Performance Evaluation Methods for Multi-stream Plate-Fin Heat Exchanger

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Abstract: Mathematical model of cross type multi-stream plate-fin heat exchanger is established. Meanwhile, mean square error of accumulative heat load is normalized by dimensionless, and the equations of temperature-difference uniformity factor are improved. Evaluation factors above and performance of heat exchanger are compared and analyzed by taking aircraft three-stream condenser as an example. The results demonstrate that the mean square error of accumulative heat load is common result of total heat load and excess heat load between passages. So it can be influenced by passage arrangement, flow inlet parameters as well as flow patterns. Dimensionless parameter of mean square error of accumulative heat load can reflect the influence of passage arrangement to heat exchange performance and will not change dramatically with the variation of flow inlet parameters and flow patterns. Temperature-difference uniformity factor is influenced by passage arrangement and flow patterns. It remains basically unchanged under a certain range of flow inlet parameters.

Key words: multi-stream plate-fin heat exchanger; mean square error of accumulative heat load; temperature-difference uniformity factor; performance evaluation

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Nomenclature

$c_p/(\mathrm{J\cdot kg^{-1}\cdot K^{-1}})$	Fluid specific heat at constant pressure
$f/\mathrm{m^{-1}}$	Fin density
F	The value −1 and 1 for countercurrent and downstream flow separately
$G/(\mathrm{kg\cdot m^{-2}\cdot s^{-1}})$	Mass flow rate
H/m	Fin height
l	Flow direction coordinate
L	Total number of heat exchanger length
n	Transverse flow direction coordinate
N	Total number of passage layer
q/W	Heat flux
$t/℃$	Fin temperature
$T/℃$	Flow temperature
$T^*/℃$	Phase change temperature
W	Total number of heat exchanger width
x/m	Fin coordinate of single passage
$\alpha/(\mathrm{W\cdot m^{-2}\cdot K^{-1}})$	Convection heat transfer coefficient
δ/m	Fin thickness
ϕ	Temperature-difference uniformity factor
φ/W	Mean square error of accumulative heat load

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φ^*	Mean square error of accumulative heat load by dimensionless
$\lambda/(\mathrm{W}\cdot\mathrm{m}^{-1}\cdot\mathrm{K}^{-1})$	Thermal conductivity

0 Introduction

With the development of industry and technology, multi-stream plate-fin heat exchangers are widely used in fields such as petrochemical engineering, aerospace, vehicle and nuclear industry due to their advantages (i. e. small volume, light weight and small size). However, the flow temperature in each passage is affected by many factors due to the complex structure, and the phenomenon as bypass effect of fin, temperature-cross and heat consumption inside may occur under certain conditions[1]. Besides, the heat exchange mechanism will be more complex when cross flow and phase change happen.

Over the decades, uniform method to estimate the heat exchange performance of multi-stream heat exchanger has not been performed. Several theories have been proposed on the influence of passage arrangement to heat transfer performance. For example, Suessmann et al. [2] proposed the local heat balance type passage arrangement, and Prasad[3] proposed passage arrangement based on equal wall temperature principle. Meanwhile, a quantitative criterion is required to make evaluation for heat exchange performance. Mean square error of accumulative heat load method was provided in Ref. [2] in which the periodical vibration of the accumulative heat load around zero reflects the optimal passage arrangement. Other researchers[4-6] optimized passage arrangement using the above performance evaluation through a genetic algorithm. Guo et al. [7] considered that the enhance of heat exchange performance could be conducted in two steps, first of which is to increase the convection heat transfer coefficient, and the second of which is to increase the performance of heat exchanger under same convection conditions (e. g. the flow pattern can affect the heat exchanger performance). Based on the second step, a principle of field synergy was proposed and temperature-difference uniformity factor was established for two-stream. CUI et al. [8] established the temperature-difference uniformity factor for multi-stream heat exchanger by considering the synergistic impact of passage arrangement and fin bypass effect. Based on Ref. [8], LV et al. [9] normalized sub-cell flow temperature-difference with dimensionless parameter and strengthened the weighting factor of adjacent passages to evaluate the merits of passage arrangement. However, the evaluation methods above mainly reflect the influence of passage arrangement to heat exchange performance, applied range of which should be taken into further consideration.

Based on previous mathematical modeling method[10-15], a mathematical model of cross type multi-stream plate-fin heat exchanger is established. Moreover, mean square error of accumulative heat load is normalized with dimensionless parameter and the expression of temperature-difference uniformity factor is improved. On the basis, two methods are compared and analyzed by taking aircraft three-stream condenser as an example, in which application ranges of each method are distinguished and are both expanded. Finally, the origins of two methods are concluded.

1 Mathematical Model

The physical model of cross type multi-stream plate-fin heat exchanger is shown in Fig. 1. In order to simplify the calculation, the mathematical model is based on the following assumptions:

(1) Generalized fin density f is defined assuming that the heat exchange amount between fin and flow/plate within unit fin spacing is distributed into flow and plate cells, to ensure the cell independent from the fin distance.

(2) Assume that flow temperature is equal along fin height direction in a channel while temperature of fin and plate are equal along fin thickness direction. Besides fin and plate contact well

and plate temperature equals to root temperature of fins, as listed below

$$t_{\mathrm{fin},i}(H_i)=t_{\mathrm{fin},i+1}(0)$$

(3) Lateral heat conduction for flows in a passage is ignored by considering the rapid flow velocity of the fluids in heat exchanger.

(4) Lateral heat conduction for fins and plates is ignored by considering that the length of the fin or plate is much larger than its thickness, as listed below

$$\frac{\partial^2 t_i(x)}{\partial l^2}=0,\ \frac{\partial^2 t_{\mathrm{plate},i}}{\partial l^2}=0,\ \frac{\partial^2 t_{\mathrm{plate},i}}{\partial n^2}=0$$

(5) Steady and uniform flow generate in each passage, and it is close to actual in general.

(6) The type of fin is straight fin or serrated fin. The complexity of the fin temperature distribution determines that other types of fin do not apply to the following model.

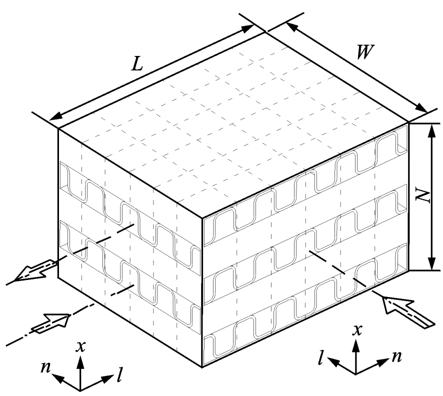

Fig. 1 Diagram of multi-stream plate-fin heat exchanger

Based on assumptions above, energy conservation equations are established for fins, plates and flows. For flows

$$\begin{cases} (FGc_p)_{\mathrm{fluid},i}H_i(1-f\delta)_i\dfrac{\mathrm{d}T_i(l)}{\mathrm{d}l}= \\ \quad \alpha_i(1-f\delta)_i[t_i(0)+t_i(H_i)-2T_i(l)]+ \\ \quad 2f_i\alpha_i\displaystyle\int_0^{H_i}[t_i(x)-T_i(l)]\mathrm{d}x \\ \qquad\qquad T_i(l)\neq T_t^* \\ (FG)_{\mathrm{fluid},i}H_i(1-f\delta)_i\dfrac{\mathrm{d}h_i(l)}{\mathrm{d}l}= \\ \quad \alpha_i(1-f\delta)_i[t_i(0)+t_i(H_i)-2T_i(l)]+ \\ \quad 2f_i\alpha_i\displaystyle\int_0^{H_i}[t_i(x)-T_i(l)]\mathrm{d}x \\ \qquad\qquad T_i(l)=T_t^* \end{cases} \tag{1}$$

The boundary conditions of Eq. (1) are obtained in Eq. (2), namely

$$\begin{cases} T_i(0)=T_{\mathrm{in},i} & F=1 \\ T_i(L)=T_{\mathrm{in},i} & F=-1 \end{cases} \tag{2}$$

Energy conservation equation for fins is shown as follows

$$\lambda\delta_i\frac{\mathrm{d}^2 t_i(x)}{\mathrm{d}x^2}+2\alpha_i[T_i(l)-t_i(x)]=0 \tag{3}$$

Energy conservation equation for plates is shown as follows

$$-\lambda(f\delta)_{\mathrm{fin},i}\frac{\mathrm{d}t_{\mathrm{fin},i}(H_i)}{\mathrm{d}x}+\lambda(f\delta)_{\mathrm{fin},i+1}\frac{\mathrm{d}t_{\mathrm{fin},i+1}(0)}{\mathrm{d}x}+$$
$$\alpha_i(1-f\delta)_{\mathrm{fin},i}[T_i(l)-t_{\mathrm{fin},i}(H_i)]+$$
$$\alpha_{i+1}(1-f\delta)_{\mathrm{fin},i+1}[T_{i+1}(l)-t_{\mathrm{fin},i+1}(0)]=0 \tag{4}$$

It is supposed that the up and down plates are adiabatic. When the up plate is adiabatic, the second and fourth sections in Eq. (4) are both zero. While the down plate is adiabatic, the first and third sections in Eq. (4) are both zero. For flow and fin $i=1,2,\cdots,N$, for plate $i=0,1,2,\cdots,N$.

Based on equations above, cross type multi-stream plate-fin heat exchanger can be divided into $W\times L$ sub-cell heat exchangers, as shown by dash line in Fig. 1. Basic numerical method is described as follows: If the size of sub-cell heat exchanger is small enough, flow direction inside the sub-cell heat exchanger can be ignored. Taking the sub-cell heat exchanger as research object, physical parameters of each stream can be obtained based on flow inlet temperature. Linear equations of flow outlet temperature can be established by energy conservation equations for fins, plates and flows. Then outlet temperature of sub-cell heat exchanger can be obtained. Based on flow direction of each stream, flow outlet temperature of sub-cell heat exchanger is set as inlet temperature of adjacent sub-cell heat exchanger while physical parameters are updated and then fluid temperature distribution can be received in turn. For countercurrent fluid, the beginning temperature field should be assumed and should be iterated until the convergence of upstream fluid temperature field.

Compared with mathematical model acquired by former researchers, the proposed numerical method considers lateral heat conduction characteristics of fin to ensure the calculation accuracy. Meanwhile, the mathematical model is expanded to the calculation of cross type multi-stream plate-fin heat exchanger. On the other hand, this numerical method only needs to iterate under reverse flow pattern, and under other flow patterns iteration is not needed which means higher calculation efficiency. The calculation accuracy and efficiency of the above method have been demonstrated in Ref. [16].

2 Performance Evaluation Methods

2.1 Mean square error of accumulative heat load

Excess heat load between passages will influence the heat exchanger performance on account of local heat balance theory of passages. Heat exchanger has better performance when mean square error of accumulative heat load is smaller. Mean square error of accumulative heat load can be expressed as

$$\varphi=\sqrt{\frac{\sum_{i=1}^{N}\left(\sum_{j=1}^{i}q_j\right)^2}{N}} \tag{5}$$

Based on Eq. (5), φ is normalized with dimensionless parameter in order to reflect the influence of heat exchanger structure to heat exchange performance adequately. The meaning of the parameter will specify in the following paragraphs. Mean square error of accumulative heat load by dimensionless is obtained in

$$\varphi^* = \sqrt{\frac{\sum_{i=1}^{N}\left(\sum_{j=1}^{i}q_j\right)^2}{N\left(\sum q_{\text{hot}}\right)^2}} \tag{6}$$

From Eq. (6) it can be seen that when φ^* approaches to zero, excess heat load between passages is smaller, and the heat transfer performance becomes better.

2.2 Temperature-difference uniformity factor

Based on Refs. [10,11], the temperature-difference uniformity factor is improved to mainly consider the temperature-difference non-uniformity along the flow direction and eliminate the influence of non-adjacent passage. Temperature-difference uniformity factor can be expressed as

$$\phi=\frac{\sum_{i=1}^{N-1}\sum_{j=1}^{W}\sum_{k=1}^{L}\left|T(i+1,j,k)-T(i,j,k)\right|}{\sqrt{(N-1)WL\sum_{i=1}^{N-1}\sum_{j=1}^{W}\sum_{k=1}^{L}\left[T(i+1,j,k)-T(i,j,k)\right]^2}} \tag{7}$$

It can be seen from Eq. (7) that $\phi=1$ when all passages are in the same temperature-difference and $\phi<1$ under different temperature-difference. The value of ϕ could reflect the uniformity of flow temperature so as to reflect the heat exchange performance of heat exchangers.

3 Comparison of Two Methods

In this paper, a three-stream condenser for aircraft vapor cycle refrigeration system is designed, of which cooling fluid is air and thermal fluid is antifreeze and R134a. R134a has a phase change when this condenser operates. Design parameters are shown in Table 1.

Table 1 Performance parameters of three-stream condenser

Medium	T_{in}/℃	T_{out}/℃	Pressure/MPa	Flow rate/(kg • h^{-1})
Air(A)	40	—	0.4	4 000
Antifreeze(B)	70	⩽62	0.8	500
R34a(C)	80	⩽62	1.85	300

For the convenience of pipe layout and reducing air flow resistance, cross flow is applied for cooling fluid. Meanwhile, the serrated fin type is applied for enhancing the heat transfer coefficient of each stream. The length and width of the heat exchanger are taken as 400 mm and 130 mm. Structural design parameters of three-stream condenser are shown in Table 2.

Table 2 Structural parameters of three-fluid condenser

Medium	H/mm	Pitch/mm	δ/mm	Uninterrupted flow length/mm
Air	6.5	2.0	0.15	3
Antifreeze	2.0	1.4	0.15	3
R134a	2.0	1.4	0.15	3

3.1 Different passage arrangements

To prove whether φ, φ^* or ϕ can reflect the influence of passage arrangement to heat transfer performance, assumptions are proposed that passage number and inlet parameters are unchanged under cross flow. By combinatory theory, assume that passage number of flow A, B and C is 14, 13 and 13, there will be $C_{40}^{14} C_{36}^{13}$ types of passage arrangement for three-stream condenser. Obviously, it is not realistic to calculate evaluation factors of all passage arrangements. According to passage arrangement principle of multi-stream plate-fin heat exchanger, heat exchanger has good performance under periodic passage arrangement. Therefore, passage arrangements are established, as is shown in Table 3. Arrangements 1—5 are periodic passage arrangements while Arrangements 6—10 are random permutations and combinations for covering other arrangements.

Table 3 Ten types of passage arrangement in aircraft multi-stream plate-fin heat exchanger

No.	Passage arrangement
1	ABACABACABACABACABACABACABACABACA
2	ACABBACAACABBACAACABBACAACABBACAA
3	ABCAABCAABCAABCAABCAABCAABCAABCAA
4	ACBBAACAACBBAACA ACBBAACA ACBBAACAA
5	ABABABABABABABABACACACACACACACACA
6	CCBCBBACCAABABBBCAACCBAAAAAAAAAAA
7	CCCACCAAABAACACCAAABBAABBBABBAAAA
8	CBAACBCBCABBAACCAABBAAAACBCAAAAAA
9	ABAACAABAABAAACBCCCBAACBABBAAAACC
10	BAACAABBCBACACCCCCBAAABAAABBAAAAA

Considering fluid A is single cooling fluid, total heat load of it can directly reflect the heat transfer performance. The relationships between evaluation factors such as φ, φ^*, ϕ and heat load of fluid A are shown in Fig. 2.

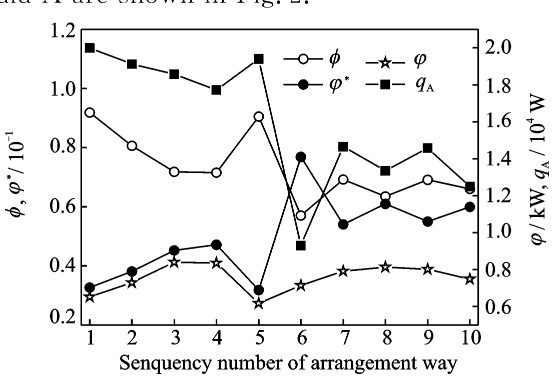

Fig. 2 Relationship between evaluation factors and heat load of fluid A under different passage arrangement

From Fig. 2, it can be seen that no matter passage arrangements are in order or not, φ^* and ϕ changes with heat load of fluid A, which can reflect the effect of passage arrangements to heat transfer performance. For Arrangements 1—5 which have practical significance, φ shows an opposite trend with heat load of fluid A. The smaller φ is, the better the heat transfer performance is. However, for arrangements in disorder, φ is not comparable. This is because φ is related with total heat load of heat exchanger. Former researchers who use φ to evaluate passage arrangements consider that total heat load is same under different passage arrangements and for one flow there is equal distribution of heat load between different passages. This consideration ignores heat transfer characteristics between passages. However, when applying the proposed numerical to conduct heat transfer calculation under a certain heat exchanger boundary dimension, total heat load is different under different arrangements. If there is large difference in total heat load between two passage arrangements, φ is small under passage arrangement that has poor heat transfer performance. This shows that φ normalized with dimensionless parameter to φ^* has significant value.

Outlet temperatures of each fluid under Arrangements 1—5 are listed in Table 4. Because required outlet temperature of hot fluid is less than 62 ℃, it can be concluded from Table 4 that although Arrangement 1 has the largest heat transfer amount, energy distribution of target fluids B and C in Arrangement 2 is more reasonable. This indicates that when temperature-difference uniformity factor is close to 1, the energy efficiency becomes higher. However, this does not represent reasonable energy distribution.

Table 4 Outlet temperature under different passage arrangement ways

Fluid	Outlet temperature/℃				
	1	2	3	4	5
A	58.05	57.20	56.72	55.95	57.46
B	53.10	55.65	55.96	58.07	49.91
C	61.43	59.65	62.30	62.32	64.00

3.2 Different inlet parameters

Based on Arrangement 2, passage arrangement is optimized that the last passage A is discarded and adjacent passages B are combined on the condition of enough strength of passage B. Meanwhile, fin height is twice of original height and passage arrangement is ACABACAACABACAACABACAACABACA. On the basis, inlet flow rate of fluids A and B and inlet temperature of fluids A and C are changed. The relationship between φ, φ^*, ϕ and inlet flow rate and temperature are shown in Figs. 3—6.

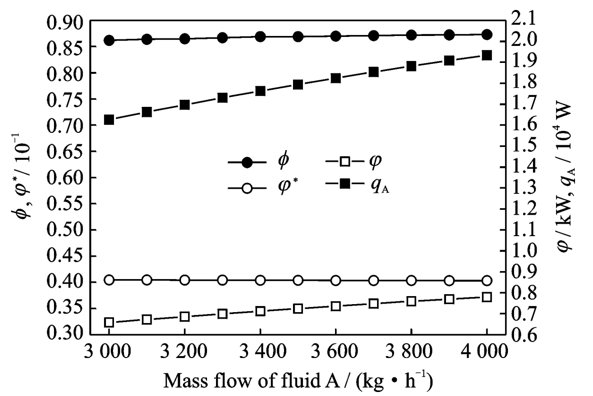

Fig. 3 Relationship between evaluation factors and heat load of A under different flow rates of A

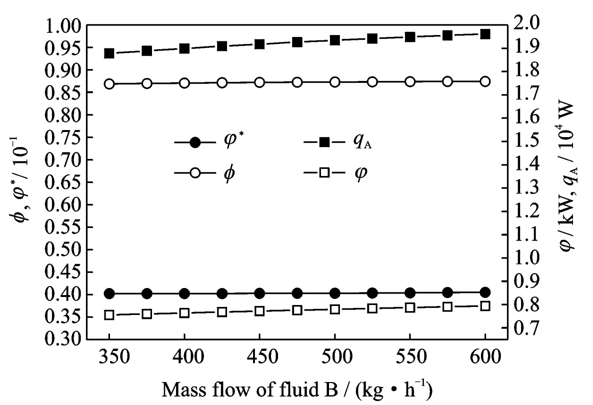

Fig. 4 Relationship between evaluation factors and heat load of A under different flow rates of B

From Figs. 3—6, it can be seen that φ is related with fluid total heat load. When heat load has significant variation, φ has the same variation trend. Within a certain range, φ^* and ϕ are more steady which are not influenced by inlet parameters so that φ^* and ϕ can reflect the effects of structure characteristics to heat exchange performance.

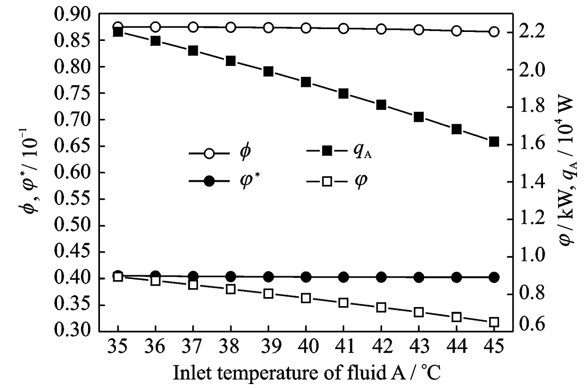

Fig. 5 Relationship between evaluation factors and heat load of A under different inlet temperature of A

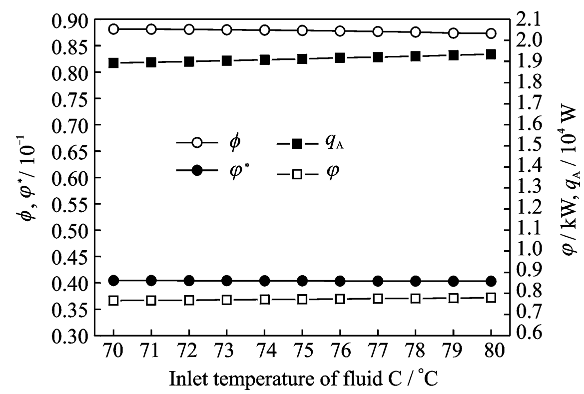

Fig. 6 Relationship between evaluation factors and heat load of A under different inlet temperature of C

3.3 Different flow patterns

Passage arrangement is ACABACAACABACAACABACAACABACA. Evaluation factors of cross flow and parallel flow are calculated. the results are listed in Table 5. Heat exchanger size of flow patterns 1—3 is 130 mm × 400 mm and which of flow patterns 4—6 is 400 mm × 400 mm.

Comparing the Arrangements 1, 2, it can be concluded that when flow pattern changes from cross flow to parallel flow without considering the growth of resistance, although ϕ decreases greatly due to poor temperature-difference field, heat load shows a little difference for convection heat transfer coefficient of fluid A increases. Comparing Arrangements 1 and 3, flow directions of thermal fluids B and C vary from same to adverse. In this case, convection heat transfer coefficient remains unchanged, and heat load as well as ϕ has a slightly decrease. Under Arrangements 4 and 5, if length and width of heat exchanger are similar, it can be seen that convection heat transfer coefficient of fluid A remains unchanged while heat load of fluid A and ϕ have similar change trend. It can be concluded from above that using ϕ to evaluate the influence of flow patterns to heat transfer performance should be under the condition of convection heat transfer coefficient of each stream remaining unchanged, which is opposite to Refs. [8,9]. The reason is that the possible change of surface convection heat transfer coefficient caused by cross flow does not take into consideration in Refs. [8,9].

Table 5 Calculating results under different flow patterns

Size/ (mm×mm)	No. of flow patterns	Flow direction A	B	C	T_{out}/℃ A	B	C	q_A/kW	φ/kW	φ^*/10^{-1}	ϕ	α_A/(W·m^{-2}·K^{-1})
130×400	1	↑	→	→	57.4	55.79	57.62	19.34	0.780	0.403	0.873	359.4
	2	→	→	→	56.7	57.94	55.94	18.55	0.751	0.405	0.658	643.0
	3	↑	←	→	57.03	57.49	57.34	18.92	0.761	0.402	0.858	359.4
400×400	4	↑	→	→	62.52	47.19	48.27	25.01	1.054	0.421	0.746	360.2
	5	→	→	→	57.38	57.43	57.39	19.31	0.790	0.409	0.497	359.5
	6	↑	←	→	59.57	56.62	55.31	21.74	0.885	0.407	0.676	359.8

It also can be concluded that change flow patterns under a certain structure size of heat exchanger, φ^* keeps unchanged while φ increases with increasing total heat load, which has opposite trend comparing with φ under different passage arrangements. This indicates that φ is common result of total heat load and excess heat load between passages.

4 Conclusions

Mathematical model of cross type multi-stream plate-fin heat exchanger is established, and equations of mean square error of accumulative heat load and temperature-difference uniformity factor are modified. Moreover, taking aircraft three-stream condenser as an example, evaluation factors and heat exchange performance are compared and analyzed, then the applicability of the two evaluation methods is obtained at last. Conclusions are as follows:

(1) Temperature-difference uniformity factor (ϕ) reflects the influence of non-uniformity of temperature-difference field to heat exchange performance. It is influenced by passage arrangement and flow patterns. As passage arrangement and flow patterns become better, ϕ is closer to 1 while ϕ remains basically unchanged under a certain range of flow inlet parameters.

(2) Mean square error of accumulative heat load (φ) is common result of total heat load and excess heat load between passages, which can be influenced by passage arrangement, flow inlet parameters as well as flow patterns.

(3) Dimensionless parameter of mean square error of accumulative heat load (φ^*) eliminates the effect of total heat load, which can reflect the influence of passage arrangement to heat exchange performance and will not change dramatically with the variation of flow inlet parameters and flow patterns.

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