Numerical Study of a Three-Stream Plate-Fin Heat Exchanger Under Flow Maldistribution: Postprint

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Abstract

A general heat transfer mathematical model for multi-stream plate-fin heat exchangers under non-uniform flow distribution was established, and numerical computation of plate-fin heat exchangers was implemented through a combined approach of FLUENT simulation and in-house programming. Using an airborne crossflow three-stream plate-fin heat exchanger as a case study, the validity of the numerical method was verified by comparison with experimental data from the literature, and the effects of flow maldistribution on the heat transfer performance of multi-stream plate-fin heat exchangers were analyzed under varying conditions of 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: This study establishes a generalized heat transfer mathematical model for multi-stream plate-fin heat exchangers (MPFHE) under flow maldistribution conditions and implements numerical calculations through integrated FLUENT simulations and independent programming. Using an airborne crossflow three-stream plate-fin heat exchanger as a case study, the validity of the numerical method is demonstrated through comparison with experimental data from the literature. The effects of flow maldistribution on MPFHE thermal performance are analyzed under varying fluid specific heat capacities, inlet temperatures, and flow resistance conditions.

Keywords: three-stream plate-fin heat exchanger; flow maldistribution; numerical simulation

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Multi-stream plate-fin heat exchangers offer significant advantages in aerospace applications through integrated thermal management of multiple hot and cold fluids, including high system integration, simplified management, reduced investment costs, and compact footprint. However, conventional design practices typically assume uniform fluid flow distribution, which represents an idealized condition. In reality, flow maldistribution inevitably occurs due to header and distributor structural effects, particularly for heat exchangers with large heat transfer unit numbers (NTU), where these effects become more pronounced. Consequently, flow maldistribution represents a critical issue that cannot be ignored and has been extensively investigated by researchers worldwide.

International studies have systematically examined this phenomenon. Ranganayakulu [1,2] employed finite element methods to investigate the combined effects of longitudinal heat conduction, inlet temperature nonuniformity, and flow maldistribution on crossflow heat exchanger performance. Lalot [3] experimentally studied flow maldistribution using electrically heated test sections, revealing heat transfer performance degradation up to 25% in crossflow exchangers due to non-uniform fluid distribution. Mishra [4] utilized finite difference methods with step, exponential, and sinusoidal distribution models to analyze the dynamic effects of flow maldistribution on crossflow two-stream heat exchangers. Yuan [5] investigated the relationship between flow maldistribution characteristics and thermal performance in three-stream heat exchangers based on four distribution models, suggesting that flow maldistribution might actually enhance exchanger performance under certain conditions.

Domestic research has primarily focused on header and distributor structural optimization. The heat transfer enhancement research group at Xi’ an Jiaotong University conducted comprehensive studies on flow maldistribution induced by header and distributor geometries. Wen [6] numerically simulated flow distribution characteristics in industrial basic headers and perforated plate configurations using CFD, demonstrating that header structural improvements effectively reduce maximum velocity ratios and nonuniformity parameters, with subsequent experimental validation [7]. Similar experimental methods were applied to investigate two-phase flow distribution characteristics [8], revealing more complex and non-uniform patterns compared to single-phase flow. Jiao [9] improved basic header designs through secondary header modifications, experimentally investigating the effects of inlet/outlet diameter ratios on flow uniformity, concluding that equal diameter ratios between primary and secondary headers yield more uniform distribution. Further studies by Jiao [10,11] examined distributor structural parameters and deflection angles on flow distribution and heat transfer performance. Zhang [12] performed CFD simulations comparing basic and improved secondary header configurations, demonstrating significantly enhanced flow uniformity and reduced velocity ratios, with experimental verification by Zhang [13,14].

Despite these advances, research on flow maldistribution effects in multi-stream plate-fin heat exchangers remains limited. While similarities exist with two-stream exchangers, multi-stream configurations exhibit diverse flow arrangements and temperature coupling among fluids, resulting in novel manifestations of heat transfer characteristics under maldistribution conditions. This study develops a numerical methodology combining FLUENT simulations with independently programmed algorithms to analyze flow maldistribution effects on thermal performance in multi-stream plate-fin heat exchangers.

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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 adopted to facilitate modeling: fluid temperature is uniform along the channel height direction; fin and plate temperatures are uniform through their thickness; perfect contact exists between fins and plates; plate temperature equals fin root temperature; lateral conduction within the same fluid channel is neglected; and transverse conduction in fins and plates is ignored.

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

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

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

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

The plate energy conservation equation is:

$$\alpha_i f_i H_i \left[t_i(l) - T_i(l)\right] + \lambda_i f_i \delta_i \frac{d^2 t_i(l)}{dn^2} = \alpha_{i+1} f_{i+1} H_{i+1} \left[t_{i+1}(l) - T_{i+1}(l)\right] + \lambda_{i+1} f_{i+1} \delta_{i+1} \frac{d^2 t_{i+1}(l)}{dn^2}$$

where subscript i indicates plate number ($i = 0, 1, 2, \ldots, N$). The first and third terms represent heat flow between plate upper surface and channel i fins/fluid, while the second and fourth terms represent heat flow between plate lower surface and channel ($i+1$) fins/fluid. Based on the adiabatic boundary condition at top and bottom plates, corresponding terms become zero for $i = N$ and $i = 0$.

Additionally, plate surface temperatures are assumed equal at both ends:

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

The fin energy conservation equation is:

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

where x is the Cartesian coordinate along fin height (m). The general solution takes the form:

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

where $m_i = \sqrt{2\alpha_i/(\lambda_i \delta_i)}$, and A_i, B_i are constants determined by boundary conditions.

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 Eq. (1), first-order central difference discretization is applied:

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

where j and k denote indices along flow and transverse directions, respectively.

For convective heat transfer between fluid and plates/fins, the arithmetic mean of inlet and outlet temperatures in each sub-unit is used as the reference temperature:

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

1.3 Fin Database Accurate numerical calculation of multi-stream plate-fin heat exchangers requires not only appropriate numerical methods but also reliable convective heat transfer coefficient correlations. Fin heat transfer characteristics are typically characterized by the j-factor. For the serrated fins considered herein, the Weiting correlation [15] is employed:

$$j = \begin{cases} 0.483Re^{-0.462} & \text{for } Re \leq 1000 \\ 0.242Re^{-0.538} & \text{for } Re > 1000 \end{cases}$$

where a represents uninterrupted flow length of serrated fins (m), d_e is equivalent diameter (m), s_{fin} is fin spacing (m), H_{fin} is fin height (m), δ_{fin} is fin thickness (m), and Re is Reynolds number. The transition region j-factor is determined by the intersection point of j-Re curves; laminar formulas apply below this Re, while turbulent formulas apply above.

1.4 Flow Distribution Model Establishment Plate-fin heat exchanger structures are complex (Fig. 3 [Figure 3: see original paper]), with inlet configurations comprising headers and distributors. Simultaneous CFD analysis of both components requires excessive mesh counts. Therefore, simplified models are necessary. The assumptions include: constant fluid properties with negligible temperature field effects on velocity field; separate analysis of header and distributor flow distribution characteristics; and replacement of core and distributor sections with porous media of equivalent flow resistance when analyzing header effects.

The implementation methodology involves:

1. **Single-channel model**: Based on average mass flow rate, simulate flow distribution and pressure drop characteristics in a single channel. The 2D velocity field is simplified to a 1D distribution using weighted averaging. The channel is divided into inlet, middle, and outlet regions along the flow direction, and into W sub-units in the transverse direction. The mass flow fraction for 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 $G_{channel}$ is total channel mass flow, $G_{k,in}$, $G_{k,middle}$, $G_{k,out}$ are mass flows in sub-unit k for respective regions, and w_{in}, w_{middle}, w_{out} are weighting factors determined from simulation results and flow configuration. For the case study, values of 0.3, 0.6, and 0.1 are assigned considering the middle region as primary flow and greater temperature differences in the inlet region.

2. **Header model**: Based on single-channel flow resistance characteristics, the header model channels are configured as porous media with adjusted

resistance parameters to match single-channel behavior, enabling determination of flow distribution among different channels.

3. **2D distribution model**: Multiplying each channel flow by ψ_k yields the 2D flow distribution model $G(i, j, k)$ for each fluid.

1.5 Numerical Calculation Procedure The crossflow multi-stream plate-fin heat exchanger is divided into $W \times L$ sub-units (Fig. 1, 2). The numerical procedure (Fig. 4 [Figure 4: see original paper]) proceeds as follows:

1. Obtain 2D flow distribution models for each fluid using FLUENT based on header and distributor geometries (Section 1.4).
2. Assuming uniform flow within each sub-unit, solve for outlet temperatures $T_i(j, k)$ by: determining fluid properties from inlet conditions, calculating convective coefficients α using Eq. (8), establishing linear equation systems from Sections 1.1-1.2, and solving for $T_i(j, k)$, A_i, and B_i.
3. Obtain complete temperature field distribution through sequential row/column scanning, recalculating α for each sub-unit based on actual flow rates.

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2 Case Study Analysis

The airborne three-stream heat exchanger from reference [16] is analyzed with core dimensions 400 mm × 130 mm, plate thickness 0.8 mm, and channel arrangement (ABACABAC···). Fluid A (air) flows cold along the width direction, while fluids B and C (air) flow hot co-currently along the length direction. All streams use serrated fins with 3 mm uninterrupted length. Structural parameters are listed in Table 1 .

2.1 Hot Fluid Flow Distribution Model For the three-stream exchanger (Fig. 3), cold fluid A enters and exits through expanding/contracting tubes, assuming uniform distribution. Hot fluids B and C exhibit non-uniform intra-channel and inter-channel distribution due to header and distributor structures.

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

2.2 Numerical Method Validation Reference [16] experimentally validated the numerical method on a thermal test bench, demonstrating significant flow maldistribution effects on calculation accuracy. With the non-uniform distribution model incorporated, 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 uniform vs. non-uniform models.

The results demonstrate that despite model simplification, numerical accuracy improves substantially. The correction effect becomes more pronounced at lower cold fluid A mass flow rates, with fluid C error reduction reaching 3.9% at 800 kg/h. Higher cold fluid heat capacity rates diminish the impact of flow maldistribution on hot fluid heat transfer performance.

2.3 Effects of Flow Maldistribution on Heat Transfer Performance

For the three-stream exchanger with fluid A as cold fluid (1400 kg/h, 30°C inlet) and fluids B, C as hot fluids (330 kg/h each, 30°C and 90-130°C inlet temperatures), the relative heat load difference between uniform and maldistributed flow 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 loads under non-uniform and uniform distribution, respectively.

1) Variable Specific Heat Capacity

Varying fluid B specific heat capacity yields relative heat load differences shown in Fig. 8 [Figure 8: see original paper]. As fluid B specific heat capacity increases, its heat transfer rate rises while fluid C' s decreases, indicating greater maldistribution impact on fluids with lower heat capacity rates, with maximum $|\Delta q|$ reaching 5%. Cold fluid A represents total exchanger heat duty, showing $|\Delta q| \leq 1\%$ that approaches zero as fluid B heat capacity increases sufficiently.

Varying cold fluid A specific heat capacity (with fluids B and C constant) produces the results in Fig. 9 [Figure 9: see original paper]. Lower cold fluid heat capacity rates yield larger $|\Delta q|$ values for hot fluids. Flow maldistribution increases fluid B heat transfer by 7.5% (near cold fluid inlet) while decreasing fluid C by 4.2%, with overall performance slightly degraded. Higher cold fluid heat capacity rates reduce temperature gradients, enabling more uniform heat transfer to both hot fluids and diminishing maldistribution effects.

2) Variable Inlet Temperature

Varying hot fluid C inlet temperature (70-130°C) yields relative heat load differences in Fig. 10 [Figure 10: see original paper]. When $(T_{in,C} - T_{in,A})/(T_{in,B} - T_{in,A}) < 1$, fluid B enters at higher temperature than C, and fluid C' s large-flow region lies near cold fluid A outlet. Lower fluid C temperatures experience greater maldistribution impact, with heat transfer reduction up to 4.8%. As fluid C temperature increases, larger temperature differences promote more complete heat transfer, reducing maldistribution effects.

3) Variable Flow Resistance

Varying hot fluid channel pressure drop produces different distribution models, with relative heat load differences shown in Fig. 11 [Figure 11: see original paper]. At low channel pressure drop, all Δq_i values are negative, indicating performance degradation for all fluids. As pressure drop increases, intra-channel distribution becomes more uniform, improving performance. At infinite pressure drop (uniform inter-channel distribution), distributor structures still cause transverse maldistribution, increasing fluid B heat transfer while decreasing fluid C, demonstrating that distributor geometry affects energy allocation between hot fluids.

This study develops a numerical methodology combining CFD and independent programming for multi-stream plate-fin heat exchangers under flow maldistribution. Validation against experimental data confirms the method’ s validity. Parametric analysis of specific heat capacity, inlet temperature, and flow resistance yields the following conclusions:

1. Comparison with experimental data demonstrates that the proposed non-uniform distribution model significantly improves numerical accuracy, particularly at lower cold fluid mass flow rates.
2. For the three-stream case study, varying hot fluid heat capacity rates shows that maldistribution most affects fluids with lower heat capacity rates, reducing heat transfer by up to 5%. Varying cold/hot fluid heat capacity ratios reveals clear energy separation: heat transfer for hot fluid near the cold fluid inlet increases by 7.5% while the other decreases by 4.2%, with more pronounced effects at lower cold fluid heat capacity rates. Varying hot fluid inlet temperature shows that the lower-temperature hot fluid experiences greater maldistribution impact, with maximum 4.8% reduction. Varying flow resistance demonstrates that lower channel pressure drop exacerbates non-uniformity, degrading all fluid performance, while increased pressure drop improves uniformity and overall performance.

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