Postprint: Numerical Study on Three-Stream Plate-Fin Heat Exchanger under Flow Maldistribution

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Abstract

A general heat transfer mathematical model for multi-stream plate-fin heat exchangers under flow maldistribution was established, and numerical calculation of plate-fin heat exchangers was implemented through a combination of FLUENT simulation and self-developed programming. Taking an airborne cross-flow three-stream plate-fin heat exchanger as an example, the rationality of the numerical method was validated by comparison with experimental data from literature, and the influence of flow maldistribution on the heat transfer performance of multi-stream plate-fin heat exchangers was analyzed under conditions of variable fluid specific heat capacity, fluid inlet temperature, and fluid flow resistance.

Full Text

Numerical Study of Three-Stream Plate-Fin Heat Exchanger under Flow Maldistribution Conditions

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Abstract: A general heat transfer mathematical model for multi-stream plate-fin heat exchangers (MPFHE) under flow maldistribution conditions was established, and numerical calculations were implemented through a combination of

FLUENT simulation and independent programming. Using an airborne crossflow three-stream plate-fin heat exchanger as an example, the rationality of the numerical method was validated by comparison with experimental data from literature. The effects of flow maldistribution on the heat transfer performance of MPFHE were analyzed under varying conditions of fluid specific heat capacity, fluid inlet temperature, and fluid flow resistance.

Keywords: three-stream plate-fin heat exchanger; flow maldistribution; numerical simulation
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Multi-stream plate-fin heat exchangers integrate hot and cold fluid resources, offering advantages such as high system integration, convenient management, reduced investment costs, and compact footprint, making them highly promising for aerospace applications. In the design of plate-fin heat exchangers, fluid flow is typically assumed to be uniformly distributed. However, due to the influence of header and distributor structures, uniform flow distribution represents only an ideal operating condition. This effect becomes particularly severe when the heat exchanger has a large number of transfer units, making flow maldistribution a non-negligible factor that has attracted extensive research attention from scholars both domestically and internationally.

Internationally, Ranganayakulu [1,2] employed the finite element method to investigate the combined effects of longitudinal heat conduction, inlet temperature nonuniformity, and flow maldistribution on the thermal performance of crossflow heat exchangers. Lalot [3] studied the impact of flow maldistribution on heat exchanger performance through electrically heated experiments, revealing that flow nonuniformity could cause up to 25% degradation in crossflow heat exchanger performance. Mishra [4] utilized the finite difference method to examine the effects of dynamic flow maldistribution on crossflow two-stream heat exchangers using step, exponential, and sinusoidal distribution models. Yuan [5] investigated the relationship between flow maldistribution characteristics and thermal performance in three-stream heat exchangers based on four flow distribution models, concluding that flow maldistribution could potentially enhance heat exchanger performance through comparative analysis of temperature fields and effectiveness.

Domestically, the heat transfer and enhancement research group at Xi' an Jiaotong University has conducted systematic studies on flow maldistribution caused by header and distributor structures. Wen Jian [6] used CFD software to numerically simulate flow maldistribution characteristics in industrial basic headers, inline perforated plate headers, and staggered perforated plate headers, demonstrating that header structure optimization could effectively reduce the maximum velocity ratio and nonuniformity parameters. Subsequent experimental validation [7] confirmed these numerical results, and similar experimental methods were applied to investigate two-phase flow distribution characteristics under different header configurations [8], revealing that two-phase flow distribution patterns are more complex and nonuniform than single-phase flow. Jiao Anjun [9] improved the basic header structure by introducing a secondary header design and experimentally studied how the diameter ratios of primary and secondary header inlet/outlet pipes affect flow maldistribution, concluding that flow distribution becomes more uniform when the diameter ratios are equal. Additionally, Jiao Anjun [10,11] conducted in-depth research on the effects of distributor structural parameters and guide angles on distribution performance and heat transfer characteristics. Zhang Zhe [12] performed CFD simulations comparing basic headers and improved secondary headers, demonstrating that the modified header structure significantly improved flow uniformity, with both flow uniformity parameters and the ratio of maximum to minimum velocities being reduced. Zhang Zhe [13,14] subsequently verified these conclusions experimentally.

Currently, research on the effects of flow maldistribution on multi-stream heat exchanger performance remains limited. Although similarities exist with two-stream heat exchangers, the diverse flow arrangements and temperature coupling among fluids in multi-stream configurations inevitably lead to new manifestations of heat transfer characteristics under maldistribution conditions. This study proposes a numerical calculation method for multi-stream plate-fin heat exchangers under flow maldistribution by combining FLUENT simulation with independently developed programs. The method is validated against experimental data, and its application demonstrates the influence of flow maldistribution on heat exchanger performance.

1 Numerical Calculation Method for MPFHE

1.1 Mathematical Model

For multi-stream plate-fin heat exchangers (Fig. 1 [Figure 1: see original paper]), the following assumptions are made for modeling convenience: fluid temperature is uniform along the channel height direction; fin and parting sheet temperatures are uniform through their thickness; perfect contact exists between fins and parting sheets; parting sheet temperature equals fin root temperature; lateral conduction within the same channel is neglected; and lateral conduction through fins and parting sheets is ignored.

Based on these assumptions, energy conservation equations are established for fins, parting sheets, and fluids. For single-phase fluids:

$$\frac{d}{dl}\left[G_i c_{p,i} T_i(l)\right] = \alpha_i H_i f_i \left[t_i(l) - T_i(l)\right] + \alpha_i H_i f_i \left[t_{i-1}(l) - T_i(l)\right]$$

where l and n represent Cartesian coordinates along the fluid flow direction and transverse direction (m), respectively; subscript i denotes channel number ($i = 1, 2, \ldots, N$); G represents mass velocity (kg/(m·s)); c_p represents specific heat at constant pressure (J/(kg·K)); H represents fin height (m); f represents fin density (1/m); δ represents fin thickness (m); T represents fluid temperature (K); t represents fin temperature (K); λ represents fin thermal conductivity (W/(m·K)); and α represents convective heat transfer coefficient (W/(m²·K)).

In the equation, F represents the relative flow direction between fluids (F equals 1 or -1). The first term represents fluid energy increment along the flow direction, the second term represents heat flow between fluid and upper/lower parting sheets, and the third term represents the difference in heat flow at fin roots, which equals the heat flow between fluid and fins.

The energy conservation equation for parting sheets is:

$$\alpha_i H_i f_i \left[t_i(l) - T_i(l)\right] + \alpha_i H_i f_i \left[t_i(l) - T_{i+1}(l)\right] = \alpha_{i+1} H_{i+1} f_{i+1} \left[t_i(l) - T_{i+1}(l)\right] + \alpha_{i+1} H_{i+1} f_{i+1} \left[t_i(l) - T_{i+2}(l)\right]$$

where x represents Cartesian coordinate along fin height direction (m) and subscript i denotes parting sheet number ($i = 0, 1, 2, \dots, N$). The first and third terms represent heat flow between fins and fluid in channel i and the upper surface of parting sheet i, while the second and fourth terms represent heat flow between fins and fluid in channel $(i+1)$ and the lower surface of parting sheet i. Based on the assumptions, the upper and lower surfaces of the top and bottom parting sheets ($i = N$ and $i = 0$) are adiabatic, making the corresponding terms zero.

Additionally, according to the assumptions, temperatures at upper and lower surfaces of parting sheets are equal:

$$t_i(l) = t_{i+1}(l) \quad (i = 0, 1, \dots, N-1)$$

The energy conservation equation for fins is:

$$\frac{d^2 t_i(x)}{dx^2} = \frac{2\alpha_i}{\lambda_i \delta_i}\left[t_i(x) - T_i(l)\right]$$

The general solution can be expressed as:

$$t_i(x) = A_i \sinh(m_i x) + B_i \cosh(m_i x) + T_i(l)$$

where $m_i = \sqrt{2\alpha_i/(\lambda_i \delta_i)}$.

Substituting this solution into equations (1)-(3) yields a system of $3N$ equations for fluid temperatures $T_i(l)$ and constants A_i and B_i.

1.2 Numerical Discretization Method

The heat exchanger is divided into $W \times L$ sub-unit heat exchangers as shown in Fig. 2 [Figure 2: see original paper]. For the first-order derivative term of $T_i(l)$ in equation (1), first-order central difference discretization is applied:

$$\left.\frac{dT_i}{dl}\right|_{j,k} \approx \frac{T_i(j,k+1) - T_i(j,k-1)}{2\Delta l}$$

For the convective heat transfer qualitative temperature between fluid and parting sheets/fins, the arithmetic mean of inlet and outlet temperatures of the sub-unit heat exchanger is used:

$$T_{ref} = \frac{T_i(j,k) + T_i(j,k+1)}{2}$$

1.3 Fin Database

For accurate numerical calculation of multi-stream plate-fin heat exchangers, reasonable numerical methods are crucial, but accurate calculation of convective heat transfer coefficients for fins is also important. The heat transfer characteristics of fin structures are commonly characterized by the j-factor. For the serrated fins discussed below, the Weiting correlation [15] is adopted:

$$j = \begin{cases} 0.483 \cdot Re^{-0.462} & (Re \leq 1000) \\ 0.242 \cdot Re^{-0.462} & (Re > 1000) \end{cases}$$

where Re is Reynolds number. The j-factor in the transition region is determined by the intersection point of the j-Re curves. If the actual Re is less than the intersection Re, the laminar region correlation is used; otherwise, the turbulent region correlation is applied.

1.4 Flow Distribution Model Establishment Method

Plate-fin heat exchanger structures are complex (Fig. 3 [Figure 3: see original paper]), with inlet structures consisting primarily of headers and distributors. Simultaneously considering both components in CFD simulations requires large mesh counts. Therefore, simplified models are necessary when studying their flow distribution characteristics. The assumptions are: constant fluid properties and negligible influence of temperature field on velocity field; separate analysis of header and distributor flow distribution characteristics; and replacement of the core and distributor with porous media materials having equivalent flow resistance characteristics when analyzing header effects.

Based on these assumptions, FLUENT simulations can obtain flow distribution characteristics for specific header and distributor structures. The implementation method is:

1. **Single-channel model:** Based on average mass flow rate, simulate flow distribution and flow resistance characteristics in a single channel. The velocity field exhibits two-dimensional distribution, which can be simplified to a one-dimensional flow distribution model using weighted averaging. The channel is divided into inlet, middle, and outlet regions along the flow direction, and into W sub-units along the transverse direction. The mass flow fraction for any sub-unit k is:

$$\psi_k = w_{in}\frac{G_{k,in}}{G_{channel}} + w_{middle}\frac{G_{k,middle}}{G_{channel}} + w_{out}\frac{G_{k,out}}{G_{channel}}$$

where G_k represents average mass flow rate of sub-unit k, $G_{channel}$ represents total channel mass flow rate, and w_{in}, w_{middle}, w_{out} represent weighting factors for inlet, middle, and outlet regions, respectively. For the example case, considering the middle region as the main flow area with large temperature differences in the inlet region enabling sufficient heat transfer, the weights are assigned as 0.3, 0.6, and 0.1, respectively.

2. **Header structure model:** Based on single-channel flow resistance characteristics, the middle channel in the header model is set as porous media with adjusted resistance parameters to match the single-channel resistance, thereby obtaining flow distribution characteristics among different channels.

3. **Two-dimensional flow distribution model:** Multiplying each channel flow rate by ψ_k yields the two-dimensional flow distribution model $G(i, j, k)$ for a given fluid.

1.5 Numerical Calculation Flow

Based on the above analysis, the crossflow multi-stream plate-fin heat exchanger is divided into $W \times L$ sub-unit heat exchangers (dashed lines in Figs. 1 and 2). The numerical calculation procedure (Fig. 4 [Figure 4: see original paper]) is: 1) Obtain two-dimensional flow distribution models for each fluid using FLUENT based on header and distributor configurations (Section 1.4); 2) Assuming uniform flow distribution within each sub-unit, solve for outlet temperatures $T_i(j, k)$ of each channel by establishing and solving linear equation systems using the methods described in Sections 1.1 and 1.2, with fluid properties determined from inlet conditions and convective coefficients α calculated from equation (8); 3) Obtain the complete temperature field distribution through row-by-row or column-by-column scanning, recalculating α for each sub-unit based on actual flow rates.

2 Example Analysis

Using the airborne three-stream heat exchanger from reference [16] as an example, the core dimensions are 400 mm length and 130 mm width, with 0.8 mm parting sheet thickness and channel arrangement (ABACABAC···). Fluid A (air) flows as cold fluid along the width direction, while fluids B and C (air) flow as hot fluids along the length direction in parallel. All three fluids use serrated fins with 3 mm uninterrupted length. Structural parameters are listed in Table 1 .

2.1 Hot Fluid Flow Distribution Model

For the described three-stream plate-fin heat exchanger (Fig. 3 [Figure 3: see original paper]), cold fluid A enters and exits through expanding and contracting tubes, allowing uniform flow distribution to be assumed. Hot fluids B and C, however, experience nonuniform intra-channel and inter-channel flow distribution due to header and distributor structures.

For hot fluid B at a total mass flow rate of 330 kg/h, the flow distribution model obtained using the method in Section 1.3 is shown in Fig. 5 [Figure 5: see original paper]. The FLUENT simulation details are conventional and omitted here for brevity. The flow distribution model for hot fluid C at 330 kg/h is axially symmetric to fluid B.

2.2 Numerical Method Experimental Validation

Reference [16] validated the numerical method through experiments on a thermal test bench, demonstrating that flow maldistribution significantly affects numerical accuracy. After introducing the nonuniform flow distribution model, Fig. 6 compares numerical and experimental outlet temperatures for hot fluids B and C under varying fluid A flow rates. Fig. 7 [Figure 7: see original paper] shows relative calculation errors for fluids B and C under uniform and nonuniform distribution models.

The results demonstrate that despite the simplified flow maldistribution model, numerical accuracy is effectively improved. The correction effect becomes more pronounced at lower cold fluid A mass flow rates: at 800 kg/h of fluid A, the error for fluid C is reduced by 3.9%. Higher heat capacity flow rates of cold fluid A diminish the impact of flow maldistribution on hot fluid heat transfer performance.

2.3 Effect of Flow Maldistribution on Heat Transfer Performance

For the three-stream heat exchanger, fluid A is assumed as cold fluid with inlet conditions of 1400 kg/h and 30°C, while fluids B and C are hot fluids at 330 kg/h each with inlet temperatures of 90°C and 130°C, respectively. The relative difference in heat duty between uniform and nonuniform flow distribution is defined as:

$$\Delta q_i = \frac{q_{i,even} - q_{i,uneven}}{q_{i,even}} \times 100\%$$

where i denotes fluid type, and $q_{i,uneven}$ and $q_{i,even}$ represent heat duties under nonuniform and uniform flow distribution, respectively.

The effects are analyzed from three perspectives: variable specific heat capacity, variable inlet temperature, and variable flow resistance.

1) Variable Specific Heat Capacity

Varying fluid B' s specific heat capacity while keeping other parameters constant yields the relative heat duty differences shown in Fig. 8 [Figure 8: see original paper]. As fluid B' s specific heat capacity increases, flow maldistribution causes increased heat transfer for fluid B and decreased heat transfer for fluid C, with greater impact on the fluid having lower heat capacity flow. The maximum absolute value of Δq_B reaches 5%. Cold fluid A, being the sole cooling source, represents total heat exchanger duty, with $|\Delta q_A| \leq 0.5\%$. As fluid B' s heat capacity flow becomes sufficiently large, Δq_A approaches zero.

Varying cold fluid A' s specific heat capacity while keeping hot fluids B and C identical yields the results in Fig. 9 [Figure 9: see original paper]. Lower heat capacity flow of cold fluid A results in larger absolute values of Δq_B and Δq_C. Flow maldistribution increases heat transfer by up to 7.5% for fluid B near the cold fluid inlet while decreasing fluid C' s heat transfer by 4.2%, with overall heat exchanger performance slightly degraded. As cold fluid A' s heat capacity flow increases, its temperature gradient decreases, enabling more uniform heat transfer to both fluids B and C, thereby reducing the impact of flow maldistribution.

2) Variable Fluid Inlet Temperature

Varying hot fluid C's inlet temperature from 70°C to 130°C produces the relative heat duty differences shown in Fig. 10 [Figure 10: see original paper]. When $(T_{in,C} - T_{in,A})/(T_{in,B} - T_{in,A}) < 1$, fluid B has higher inlet temperature than fluid C. Since fluid C' s high-flow region is near cold fluid A' s outlet, lower fluid C temperature experiences greater impact from flow maldistribution, with heat transfer reduction up to 4.8%. As fluid C temperature increases, larger temperature differences lead to more sufficient heat transfer, diminishing the effect of flow maldistribution.

3) Variable Fluid Flow Resistance

Varying hot fluid pressure drop to obtain different flow distribution models yields the relative heat duty differences shown in Fig. 11 [Figure 11: see original paper]. At low channel pressure drop, Δq_A, Δq_B, and Δq_C are all negative, indicating degraded heat transfer performance for all fluids. As channel pressure drop increases, flow distribution among channels becomes more uniform, improving performance for all fluids. At infinitely large pressure drop, inter-channel flow distribution becomes uniform, but distributor structures still cause transverse maldistribution, resulting in increased heat transfer for fluid B near the cold fluid inlet and decreased heat transfer for fluid C, demonstrating that distributor configuration affects energy distribution among hot fluids.

This study developed a numerical calculation method for multi-stream plate-fin heat exchangers under flow maldistribution by combining CFD with independent programming. Validation against experimental data confirmed the method's rationality. Case studies analyzing variable specific heat capacity, inlet temperature, and flow resistance yielded the following conclusions:

1. Comparison with experimental data demonstrates that the proposed nonuniform flow distribution model effectively improves numerical accuracy, particularly at lower cold fluid mass flow rates where correction effects are more significant.

2. For the three-stream heat exchanger case, under a fixed flow distribution pattern, varying hot fluid heat capacity flow ratios shows that maldistribution significantly affects the fluid with lower heat capacity flow, reducing its heat transfer by up to 5%. Varying hot-to-cold fluid heat capacity flow ratios reveals clear energy separation: heat transfer for the hot fluid near the cold fluid inlet can increase by 7.5% while the other hot fluid decreases by 4.2%, with more pronounced effects at lower cold fluid heat capacity flow. Varying hot fluid inlet temperature shows that the lower-temperature hot fluid experiences greater impact, with heat transfer reduction up to 4.8%. Varying flow resistance demonstrates that lower channel pressure drop leads to more severe inter-channel maldistribution and degraded performance for all fluids.

References

[1] Ranganayakulu C, Seetharamu K N. The Combined Effects of Longitudinal Heat Conduction, Flow Nonuniformity and Temperature Nonuniformity in Crossflow Plate-Fin Heat Exchangers [J]. International Journal of Heat and Mass Transfer, 1997, 40(1): 27-38.

[2] Ranganayakulu C, Seetharamu K N, Sreevatsan K V. The Effects of Inlet Fluid Flow Nonuniformity on Thermal Performance and Pressure Drops in Crossflow Plate-Fin Heat Exchangers [J]. International Communication in Heat and Mass Transfer, 1999, 26(4): 545-556.

[3] Lalot S, Florent P, Lang S K, Bergles A E, et al. Flow Maldistribution in Heat Exchangers [J]. Applied Thermal Engineering, 1999, 19(8): 847-863.

[4] Mishra M, Das P K, Sarangi S. Effect of Temperature and Flow Nonuniformity on Transient Behaviour of Crossflow Heat Exchanger [J]. International Journal of Heat and Mass Transfer, 2008, 51(9): 2583-2592.

[5] Yuan P. Effect of Inlet Flow Maldistribution on the Thermal Performance of a Three-fluid Crossflow Heat Exchange [J]. International Journal of Heat and Mass Transfer, 2003, 46(20): 3777-3787.

[6] Wen Jian, Li Yanzhong. Study of Flow Distribution and its Improvement on the Header of Plate-fin Heat Exchanger [J]. Cryogenics, 2004, 44(11): 823-831.

[7] Wen Jian. The Investigation of Flow Field Distribution in Plate-Fin Heat Exchangers Based on PIV Technology [D]. Xi' an: Xi' an Jiaotong University, 2006.

[8] Wen Jian, Wang Simin, Li Yanzhong. Two-Phase Flow Distribution in Plate-Fin Heat Exchanger [J]. Chemical Engineering, 2010, 38(12): 26-29.

[9] Jiao Anjun, Zhang Rui, Jeong S. Experimental Investigation of Header Configuration on Flow Maldistribution in Plate-Fin Heat Exchanger [J]. Applied Thermal Engineering, 2003, 23(10): 1235-1246.

[10] Jiao Anjun, Li Yanzhong, Zhang Rui, et al. Flow Distribution Performance of Different Distributor's Configuration in Plate-Fin Heat Exchanger [J]. Journal of Xi' an Jiaotong University, 2001, 35(11): 1113-1117.

[11] Jiao Anjun, Li Yanzhong, Zhang Rui, et al. Effects of Different Distributor Configuration Parameters on Fluid Flow Distribution in Plate-Fin Heat Exchanger [J]. Journal of Chemical Industry and Engineering, 2003, 54(2): 153-158.

[12] Zhang Zhe, Tian Jinjin, Guo Yonggang, et al. CFD Simulation on Flow Distribution in Plate-Fin Heat Exchangers [J]. Advanced Materials Research, 2013, 665-657: 445-448.

[13] Zhang Zhe, Mehendale S, Tian Jinjin, et al. Experimental Investigation of Distributor Configuration on Flow Maldistribution in Plate-Fin Heat Exchangers [J]. Applied Thermal Engineering, 2015, 85(22): 111-123.

[14] Zhang Zhe, Tian Jinjin, Guo Yonggang, et al. Effects of Header Configurations on Flow Distribution and Heat Transfer in Plate-Fin Heat Exchangers [J]. Chemical Engineering, 2013, 41(9): 40-44.

[15] Weiting A K. Empirical Correlations for Heat Transfer and Flow Friction Characteristics of Rectangular Offset-Fin Plate-Fin Heat Exchangers [J]. ASME Journal of Heat Transfer, 1975, 97: 488-490.

[16] Li Jun, Jiang Yanlong, Zhou Nianyong, et al. Numerical Study of Multi-Stream Plate-fin Heat Exchanger with Cross Type [J]. Journal of Aerospace Power, 2016, 31(5): 1087-1096.

Note: Figure translations are in progress. See original paper for figures.