International Journal of Heat and Mass Transfer 77 (2014) 1055–1062

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

# Passage arrangement design for multi-stream plate-fin heat exchanger under multiple operating conditions

Xiang Peng, Zhenyu Liu *, Chan Qiu, Jianrong Tan

*Zhejiang University, Hangzhou 310027, China*

ARTICLE INFO

*Article history:*
Received 18 March 2013
Received in revised form 24 January 2014
Accepted 19 June 2014
Available online 5 July 2014

*Keywords:*
Integral-mean temperature differences
Multi-stream
Plate-fin heat exchanger
Passage arrangement

ABSTRACT

As the development of heat exchangers from two streams and single operating condition to multi-stream and multiple operating conditions, passage arrangement becomes a key problem for heat exchanger design. In this paper, a new passage arrangement design method for multiple operating conditions is developed for multi-stream plate-fin heat exchangers. The integral-mean temperature differences method (IMTD) is expanded from three-stream to multi-stream plate-fin heat exchanger, and then the correlation model among heat transfer rate, passage arrangement and fin design parameters is constructed based on IMTD. Upon the maximum heat transfer rate under each operating condition, the passage arrangement coefficient of each condition and the passage arrangement design space of multiple operating conditions are calculated. As passage arrangements being design variables, the passage arrangement under multiple operating conditions is designed using hybrid particle swarm algorithm. Effectiveness of the passage arrangement design approach is verified through optimization design of a 24 streams heat exchanger, and the results show that the heat exchanger obtained by the proposed algorithm exhibit comfort heat transfer rates under multiple operating conditions. Finally, the method is used to design the passage arrangement of main heat exchanger in 80,000 $Nm^3$ air separation unit.


## 1. Introduction

Plate-fin heat exchangers are widely used in aerospace, automobile and cryogenic industries due to their compactness (i.e., high heat transfer surface area-to-volume ratio) for desired thermal performance, resulting in reduced space, weight, support structure, footprint, energy requirement and cost [1]. To meet the variation of these complex equipment, heat exchanger changes from two streams to multiple streams, and the operating condition of heat exchanger usually changes dynamically within a certain range. For example, the oxygen product requirement of air separation equipment changes according to the demand of steel factory, and then changes within 80–110% of standard operation condition, so the demand of heat transfer rate of heat exchanger in air separation equipment changes within a certain range. With the increase in the number of streams, the options of arranging the streams also increase. All passage arrangements do not have the same thermal performances under a certain operating condition, and the thermal performances of a passage arrangement are not the same under different operating conditions. To get the best thermal performance of heat exchanger, some methods have been proposed on establishment of the thermal model and optimization design of passage arrangement of heat exchangers.

Over the decades, the thermal model of two-stream heat exchanger gets standardized. Effectiveness method, number of heat transfer unit method (NTU), temperature effectiveness method [2], log-mean temperature difference method (LMTD) and many other methods are established and used broadly. Unfortunately, there is no universally accepted methodology for the “thermal design” of multi-stream plate-fin heat exchangers to date [3].

As the simplest of multi-stream heat exchangers, the three-stream heat exchanger is analyzed firstly by Morley in 1933 [4]. He derived the differential equation of 3rd order satisfied by the temperature of fluid streams, and obtained the analytical solution through integration. Since then, this research has been studied widely. For example, Bielski and Malinowski [5] used the method of Laplace transforms to solve the set of partial differential equations describing the transient temperature field in a parallel-flow three-fluid heat exchanger. Saeid et al. [6] used finite element method to study the thermal performance of both co-current and counter-current parallel flow heat exchangers. Zhao et al. [7] developed an integral mean temperature difference method (IMTD) for the parallel flow three-fluid heat exchanger with two

* Corresponding author. Tel.: +86 571 87951273; fax: +86 571 87951899.
*E-mail address:* liuzy@zju.edu.cn (Z. Liu).

http://dx.doi.org/10.1016/j.ijheatmasstransfer.2014.06.058
0017-9310/

communications. Their findings showed that the IMTD model had the same accuracy as past exact analytical model, and the model realized a fast convergence for all parallel flow arrangements.

For the multi-stream heat exchanger, Prasad et al. [8,9] formulated the thermal model of multi-stream plate-fin heat exchanger considering unequal temperatures of fin bases. Luo et al. [10] developed a mathematical model for predicting the steady-state thermal performance of single dimensional (concurrent and countercurrent) multi-stream heat exchangers and their networks, and solved it for constant physical properties of streams. Ghosh et al. [11] proposed the successive partitioning and area splitting method for the design of multi-stream plate fin heat exchangers. Though results were high precision, the method was hard to converge because of too much iteration in the calculation. Sun et al. [12] proposed a numerical model to simultaneously predict the fluid flow and heat transfer on both airside and water-side of elliptical finned-tube heat exchangers.

Upon the thermal model of heat exchangers, some researchers have paid their attentions on passage arrangement optimization design and thermal optimization design. Boehme et al. [13] simulated two batteries of multi-stream plate fin heat exchangers from the air separation unit of a steel manufacturer based on the thermal method proposed by Chato [14]. Picon et al. [15] presented a general design methodology for multi-stream plate fin heat exchangers that incorporated the uniform heat load per passage, second surface and fin selection through the manipulation of stream flow passage arrangement. Hassan et al. [16] revealed a relation between Colburn factor and Fanning friction factor for the triangle fin geometry based on CFD analysis coupled with artificial neural network, and then applied NSGA-II to obtain the optimal structural design of the heat exchanger. Ghosh et al. [17] used a genetic algorithm to determine the optimal stacking pattern of multi-stream plate-fin heat exchangers under single operating condition. Joda et al. [18] used genetic algorithm to reduce total annual cost of the heat exchanger. Zhao et al. [19] developed an effective passage arrangement optimization model for multi-stream plate-fin heat exchangers. Mao et al. [20] proposed a methodology to evaluate the coupling effects on the performance of fin and heat exchangers under airflow maldistribution.

To the best of the authors' knowledge, till date the passage arrangement of heat exchanger is designed only for single operating condition. Therefore, a methodology is developed for designing the passage arrangement of multi-stream plate-fin heat exchangers under multiple operating conditions. The remaining part of the paper has been organized as follows: In Section 2, the multi-stream plate-fin heat exchanger model based on integral-mean temperature differences method (IMTD) is established. Passage arrangement coefficients and passage arrangement design space are proposed, and passage arrangement under multiple operating conditions is designed in Section 3. Example and industrial application are investigated and discussed in Section 4. Finally, conclusions are given in Section 5.

## 2. Formulation of multi-stream plate-fin heat exchanger thermal model

The integral mean temperature difference (IMTD) $\Delta T^{IMTD}$ is defined as

$$\Delta T^{IMTD} = \frac{\int_0^L \Delta T dx}{\int_0^L dx} = \frac{1}{L}\int_0^L \Delta T dx \tag{1}$$

Here $\Delta T$ is the temperature difference between two fluids at $dx$ along the length direction of a parallel-flow heat exchanger.

The IMTD method is derived for three-stream heat exchanger in [7], and has the advantage of high convergence and accurateness, so the method is extended from three-stream to multi-stream heat exchanger. The proposed thermal model is based on the following assumptions [2]:

(1) The heat exchanger operates under steady state conditions.
(2) Heat losses to or from the surroundings are negligible.
(3) Longitudinal heat conduction in the fluids and in the wall is negligible.
(4) Wall thermal resistance is distributed uniformly in the entire exchanger.
(5) The streams are parallel, therefore there are only countercurrent and concurrent for neighbor flows.

The multi-stream plate fin heat exchanger can be imagined as a combination of several overlapping three-stream units (sub-exchangers) stacked in a pile. The interactions between the sub-exchangers take place through their common streams and boundaries. The thermal model of multi-stream heat exchanger is constructed from the analysis of three-stream sub-exchangers.

### 2.1. Analysis of three-stream sub-exchangers

For a multi-stream heat exchanger which has $N$ streams, there are $N-2$ three-stream sub-exchangers. According to the flow directions of the streams, there are 8 possible flow arrangements for sub-exchanger $i (i \in [1, N-2])$, which consists of stream $i$, stream $i+1$ and stream $i+2$. All these 8 possible flow arrangements are listed in Table 1, where $j_i = 1$ represents fluid $i$ flows in the $x$ increasing direction along the length of heat exchanger $L$, while $j_i = -1$ represents fluid $i$ flows in the opposite direction. The schematic of these 8 passage arrangements are shown in Fig. 1. The governing differential energy conservation equations of sub-exchanger $i$ considering all these 8 arrangements can be written as Eqs. (2)–(4).

$$j_i(\dot{m}C_p)_i dT_i = \frac{Q_{i-1}}{L}dx - \frac{(UA)_{i+1,i}}{L}(T_i - T_{i+1})dx \tag{2}$$

$$j_{i+1}(\dot{m}C_p)_{i+1} dT_{i+1} = \frac{(UA)_{i+1,i}}{L}(T_i - T_{i+1})dx - \frac{(UA)_{i+2,i+1}}{L}(T_{i+1} - T_{i+2})dx \tag{3}$$

$$j_{i+2}(\dot{m}C_p)_{i+2} dT_{i+2} = \frac{(UA)_{i+2,i+1}}{L}(T_{i+1} - T_{i+2})dx - \frac{Q_{i+2}}{L}dx \tag{4}$$

where $\dot{m}$ is the mass flow rate, $C_p$ is the specific heat at constant pressure, $T$ is the temperature, the reciprocal of UA represents the overall thermal resistance that includes fluid convection, wall conduction and other fouling thermal resistances. The subscript $i$ reflects fluid stream $i$. $Q_i$ is the heat transfer rate between stream $i$ and $(i+1)$, which is positive when the thermal flow direction is from stream $i$ to stream $(i+1)$, and negative when the flow direction is from stream $(i+1)$ to stream $i$. $Q_0$ and $Q_N$ are equal to zeros according to the assumption (2). The heat transfer between streams $(i-1)$ and $i$ at $dx$ is $\frac{Q_{i-1}}{L}dx$ according to the assumption (4).

**Table 1**
The possible passage arrangements for three-stream sub-exchanger $i$.

| | P1 | P2 | P3 | P4 | P5 | P6 | P7 | P8 |
|---|---|---|---|---|---|---|---|---|
| $j_i$ | +1 | −1 | +1 | +1 | +1 | −1 | −1 | −1 |
| $j_{i+1}$ | +1 | −1 | +1 | −1 | −1 | +1 | −1 | +1 |
| $j_{i+2}$ | +1 | −1 | −1 | +1 | −1 | −1 | +1 | +1 |