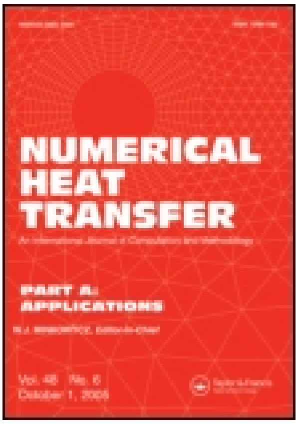

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OPTIMUM DESIGN OF TWO-ROW SLOTTED FIN SURFACE WITH X-SHAPE STRIP ARRANGEMENT POSITIONED BY "FRONT COARSE AND REAR DENSE" PRINCIPLE, PART II: RESULTS AND DISCUSSION

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In Part I of this article, design considerations for 15 slotted fin surfaces are introduced, and the physical/mathematical model and numerical methods are described. The strips in all slotted fin surface are designed according to two general guidelines: "front coarse and rear dense" and "X-shape arrangement." In this article, the specific flow and heat transfer characteristics of the 15 slotted fin surfaces and their comparisons with the plain plate surface are presented. The major findings are as follows. For all 15 slotted fin surfaces, their j/f versus Re curves cross with the curve of the plain plate fin surface at some turning Reynolds number, beyond which the j/f ratio of slotted fin surfaces is higher than that of plain plate fin surfaces, and vice versa. The variation character of heat transfer rate versus Reynolds number under identical pumping power condition has the same feature, with its turning Reynolds number being appreciably lower than the former one. However, at the identical flow rate the heat transfer rates of all the slotted fin surfaces are higher than that of the plain plate fin, and the larger the velocity, the more significant the enhancement. Among the four techniques adopted—changing the number of strips, shortening the strip length, changing the location of the strips, and splitting one strip into two—the effect of strip location is the most and that of splitting one into two is the least. The field synergy principle analysis shows that the higher heat transfer rate of the slotted fin surfaces comes from their better synergy between velocity and temperature gradient. Two patterns of slotted fin are recommended for the two-row plate fin-and-tube surface.

1. INTRODUCTION

In Part I of this article [1], the physical model and mathematical formulation are described in detail and design considerations for three groups including 15 slotted

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NOMENCLATURE

A	heat transfer area, m^2	θ	local intersection angle, deg
c_p	specific heat at constant pressure, kJ/kg K	θ_m	mean intersection angle, deg
D_c	outer tube diameter, m	λ	thermal conductivity, W/m K
f	friction factor	μ	dynamic viscosity, kg/m s
h	heat transfer coefficient, W/m^2 K	ρ	air density, kg/m^3
Int	integral $[-\iint_v \int \rho c_p (U \cdot \nabla T)\, dv]$, W	**Subscripts**	
j	Colburn factor, dimensionless	idfr	identical flow rate
L	fin depth in air flow direction, m	idpp	identical pumping power
p	pressure, Pa	in	inlet
Δp	pressure drop, Pa	m	mean
Re	Reynolds number	max	maximum
T	temperature, K	min	minimum
U	velocity vector	out	outlet
V	volume, m^3	w	wall

fin surfaces are introduced. In this article, the specific flow and heat transfer characteristics of the 15 slotted fin surfaces and their comparisons with the plain plate surface will be presented. Performance comparisons are conducted by using two constraints: the j/f ratio at identical Reynolds number and the heat transfer rate under identical pumping power. In addition, the inherent relation between the field synergy angle and the heat transfer rate for the 16 fin-and-tube surfaces are examined.

In the following, the definitions of some parameters and the validation of the code and mesh independence of the solutions are described first, followed by detailed performance comparisons according to the above two constraints. Then performance analysis from the field synergy principle for the 15 slotted fin surfaces and the plain plate surface are provided. Finally, a series of conclusions is drawn.

2. RESULTS AND DISCUSSION

2.1. Parameter Definitions

To present the numerical results, some parameters should be defined. Apart from the eight parameters defined in the first part of the article [1], the following additional parameters will be used in the discussion:

$$\theta = \arccos\left(\frac{U \cdot \nabla T}{|U||\nabla T|}\right) \tag{1}$$

$$\theta_m = \frac{\iiint_V \theta dv}{\iiint_V dv} \tag{2}$$

$$\mathrm{Int} = -\iiint_V \rho c_p (U \cdot \nabla T) dv \tag{3}$$

It should be noted that for the case studied, air is cooled and the integral of the convective term $\iiint_V \rho c_p (U \cdot \nabla T)\, dv$ is less than zero. For convenience of presentation, a minus sign is added in the definition of Int in Eq. (3). Further, for the case of fluid being cooled, the synergy angle is greater than 90°. For convenience of presentation its supplementary angle is adopted as the synergy angle.

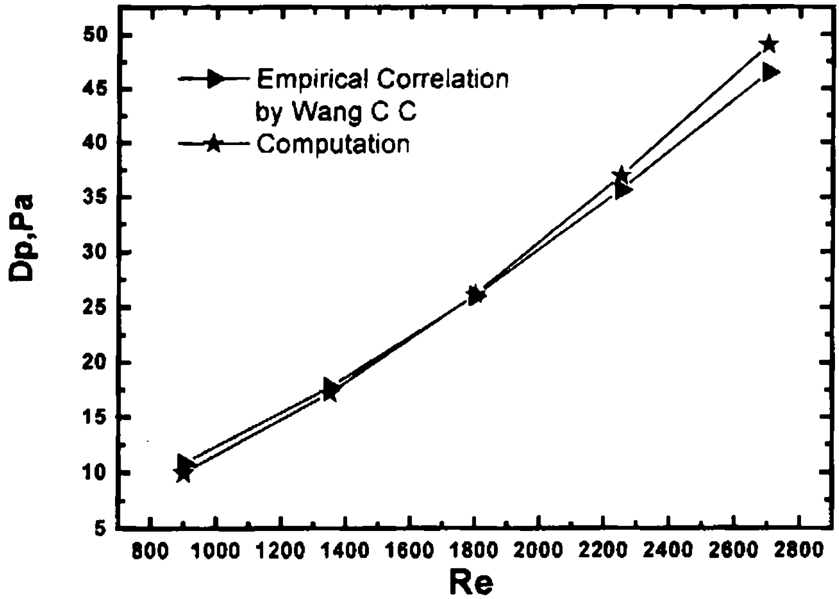
2.2. Validation of the Code and Mesh Independence of the Solutions

To validate the computational model and the code developed, preliminary computations were first conducted for the plain plate fin-and-tube surface, and the predicted pressure drop and Colburn *j* factor were compared with the experimental correlations provided by Wang et al. [2]. Their correlations were developed based on 74 samples of plain plate fins. The variation range of the Reynolds numbers is from 2.0×10^2 to 2.0×10^4. These are the most accurate and reliable correlations with wide applicable ranges known to the present authors. The comparison results are provided in Figure 1: the maximum deviation in pressure drop is less than 8% and that of the Colburn *j* factor is less than 10%. Such small discrepancy between numerically predicted and experimental results should be regarded as quite good [3–6]. The good agreement between the predicted and tested results shows the reliability of the physical model and the code developed. It should be noted that, according to the test data reduction of [2], the *j* correlation is the true value of the *j* factor. Thus, in the above comparison our numerical data are the corresponding values for which the total fin effectiveness has been separated.

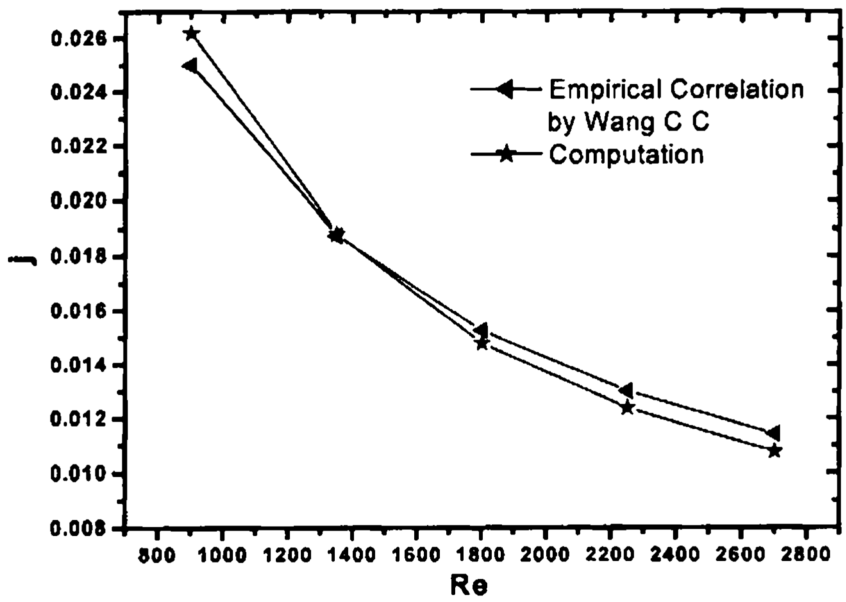
In order to adopt an appropriate grid system, a grid refinement was conducted to investigate the influence of the grid density on the computational results. Consider the j/f ratio of slit fin 2 as an example. The results of four different grid systems are shown in Figure 2. Compared to the finest grid, $143 \times 114 \times 24$, the grid $143 \times 66 \times 24$ yields a value of j/f ratio which is 1.5% lower than that of the finest grid system. Thus, in order to save computer resources, the grid system $143 \times 66 \times 24$ was adopted in the other numerical simulations. The structure of grid is shown in Figure 3, in which the inlet and outlet extended domains are omitted. The code validation was also conducted on this grid system.

2.3. Performance Comparison

In the following presentation, comparisons between plain plate fin and the slotted fin surfaces will be presented. To every group of slotted fin surface, comparisons are made for three aspects: (1) comparison of j/f value under identical Re number [7]; (2) comparison of heat transfer rate under identical pumping power [8]; and (3) comparison of the relation between domain-average synergy angle and the heat transfer rate. In addition, in the design of heat exchangers, the heat transfer rate under a given flow rate is an important parameter which will directly affect the volume of the heat exchanger designed: the more the heat transfer rate per unit surface area, the less the volume of heat exchanger. Therefore, in the discussion, apart from the heat transfer rate under identical pumping power, the heat transfer rate under identical flow rate is also taken into account.



(*a*) Pressure drop



(*b*) Colburn factor j

Figure 1. Comparison between predicted and empirical correlation results.

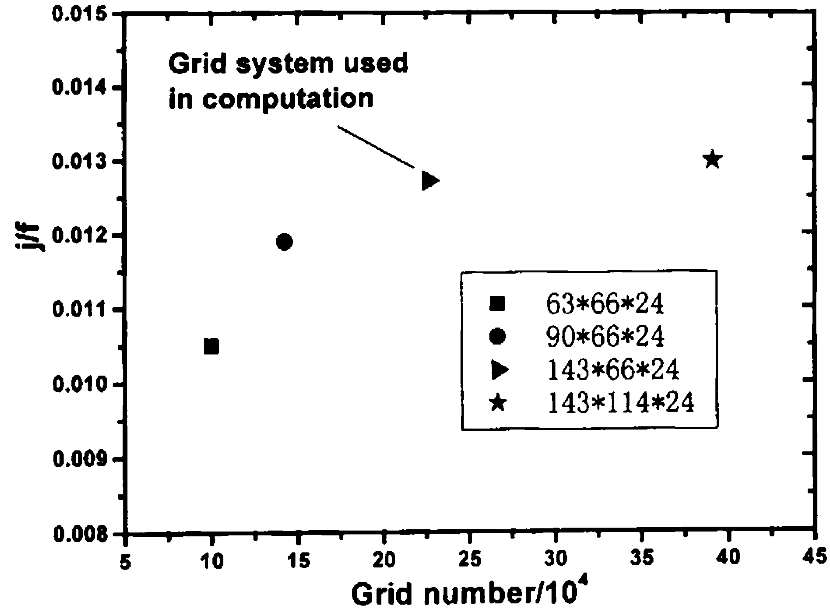
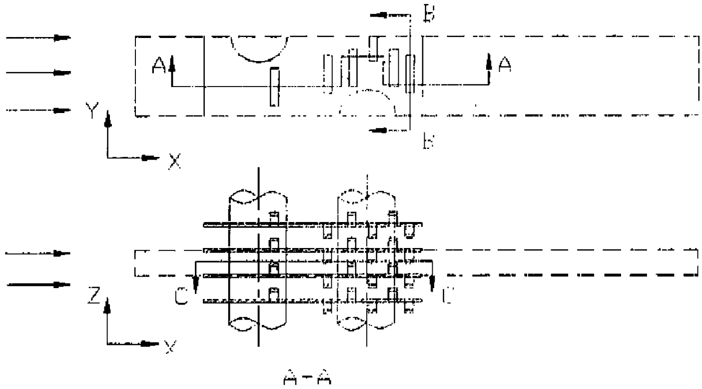


Figure 2. Value of j/f of slit fin 2 versus grid number.

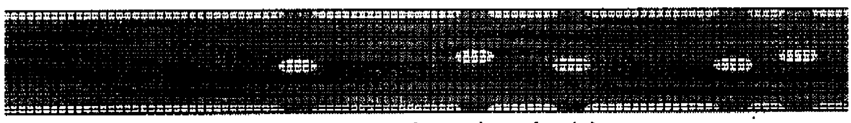
2.3.1. Comparisons of j/f under identical Re number. In Figures 4*a*, 5*a*, 6*a*, and 7*a*, comparison results of j/f ratio of the three groups introduced in our first article are presented in the Reynolds number range from 901 to 2,702 (the corresponding frontal velocities range from 1.0 to 3 m/s, which covers the working flow velocity for air conditioning).

From Figure 4*a*, three notable features can be found. First, the j/f versus Re curve of each slotted fin surface has a turning point where the j/f versus Re curves of the plain plate fin surface and the slotted fin surface cross each other. When the Reynolds number is greater than that of the turning point, the j/f value of the slotted fin surface is greater than that of the plain plate fin, and vice versa. Second, with a decrease in strip number, the turning Re number decreases: the turning Re numbers for slit 4, slit 3/slit 2, and slit 1 are about 2,252, 1,802, and 1,700, respectively. Third, the value of j/f at low frontal velocity deceases with increase in strip number: the value of j/f of slit fin 1 is 10% higher than that of slit fin 4, and 6% and 3% higher than that of slit fin 3 and slit fin 2, respectively. However, in the higher Reynolds number region (over Re $\approx$ 2,252), all four slotted fin surfaces have a larger value of j/f and the difference between slotted fin surfaces becomes not so significant. Thus we can conclude that, for the X arrangement of strips with "front coarse and rear dense" principle, decreasing the strip number will benefit for increasing the ratio of j/f at low frontal oncoming velocity and decreasing the turning Re number, and the values of j/f for different slotted fin surfaces are all higher than that of the plain plate fin beyond the turning Reynolds number.

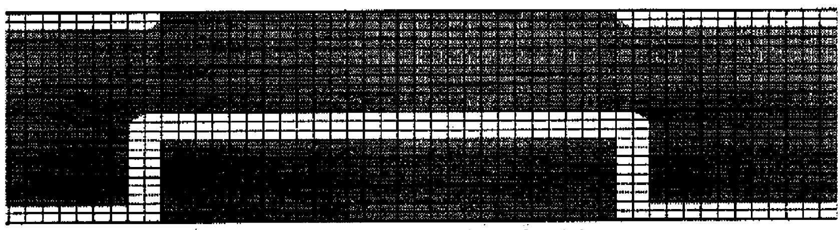
Now the predicted results of the second group are examined. As indicated above, slit fin 3, with five strips in the first group, is partly shortened in strip length



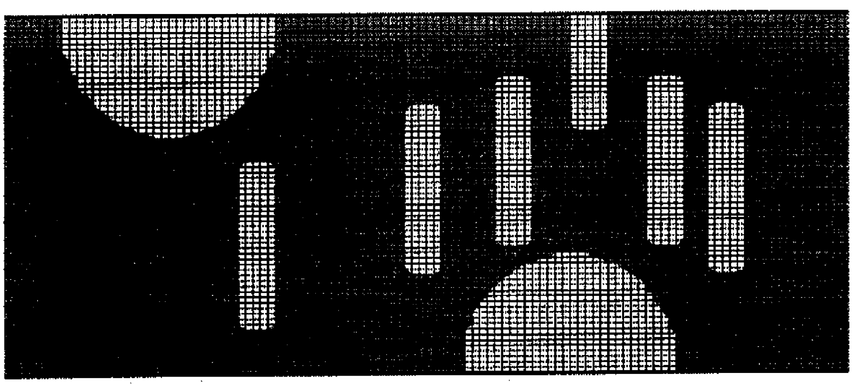
(*a*) Cutaway position of computational domain of strip plain fin



(*b*) A-A section view in (a)

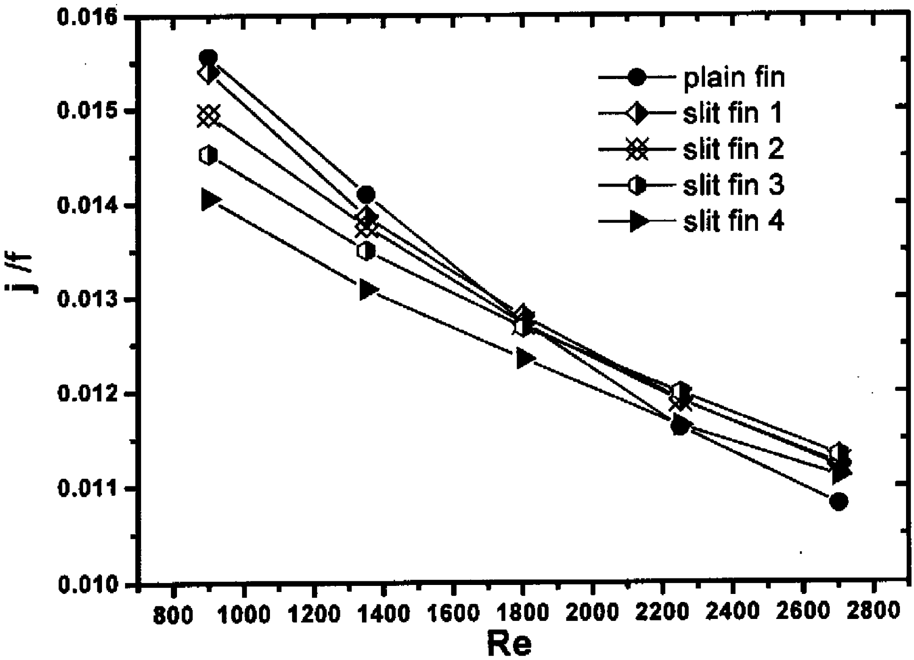


(*c*) B-B section view in (a)

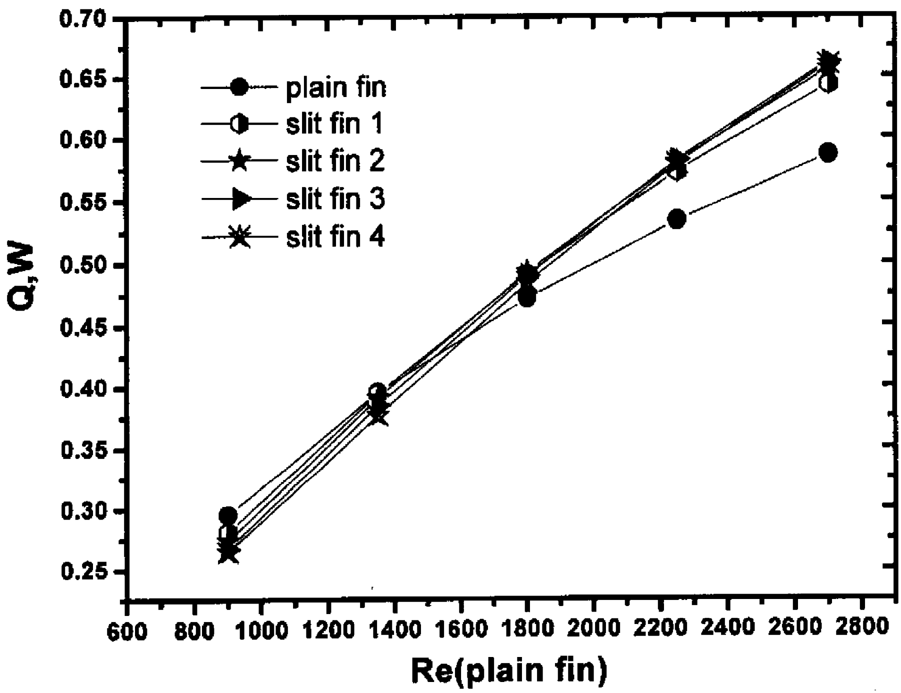


(*d*) C-C section view in (a)

Figure 3. Grid structure of computation.

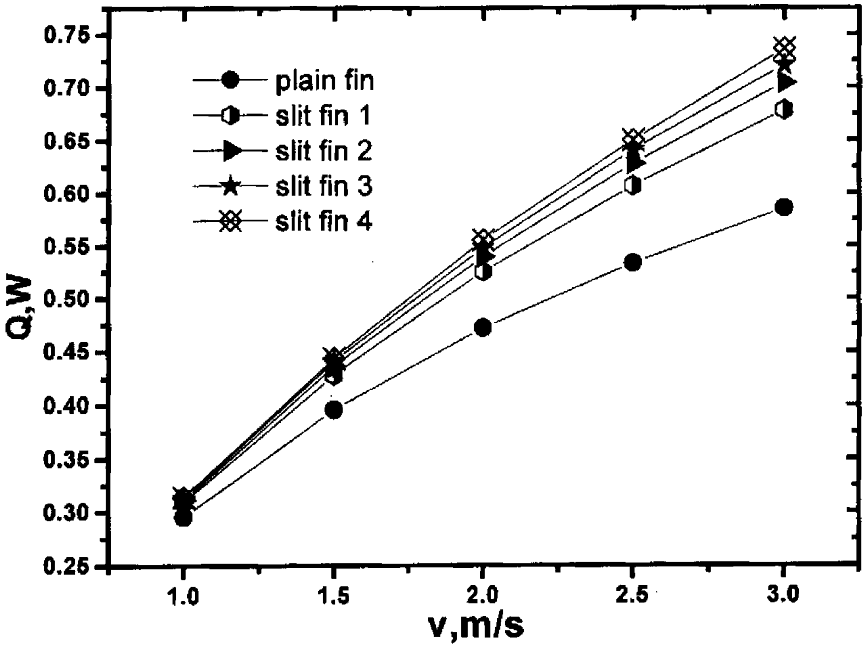


(a) j/f vs. Re

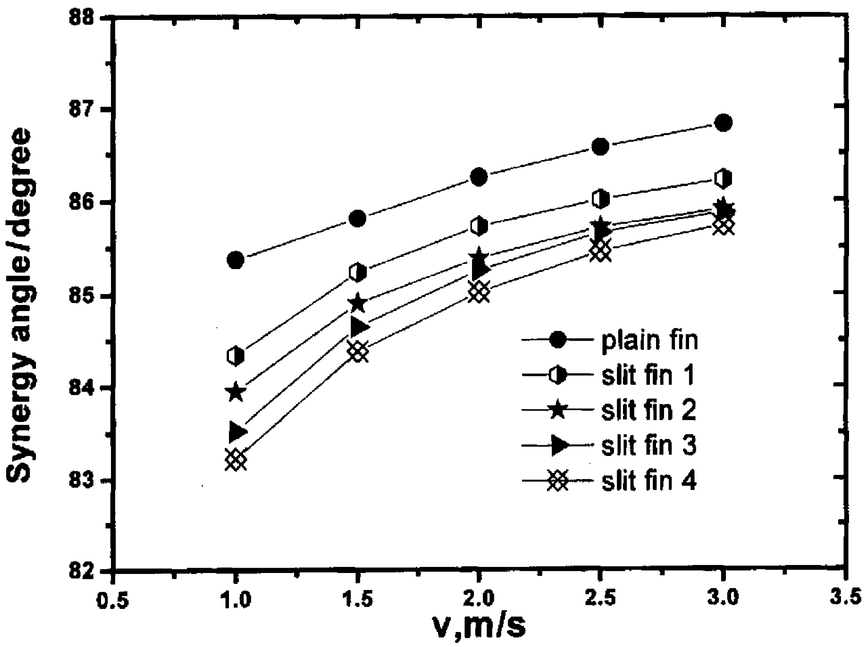


(b) Q vs. Re (plain fin)

Figure 4. Computational results for the first group.

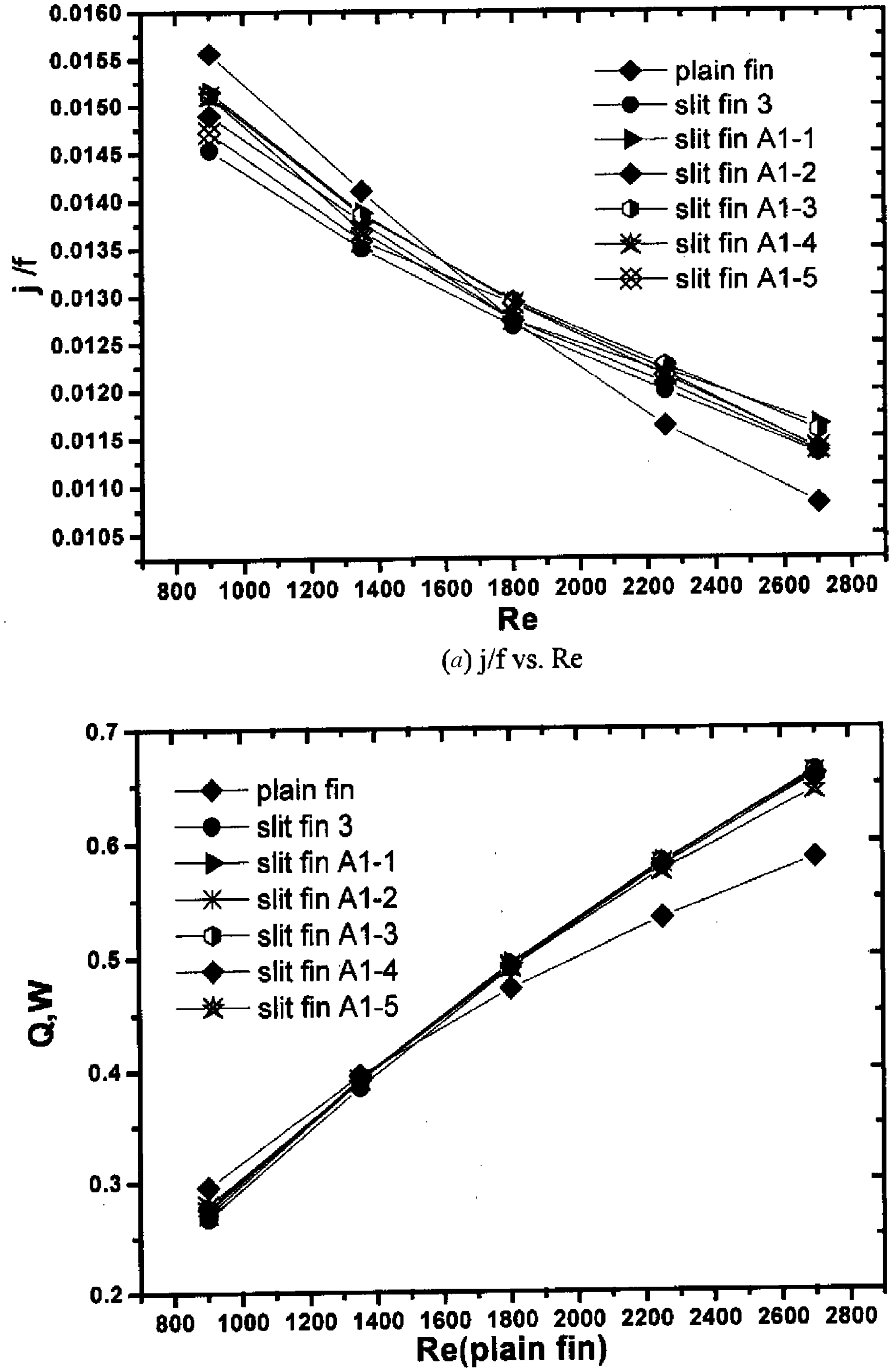


(c) Q vs. v



(d) Synergy angle vs. v

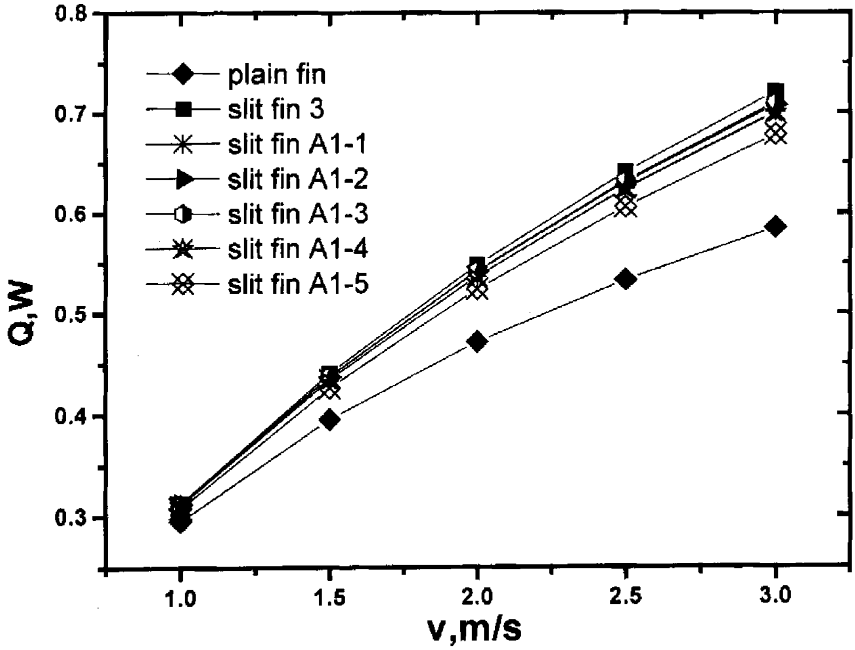
Figure 4. Continued.



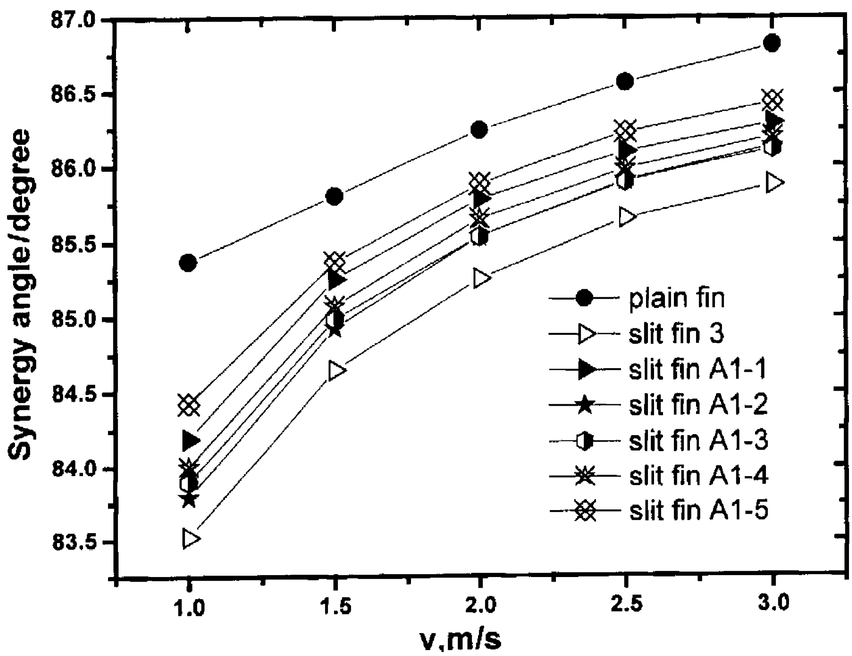
(*a*) j/f vs. Re

(*b*) Q vs. Re (plain fin)

Figure 5. Computational results for j/f versus Re for the second group (first series).

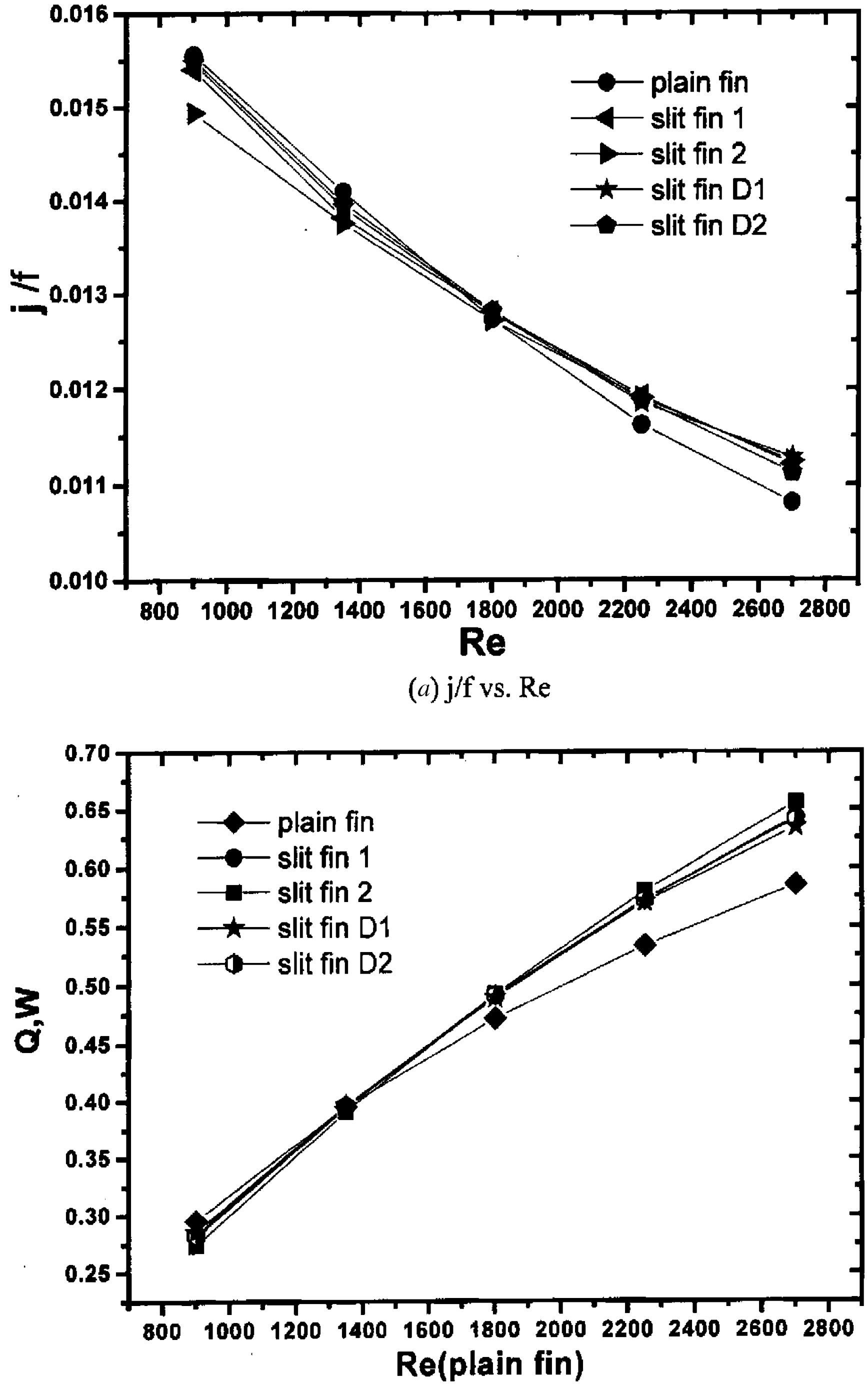


(*c*) Q vs. v



(*d*) Synergy angle vs. v

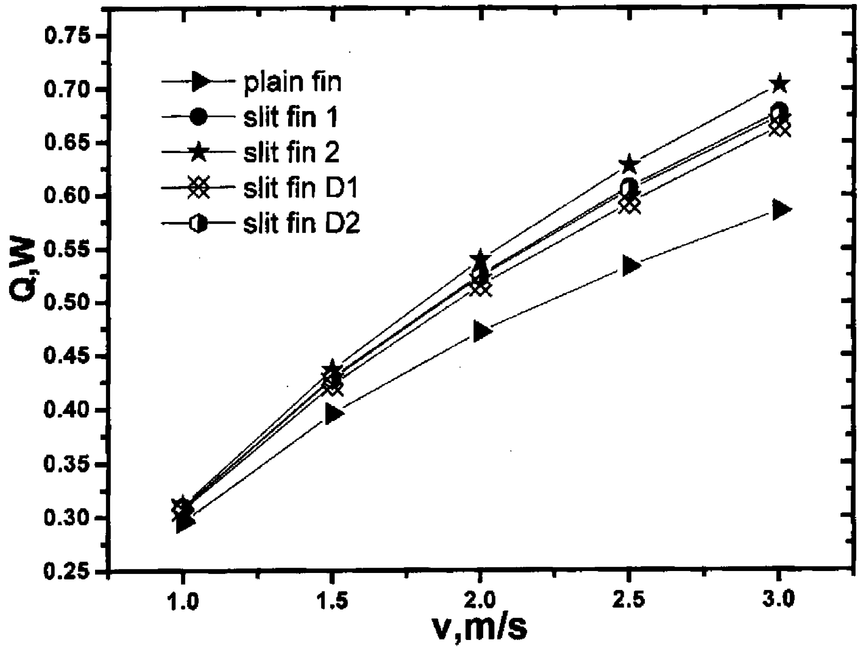
Figure 5. Continued.



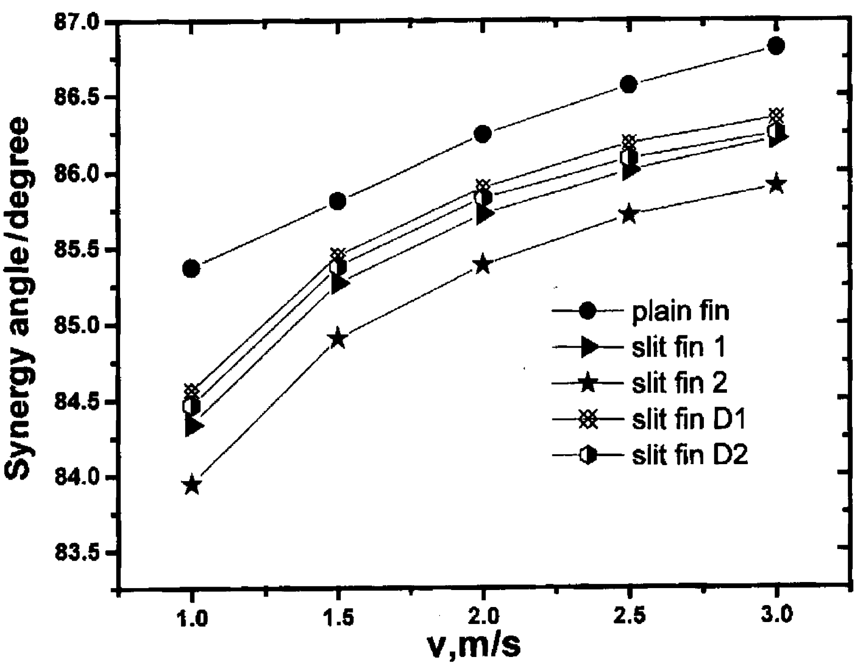
(*a*) j/f vs. Re

(*b*) Q vs. Re (plain fin)

Figure 6. Computational results for j/f versus Re for the second group (second series).

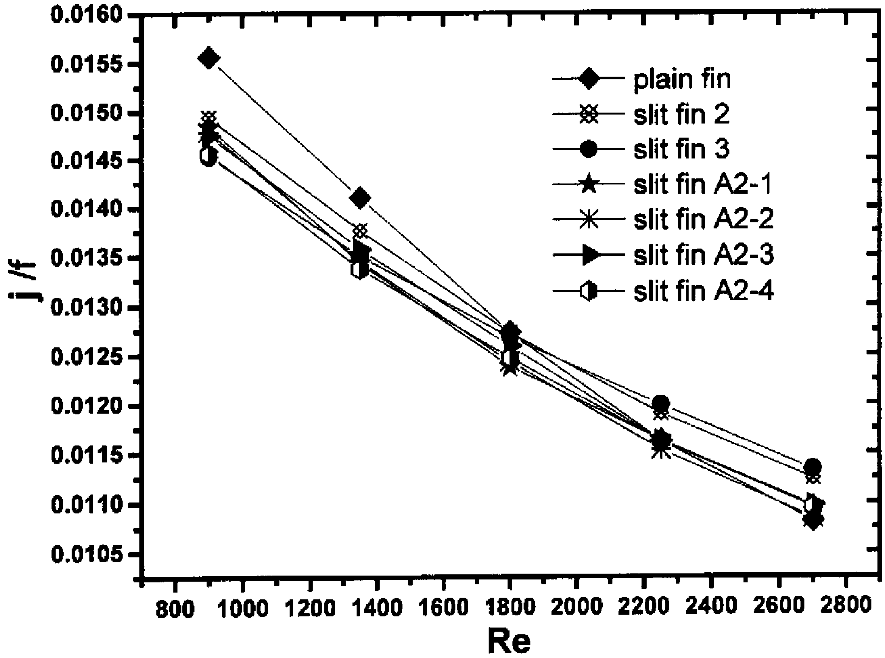


(*c*) Q vs. v

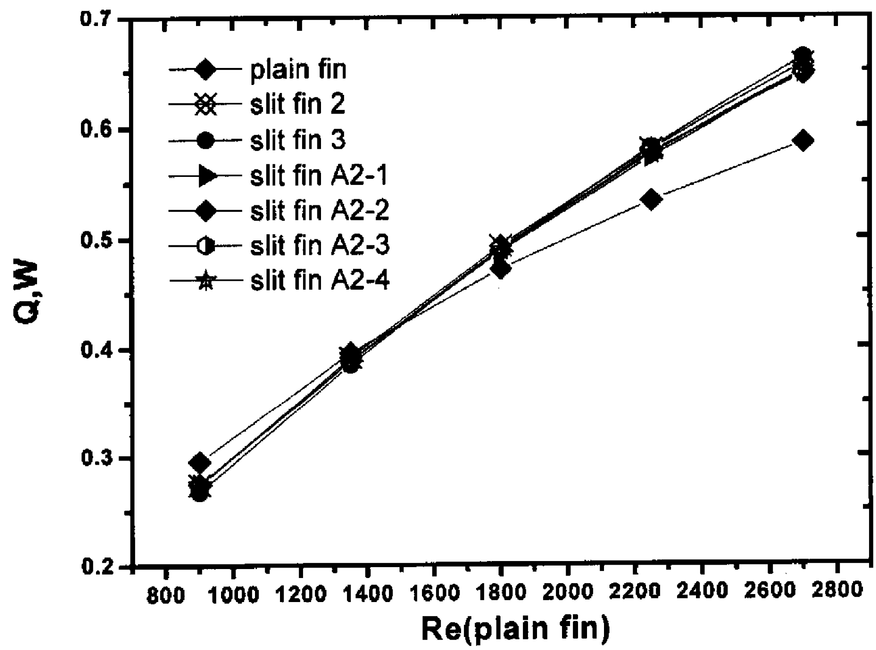


(*d*) Synergy angle vs. v

Figure 6. Continued.

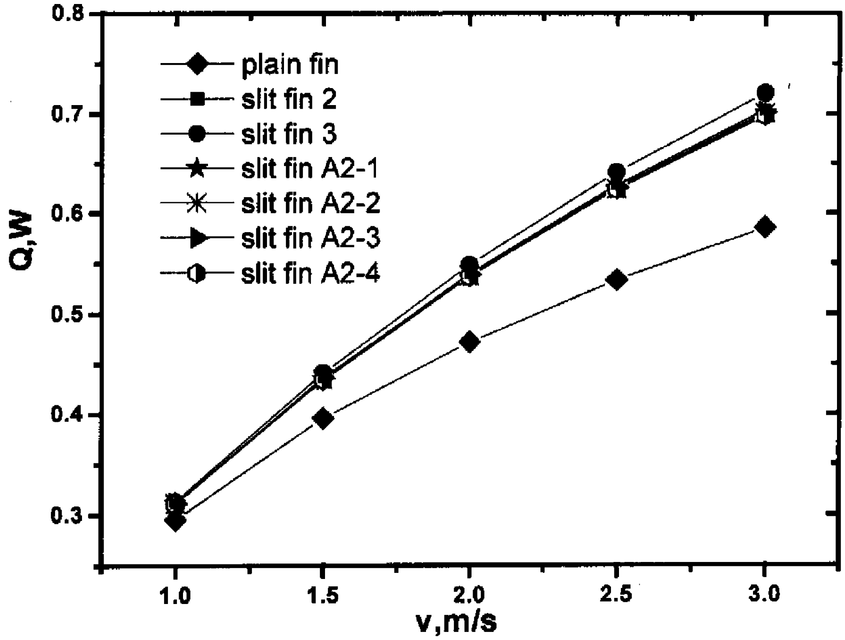


(*a*) j/f vs. Re

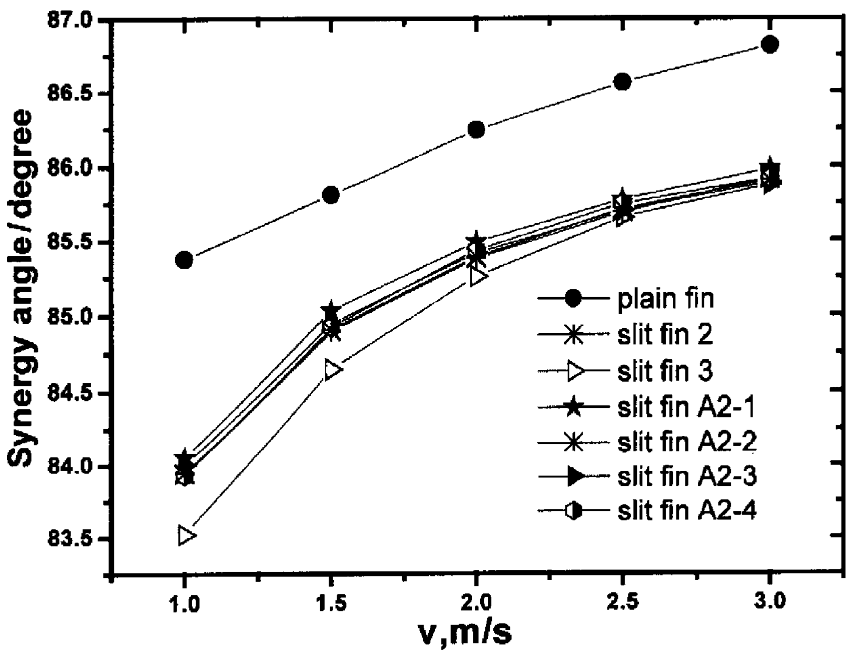


(*b*) Q vs. Re (plain fin)

Figure 7. Computational results for the third group.



(*c*) Q vs. v



(*d*) Synergy angle vs. v

Figure 7. Continued.

to form the first series, leading to slit fins A1-1 to A1-5 as shown in Figures 5*a*–5*e* of our 1st article [1]. The predicted results of this series are presented in Figure 5. From Figure 5*a*, we can easily find that the values of j/f of the slit fins of this series are all higher than that of slit fin 3 in the entire working flow velocity. Moreover, it is to be noted that not only are all the j/f values of the slit fins of this series higher than that of slit fin 3, but also the turning Re number is clearly decreased. Among the five types of slotted fin surfaces designed with strip length variation, the j/f values of slit fin A1-1 and slit fin A1-3 are almost the same and are the best in the whole ranges of working flow velocity. The value of j/f for slit fin A1-3 is 4.1% and 2.1% higher than that of slit fin 3 at 1 m/s and 3 m/s oncoming flow velocity, respectively, but the value of heat transfer rate of slit fin A1-3 is only 0.42% and 1.5% lower than that of slit fin 3 at 1 m/s and 3 m/s oncoming flow velocity. In addition, its turning Re number drops from 1,850 of slit fin 3 to about 1,550, which is even less than that of slit fin 1 of the first group.

In the field of air conditioning, the operation noise of indoor air conditioning is often strictly limited, leading to a design of low frontal oncoming velocity (say, about 1 m/s). Hence it is important to further improve the performance of the slit fin surface at low frontal oncoming velocity. From the analysis of the values of j/f for above two groups, it can be easily found that decreasing the number of strips and partly shortening the length of strips both can enhance the value of j/f at low oncoming velocity, and strip length variation seems to be more efficient in this regard. For example, the j/f for slit fin 2 with four strips is 2.7% higher than that for slit fin 3 at 1 m/s oncoming flow velocity, the heat transfer rate of slit fin 2 is 5.4% lower than that of slit fin 3 at the above velocity, and its turning Re number 1,820 is almost equal to that of slit fin 3 (1,850). However, the j/f of slit fin A1-3 is 4.1% higher than that of slit fin 3 at the velocity of 1 m/s, its heat transfer rate is almost the same as that of slit fin 3 (only 0.42% lower), while its turning Re number is only 1,550. Thus, for the X arrangement of strips with "front coarse and rear dense" principle, the technique of strip length shortening is a more effective way to enhance the j/f ratio of the slit fin surface.

The results of the second series of the second group are now examined. As indicated above, based on the results of slit fin 1 and slit fin 2, the technique of strip length shortening is adopted to form two new slotted fin surfaces, namely, slit fin D1 and slit fin D2 as shown in Figures 5*f* and 5*g* of our first article [1]. The predicted results are presented in Figure 6. The values of j/f for slit fin D1 and slit fin D2 are almost the same as that for the plain plate fin surface at 1 m/s velocity (99.7% and 99.4%, respectively); at the same time, their values for j/f are 2.9% and 4.3%, respectively, higher than that for the plain plate fin at the velocity of 3 m/s. Meanwhile, their turning Re number is almost the same as that of slit fin 1 and slit fin 2 of the first group. Thus, with regard to j/f, the performance of this series is better than that of slit fin 1 and slit fin 2 of the first group. This is a quite interesting result. As indicated by Shah and Sekulic, for offset strip fins, the ratio of j to f for a plain fin is only about 80% [9]. Our experience shows that by careful design of the strips, the ratio of j/f may be greater than 80% and close to that of a plain plate fin.

Attention is now turned to the numerical results of the third group. Slit fins A2-1 to A2-4 of the third group are formed by cutting one strip of slit fin

Table 1. Turning Reynolds number of slotted fin surface for j/f versus Re (identical flow rate)

Fin group no.	Turning Re
1	1,700–2,252
2	1,550–1,850
3	~2,250
4	~1,700

2 of four strips in the first group into two shorter strips (splitting variation), and the position of strips is the same as slit fin 2. In the following, we will compare performance with both slit fin 2 and slit fin 3, since the strip number of this group is the same as that of slit fin 3 of the first group (i.e., five strips). The comparison results are provided in Figure 7. From Figure 7*a* we can easily find that the values of j/f are a bit higher than that of slit fin 3 at low oncoming velocity, but much lower than that of slit fin 3 at high oncoming velocity and, in the entire working flow velocity, all much lower than that of slit fin 2. Thus, as far as j/f is concerned, the performance of this group is worse than that of slit fin 2. Thus, splitting strip length without any other change is not a good practice for enhancing the ratio of j/f.

For the 15 slotted fin surfaces, the turning Reynolds numbers of the j/f versus Re curve are summarized in Table 1.

2.3.2. Comparison of heat transfer under identical pumping power. Another constraint to appraise the performance of three groups and 15 types of slit fin is identical pumping power. As the characteristic dimensions of the 16 types (including the plain plate one) of fin surface are all the same, the following conditions must be met to satisfy this constraint:

$$\left(f\mathrm{Re}^3\right)_{\mathrm{Plain}} = \left(f\mathrm{Re}^3\right)_{\mathrm{Slit\ fin}} \tag{4}$$

The evaluation proceeds in the following way. First, according to the computational results, the relations between f factor, Nusselt number, and Reynolds number are obtained. Then select a Reynolds number for the plain fin; the corresponding Reynolds numbers for the slit fin can be determined in an iterative manner from the relations of f versus Re and Nu versus Re. Taking the Re of the plain plate fin as the abscissa, the corresponding Reynolds number for the other 15 cases can be found and, hence, the related heat transfer rate (Q) can be found. The values of Q are plotted as the ordinates. The results of the three groups are shown in Figures 4*b*, 5*b*, 6*b*, and 7*b*, respectively.

It can be seen that, for every group, the comparison result is qualitatively similar to that of j/f under identical Re number constraint. That is, each curve of the slotted fin surface has a tuning point beyond which the performance of the slotted fin surface is better than that of the plain plate fin surface. And the corresponding turning Reynolds numbers of all the slotted fin surfaces are appreciably lower than

Table 2. Turning Reynolds number of slotted fin surface for Q versus Re (identical pumping power)

No.	Fin name	Turning Re
1	Slit 1	1,228
2	Slit 2	1,250
3	Slit 3	1,294
4	Slit 4	1,348
5	A1-1	1,241
6	A1-2	1,260
7	A1-3	1,257
8	A1-4	1,270
9	A1-5	1,275
10	A2-1	1,313
11	A2-2	1,306
12	A2-3	1,280
13	A2-4	1,303
14	D1	1,194
15	D2	1,218

those found in the comparison with identical Reynolds number. After careful examination, the turning Reynolds number of 15 types of slotted fin surfaces have been revealed and are listed in Table 2. As can be seen there, the turning Reynolds numbers of the 15 slotted fin surfaces studied range from 1,200 to 1,350, which occurs at a bit lower than the oncoming flow velocity of 1.5 m/s.

The effect of strip number on fin performance can be found from Figure 4*b*. From this figure, it can be found that, below the turning point, with decrease of strip number, the heat transfer rate Q increases, but beyond the turning point, heat transfer rate increases with increase in strip number. For example, the heat transfer rate Q of slit fin 1 is 6.2% higher than that of slit fin 4 at 1 m/s frontal oncoming velocity but 2.6% lower than that of slit fin 4 at 3 m/s frontal oncoming velocity. This implies that the advantage of the slotted fin surface becomes more significant at higher oncoming velocity. At the oncoming flow velocity of 3 m/s of the plain plate fin surface, the heat transfer rate of slit 1 to slit 4 under identical pumping power is about 9.8% to 12.7% higher than that of plain one.

In Figure 5*b*, it is shown that, among the five slotted fins with strip length variation (the first series of the second group), the performance of slit fin A1-1 is the best. Its heat transfer Q is 3.5% higher than that of slit fin 3 at 1 m/s velocity and almost equal to that of slit fin 3 at velocities above 2.5 m/s.

From Figure 6*b*, it can be seen that the second series of the second group can also further enhance heat transfer under identical pumping power at a higher frontal oncoming velocity. For example, the heat transfer rate of slit fins D1 and D2 can reach 95.7% and 96.6%, respectively, of that of the plain plate fin at 1 m/s oncoming velocity, and 9.5% and 8.4% higher than that of the plain plate fin at 3 m/s oncoming velocity.

The performance of the slit fins of the third group under identical pumping power is a bit worse than that of their progenitor (slit fin 2).

2.3.3. Comparison of heat transfer under identical flow rate. Figures 4*c* to 7*c* provide the results of such comparison. It can be clearly observed that for the 15 slotted fin surfaces, their heat transfer rates are all higher than that of the plain plate fin, and the larger the oncoming velocity, the more significant is the heat transfer enhancement. By carefully inspecting these figures, we can find the following features. Among all the 15 slotted fin surfaces, the heat transfer rates of fin 3 and 4 are the highest. At the frontal velocities of 1 and 3 m/s, the enhancement of heat transfer rate of fin 3 over the plain plate fin is 6% and 23.1%, respectively. Special attention should be paid to slit fin A1-3. Its heat transfer rate is only a bit less than that of fin 3 (see Figure 5*c*), while its ratio of j/f is higher than that of slit fin 3 and is almost the highest among the slotted fins of the first series of group 2 (Figure 5*a*). In addition, its curve of Q versus $(\mathrm{Re})_{\mathrm{idpp}}$ is quite close to that of slit fin 3 (Figure 5*b*). Thus fin A1-3 may be regarded as a good candidate.

2.4. Analysis from Field Synergy Principle

From the traditional viewpoint, strips on the fin surface can enhance convective heat transfer because they can interrupt the flow to reduce the thermal boundary-layer thickness by repeatedly re-creating the thermal boundary layers, or can increase the disturbance in the flow field. But in fact, all these functions come from the same source: the reductions of the synergy angle between velocity and temperature gradient, which have been proved numerically in [10, 11]. Here we once again examine the inherent relation between heat transfer enhancement and the synergy of velocity and temperature gradient. The comparisons of the domain-averaged synergy angle of 15 types of fin surface against the frontal oncoming velocity are presented in Figures 4*d*, 5*d*, 6*d*, and 7*d*. From these figures, the following features may be noted. First, the results show that the average synergy angles increase with an increase of the frontal oncoming velocity, which indicates that the synergy between velocity and the temperature gradient becomes worse with increasing velocity. This can explain why with the increase in flow rate the convective heat transfer rate does not increase linearly, and the larger the velocity and Reynolds number, the less significant is the increase in heat transfer rate (see Figures 4*c*, 5*c*, 6*c*, and 7*c*). Second, anyone of the 15 slotted fins always have lower averaged synergy angle than that of the plain plate fin. Third, it is especially interesting to note that, for every group, the averaged synergy angle of the fin with the highest heat transfer rate always has the lowest synergy angle among the slit fins compared. Thus we can conclude that the function of strips on the slotted fin surface is to improve the synergy between the velocity and the temperature gradient.

2.5. Discussion

It is quite clear that if the ratio of j/f or the heat transfer under identical pumping power at low oncoming velocity ($\sim$1 m/s) is the only purpose pursued, then the plain plate fin is the first choice. Of course, this by no means implies that the slotted fin surface is not helpful. Rather, considering the following two ingredients, the slotted fin surface with X-shaped strips positioned according to the "front coarse and rear dense" principle should generally be recommended. The first ingredient is that at the same oncoming velocity the heat transfer rate of the plain plate fin is less than that of the slotted fin surface. The adoption of a plain plate fin implies an increase in the heat transfer surface and the volume of the heat exchanger. The second ingredient is that, beyond the tuning point, the slotted fin surface always behaves better than the plain plate fin, either from the ratio of j/f, from heat transfer under identical pumping power, or from heat transfer under the same flow rate. Moreover, among all 15 slotted fin surfaces, slit fin D1 and slit fin D2 have the highest j/f ratio (almost equal to that of the plain plate fin) at low oncoming velocity and the lowest turning Re and heat transfer under identical velocity, but their average heat transfers within the range 1–3 m/s are still 11% and 10% higher, respectively, than that of the plain plate fin surface.

As far as increasing the j/f ratio is concerned, the slotted fins with strip length variation and decreasing strips number both have some advantage; however, strip length variation seems to be more efficient. Strip length variation can make the j/f ratio higher than that of its counterpart in number variation, but the reduction in heat transfer rate under identical velocity is smaller than that of the number variation and their turning Re numbers are appreciably less than that of number variation. In addition, the slotted fins D1 and D2 (the second series of the second group), behave greatly superior to their original counterparts (slit fin 1 and slit fin 2), can make the j/f ratio as high as 99.7% and 99.4% of that of plain plate fin at the oncoming velocity of 1 m/s, and their turning Re numbers drop to the least among all the slotted fin surface. However, their heat transfer rates under identical flow rate also drop the most. Thus, for the X arrangement of strips with the "front coarse and rear dense" principle, the technique of the first series of the second group (strip length variation) is a more useful way to further enhance the ratio of j/f of the slit fin surface. To increase the heat transfer rate under identical pumping power for low oncoming fluid velocity (~1 m/s), both strip length variation and strip number variation offer some advantage, but the technique of strip length shortening is most efficient.

To sum up, under the two general guidelines for the positioning of strips, i.e., "front coarse and rear dense" and "X-shape arrangement," we tried four techniques to further enhance the slit fin performance. These are changing the number of strips, shortening the strip length, changing the location of the strips, and splitting one strip into two. It has been found that the location of strips is the most important factor affecting heat transfer and flow characteristics, followed by strip length and strip number. Splitting one strip into two is the least effective strategy.

3. CONCLUSIONS

In this article, the air-side heat transfer and pressure drop characteristics of 15 types of slotted fin and a plain plate fin have been investigated systemically using a three-dimensional steady laminar model within Re ranging from 901 to 2702. The slotted fin surfaces have an X arrangement of strips positioned with the "front coarse and rear dense" principle. The predicted results of Colburn j factor and pressure drop for the plain plate fin agree well with the tested data available in the literature. The numerical results have been compared for j/f under identical flow rate and for heat transfer rate under identical pumping power. Analysis has also been conducted from the field synergy principle. The major findings can be summarized as follows.

1. For all 15 slotted fin surfaces, their j/f versus $(\mathrm{Re})_{\mathrm{idfr}}$ curves cross with the curve of the plain plate fin surface at some turning Reynolds number. When the Reynolds number is below this turning point, their ratio of j/f is less than that of the plain plate fin surface, while beyond this turning point their j/f values are all greater than that of the plain plate fin surface. The Q versus $(\mathrm{Re})_{\mathrm{idpp}}$ curves have similar character, with the turning Reynolds number being appreciably less than the corresponding turning Reynolds number for the j/f versus $(\mathrm{Re})_{\mathrm{idfr}}$ curve.

   We are seemingly the first in the literature to confirm the fact that for the slotted fin surface there exits a turning Reynolds number below which its performance is worse than that of the plain plate fin either from j/f ratio or from the heat transfer rate under identical pumping power condition.
2. The comparison of heat transfer rate under identical flow rate shows that all the 15 slotted fin surfaces behave better than the plain plate fin surface; and the larger the oncoming velocity, the more significant is the advantage of the slotted fin surfaces. For example, the heat transfer rates of slit 3 are 6.0% and 23.1% higher than that of the plain plate fin at oncoming velocities of 1 and 3 m/s, respectively.
3. The higher heat transfer rate corresponds to a lower average synergy angle for any slotted fin surface studied. This finding once again demonstrates that the field synergy principle reveals the fundamental mechanism for enhancing convective heat transfer and is a very useful tool in developing new types of enhanced heat transfer structure.
4. For the two-row plate fin-and-tube heat exchanger, slit fin A1-3 and slit fin 3 with five strips are recommended for the use in air-conditioning.

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