4. Surface temperature profiles of example heat exchanger: (a) thermal conductivity of exchanger material 1/10th normal value; (b) thermal conductivity of exchanger material normal; (c) thermal conductivity of exchanger material 10 times normal value.

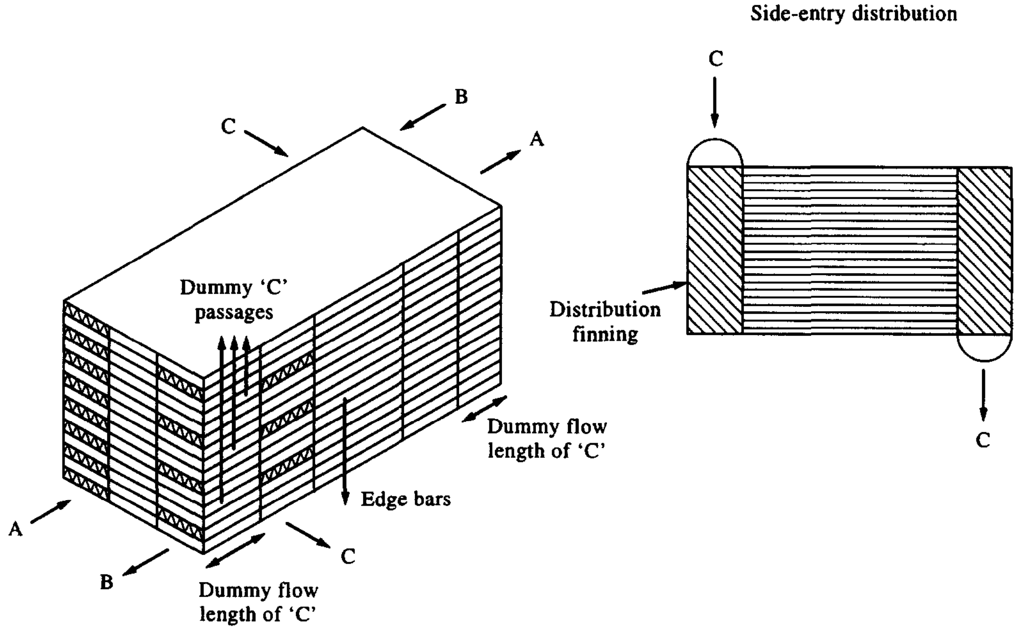

Fig. 5. Side-entry streams and dummy passages in multi-stream plate-fin heat exchangers.

half-fin-length formalism, and the constant surface temperature formalism (obtained by the computer code PERF, which employs this formalism [15]) are also given in Table 2. Note that the performance predicted by the constant surface temperature formalism is very close to that of the high thermal conductivity model in the previous column, as can be expected. The slightly lower performance in the minor 'D' and 'E' streams is offset by higher performance in the major stream 'C' (the constant surface temperature formalism has no sensitivity to stacking pattern [15, 16]).

The excellent agreement between the results predicted by the half-fin-length formalism (with transverse conduction, column 2) and the present formalism (column 4) shows that in most practical situations the former is fairly reliable. However, it can fail in circumstances where passage-wise heat transfer rates vary by one or more orders of magnitude across the block, as will be shown in the ensuing discussion.

One typical situation in which the effect of transverse conduction is particularly important occurs when side-entry streams are present in the heat exchanger, as shown in Fig. 5. When a large number of streams are present in the heat exchanger, some of them have to enter and leave from the sides, rather than from the ends. Also, often process conditions dictate intermediate entry and exit locations for some streams. Side entry streams have a lower flow length than others. However, the passages which they use have to continue to the respective ends of the core with no flow, for structural reasons. When a number of such streams are present, passages of the end-entry streams become interspersed with several 'dummy' passages towards the ends of the exchanger. The dummy passages serve as conduits for heat transfer between widely separated passages of other streams by transverse conduction. The half-fin-length model [16] would be unable to deal with such situations. It would predict a zero transfer if the transverse conduction were neglected. If the latter were considered, the linear conduction model employed in ref. [16] would be too inaccurate for accounting for heat conduction across many dummy passages. Since the local heat transfer coefficient in the dummy passages would be very low, the fin effectiveness would be very high; this should produce a linear temperature profile in the dummy passages, with Mechanism 2 dominating. It is clear that substantial heat transfer can occur through the fins in the dummy passages by conduction under these circumstance.

To study such a situation, a balanced two-stream exchanger in which a cold stream passage 'B' is separated from two hot stream passages 'A' by eight dummy 'X' passages on either side was analysed with STACK. Figure 6(a) shows the surface temperature profiles of the exchanger predicted by the present formalism, while Fig. 6(b) shows those predicted by the half-fin-length formalism. Note that the temperature profiles of Fig. 6(a) confirm our observation above. Table 3 lists the respective temperatures predicted by STACK using the present and the half-fin-length formalisms. As can be expected, the half-fin-length formalism predicts no heat transfer between 'A' and 'B'.

Even though a number of authors have dealt with the design of multi-stream plate-fin heat exchangers, there are no references in which the performance of a fully specified exchanger is given [18]. For this reason, no comparisons could be made, but experience in rating numerous heat exchangers with STACK and PERF [15] (Table 2) confirms that the present formalism is sufficiently general and realistic for appli-

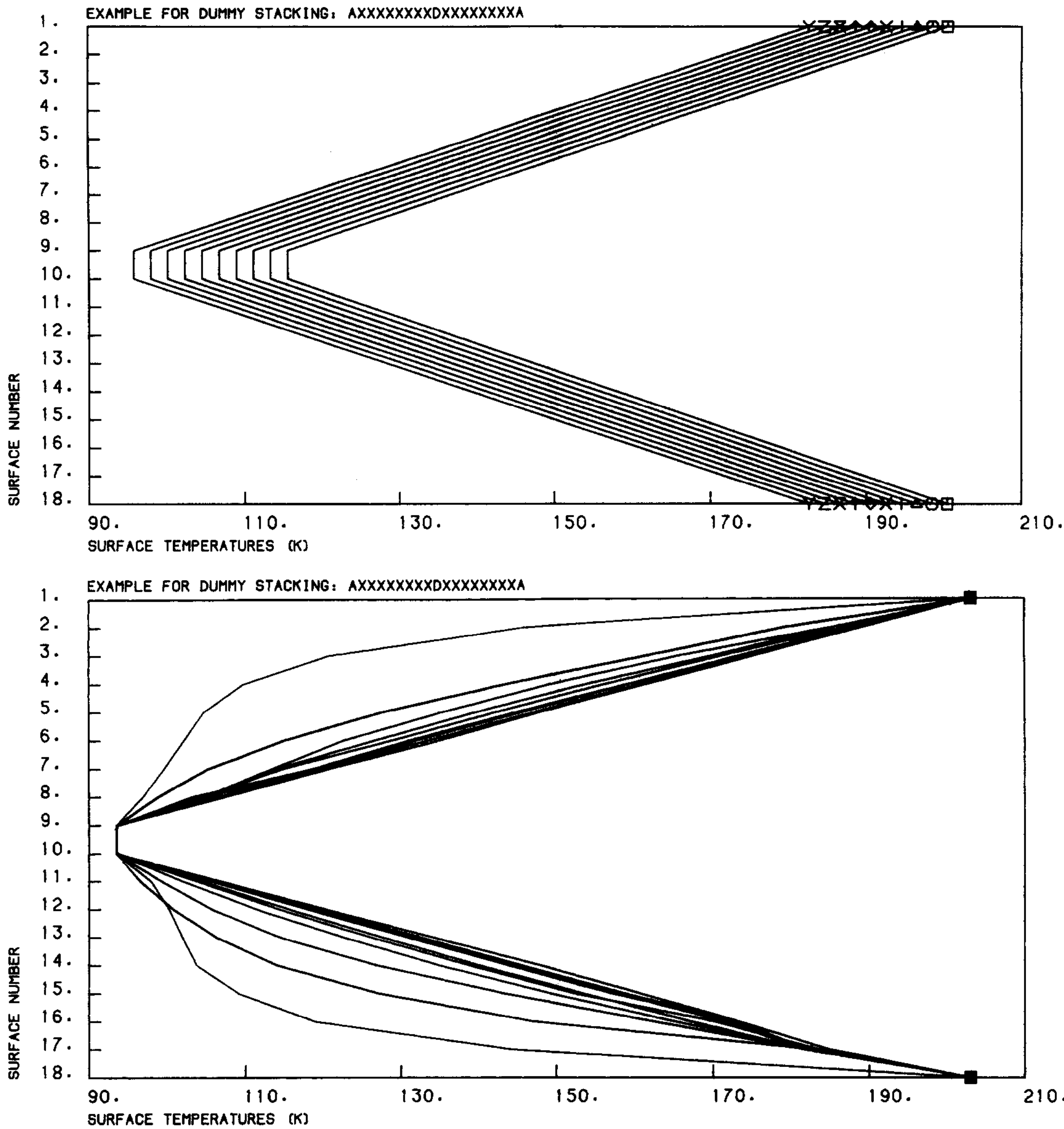

Fig. 6. Surface temperature profiles in dummy passages predicted with: (a) multi-passage formalism; (b) half-fin-length formalism with superimposed linear conduction.

Table 3. Heat transfer by transverse conduction in dummy passages

Stream	Present formalism		Half-fin-length formalism without transverse conduction	
	Entry temp. (K)	Exit temp. (K)	Entry temp. (K)	Exit temp. (K)
A	203	183.4	203	203
B	93.5	115.5	93.5	93.5

cation in the design of multi-stream plate-fin heat exchangers. The incorporation of secondary effects such as: (i) flow maldistribution; (ii) longitudinal heat conduction; and (iii) heat exchange with abient in the present formalism, and their effects on thermal performance, will be discussed in a subsequent paper.

5. CONCLUSIONS

A computer program of rating multi-stream plate-fin heat exchangers based on a formalism proposed earlier [17] was developed. The program was used to analyse several existing heat exchangers. The results confirmed that normally transverse conduction is present only in some passages of multi-stream plate-fin heat exchangers. However, it became increasingly important as the fin effectiveness increased. The program also predicted the effects of certain special situations occurring in multi-stream plate-fin heat exchangers correctly. The computer program was found to be robust and useful in the design of multi-stream plate-fin heat exchangers.

Multi-stream heat exchangers are often more economical to design, build and operate, than two-stream heat exchangers; their use often results in better integrated, more efficient, and economic process design. Increasing energy and environmental concerns mean the increased use of multi-stream designs in the future. The development of new design theory and associated computer models for multi-stream plate-fin exchangers is important in the overall context of efforts to develop formalisms for general multi-stream heat exchangers [6, 22]. It is hoped that the material presented here will be of use in generating models of a more generally applicable nature.

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