OPTIMIZATION OF A MICRO-CHANNEL STRUCTURE OF AN ELLIPTICAL LOUVER FIN

by

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The influence of the louver fin's structure on the heat transfer performance of a parallel flow gas cooler is studied, and a 3-D model for an elliptical louver fin is simulated for analysis of the heat transfer and flow resistance characteristics of the fin. The micro-channel structure of the fin is optimized to give the best comprehensive performance evaluation by suitable choice of fin's thickness and the space between the adjacent louvers for given range of Reynolds number.

Key words: *louver fins, heat transfer factor, resistance factor, comprehensive performance evaluation*

Introduction

In recent years, a series of problems such as energy security and environmental pollution are brought due to the rapid development of Chinese traditional automobile industry, which has seriously affected the sustainable development of the national economy. A parallel flow gas cooler is a large energy consuming device which is the key component of the car air-conditioning. The efficiency of automotive air-conditioning is severely restricted by the pros and cons of heat transfer and resistance characteristics on parallel flow gas coolers. There are many influencing factors on the parallel flow heat exchanger. The complex structure and diversity of parameters of the louver fins affect greatly the heat exchange efficiency and flow resistance of the fins, and much work has been done in this direction. Ferrero *et al.* [1] established 40 louver fins models with different structural parameters, and the theoretical analysis reveals that its structural parameters play an important role in the heat transfer performance of the heat exchanger, however, there are no general structural parameters which will make the best performance of various heat exchangers. Wang *et al.* [2] studied the heat transfer performance of louver fins heat exchangers with different geometric parameters, such as the space of fins, louver height, the angle of fin, *etc.*, the complex relationship between the heat transfer factors and friction coefficients are summarized. Dong *et al.* [3] studied and analyzed the length and space between the adjacent louvers. The results showed that the two factors have a great influence on the heat transfer coefficient and pressure drop of the heat exchanger. Wu *et al.* [4] analyzed the heat transfer performance and fluid-flow in louver fins with continuous

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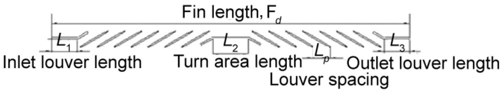

Figure 1. The structure of elliptical louver fins

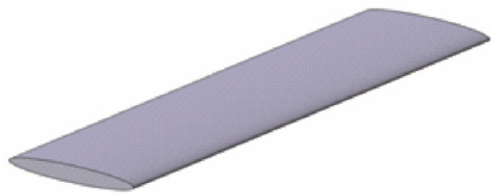

Figure 2. The configurations of elliptical louver fin

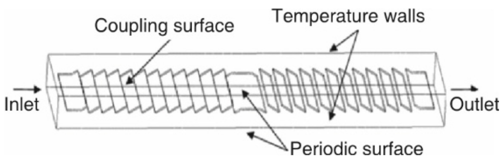

Figure 3. The boundary conditions of elliptical louver fins

variable angle of attack. It is found that there are two continuous variable angles of attack for louver fins structures with good comprehensive performance in a suitable Reynolds number range.

In this paper, the flow resistance and heat transfer performance of the elliptical louver fins channel are studied for different geometric parameters of the fin, especially the thickness of the fin, δ, and the space between the adjacent louvers, L_P, to explore the relationship between heat transfer and flow resistance characteristics of the elliptical louver fin.

Mathematical model

Geometric model

The structural model of the elliptical louver fin is shown in fig. 1. The structural parameters of the model are shown in tab. 1. The structural features of the fin are: the connection between the two ends of the flat tube is elliptical wall, the louver fins are thin at both ends, the middle is thick and symmetrical, and the fluid-flows through the louver fin to form an arc shape.

A partial enlarged view of an elliptical louver fin is shown as fig. 2. The cross-sectional view of the elliptical louver fin is an elliptical shape. When the fluid-flows through the fins, the transition will be smoother than the rectangular louver fin, and resistance will also decrease to some extent.

Table 1. The basic structure data of model

Parameter name	Parameter value	Parameter name	Parameter value
Louver space, L_P, [mm]	1.4	Louver angle, θ, [°C]	30
Louver length, L_L, [mm]	6.4	Inlet louver length, L_1, [mm]	1.4
Louver height, F_L, [mm]	7.4	Turn area length, L_2, [mm]	2.0
Fin length, F_d, [mm]	36	Outlet louver length, L_3, [mm]	1.4

Computational model and boundary conditions

In order to simplify the model and reduce the amount of computation time, the boundary conditions of the simplified model are shown in fig. 3. Boundary conditions include fluid domain, solid domain, velocity inlet, and pressure outlet. There is a postulation that the fluid in micro-channel structure of the fin flows periodically along flow direction, and takes periodic model as a simplification of whole heat exchanger, so the top and bottom surfaces of the model are set to be periodic.

As shown in fig. 3, the inlet temperature T_i = 300 K, the solid domain is mainly composed of fins and the flat tubes. For calculation purposes, the surface of flat tube is set to be constant wall temperature. The SIMPLE algorithm is used for the pressure-velocity coupling in the control equations. The two-order upwind difference scheme is used for both the momentum equation and the energy equation. The heat conduction between wall surface and the convection heat transfer among the fins is coupled.

Data processing

In this paper, a 3-D model is used to simulate and calculate the air side heat transfer and resistance performance of rectangular and elliptical fin channels at different Reynolds numbers. The relevant data is calculated as follows:

$$\mathrm{Re} = \frac{\rho u d_\mathrm{e}}{\mu}, \quad h = \frac{\dot{m} c_p (T_\mathrm{o} - T_\mathrm{i})}{A \Delta T_\mathrm{LMTD}} \tag{1}$$

Among them, the logarithmic average temperature difference ΔT_LMTD is defined by eq. (2):

$$\Delta T_\mathrm{LMTD} = \frac{(T_\mathrm{w} - T_\mathrm{o}) - (T_\mathrm{w} - T_\mathrm{i})}{\ln \dfrac{(T_\mathrm{w} - T_\mathrm{o})}{(T_\mathrm{w} - T_\mathrm{i})}} \tag{2}$$

The heat transfer coefficient factor, j, and resistance factor, f, are calculated by eq. (3):

$$j = \frac{h}{\rho u c_p} \mathrm{Pr}^{2/3}, \quad f = 2 \frac{\Delta P}{\rho u^2} \frac{d_\mathrm{e}}{L} \tag{3}$$

Among them, the pressure drop, ΔP, is calculated by eq. (4):

$$\Delta P = P_\mathrm{i} - P_\mathrm{o} \tag{4}$$

The comprehensive evaluation factor, e, is defined in eq. (5), a greater value of e implies a higher comprehensive heat transfer performance [5].

$$e = \frac{j}{f^{1/3}} \tag{5}$$

Grid independence verification

The quality of the grid has a very important influence on the accuracy of the numerical simulation. This paper gives five different meshing methods for louver fin, it is the base of further study on micro-channel by comparing and analyzing the heat transfer factor, j, and resistance factor, f, through different meshing method. The two factors j and f are used to test the independence of the grid when the wind speed is 2.8 m/s, the grid number and deviation are shown in tab. 2. It can be seen from the table that when the number of grids is 1600800, the values of the heat transfer factor, j, and the resistance factor, f, are basically no longer changed. In order to balance the calculation accu-

Table 2. Mesh independence test results

Number of grids	j	Deviation of j [%]	f	Deviation of f [%]
510895	0.0258		1.0110	
674226	0.0269	4.08	1.0149	0.3
1019590	0.0287	6.27	1.0217	0.6
1600808	0.0319	10.03	1.0227	0.09
2317242	0.0323	1.23	1.0233	0.05

racy and speed, the number of elliptical louver fin grids is 1600800, the grid's independence verification satisfies the requirements.

Verification of calculation results

In order to verify the correctness of the simulation in this paper, the experimental results given in [6] are used to verify the simulation result of elliptical louver fins. The heat transfer factor, *j*, and resistance factor, *f*, values are selected and calculated according to eq. (3), the results are shown in figs. 4 and 5.

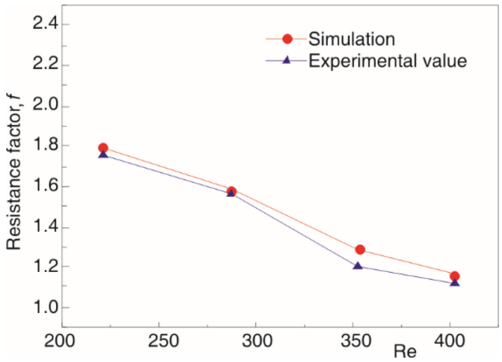

Figure 4. Comparison of resistance *f* factors

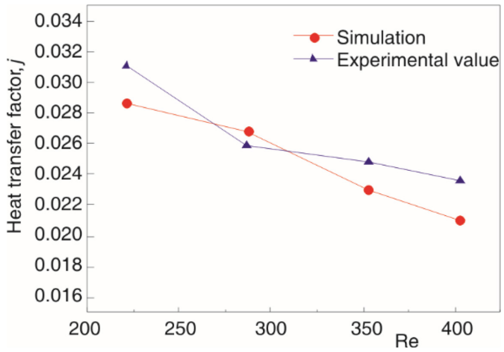

Figure 5. Comparison of heat transfer *j* factors

The experimental and numerical simulation results of the resistance factor, *f*, and the heat transfer factor, *j*, as a function of the Reynolds number are shown in figs. 4 and 5. It can be seen from the figures that there is no significant difference between experimental data and simulative data, a good agreement is observed. The error of the resistance characteristic factor, *f*, is less than 13%, and the heat transfer characteristic, *j*, factor is less than 8%.

Results analysis

Effect of louver fin thickness

In this paper, the Reynolds number ranges from 225.7 to 451.3, and the thickness of the louver fins is 0.06 mm, 0.1 mm, 0.2 mm, and 0.3 mm, respectively, the heat transfer and flow resistance characteristics of the elliptical louver fins are studied and analyzed.

The curves of resistance factor, *f*, of the elliptical louver fins with Reynolds number at different louver thickness are shown in fig. 6. It can be seen from the figure that the resistance factor, *f*, decreases with the increase of the Reynolds number, but increases with the thickness of the louver fins. When δ = 0.06 mm, the factor *f* is the smallest; when δ = 0.3 mm, the factor f is the maximum. The deviation of factor *f* is 31%, 32.96%, 34.36%, and 35.25% in the range of Reynolds number, respectively. The reason for this phenomenon is that as the thickness of the louver fins increases, the louver space decreases, and the flow area when the air passes through decreases accordingly, so the resistance factor, *f,* becomes larger.

The curves of heat transfer factor, *j,* of the elliptical louver fins with the different Reynolds numbers at different louver thickness are shown in fig. 7. It can be seen from the figure that the heat transfer factor, *j,* decreases with the increase of the Reynolds number. If the other structural parameters of fins are not changed, and the Reynolds number is in the range of this paper, δ = 0.06 mm has the smallest heat transfer factor, *j,* when the Reynolds number Re = 451.3, and δ = 0.1 mm has the largest heat transfer factor, *j*, the difference between them is 6.9%, 7%, 7.9%, and 10.3%, respectively. This shows that appropriate increase of the thickness of the louver fins is beneficial to enhance the heat transfer effect.

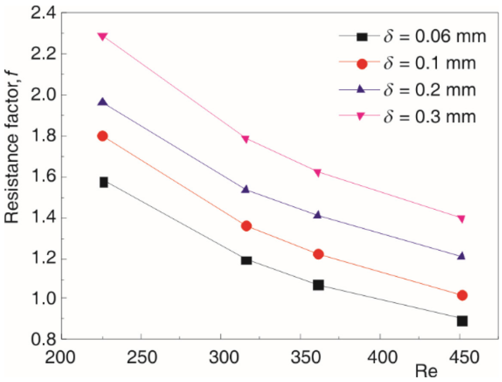

Figure 6. Influence of louver fins thickness on resistance factor, *f*

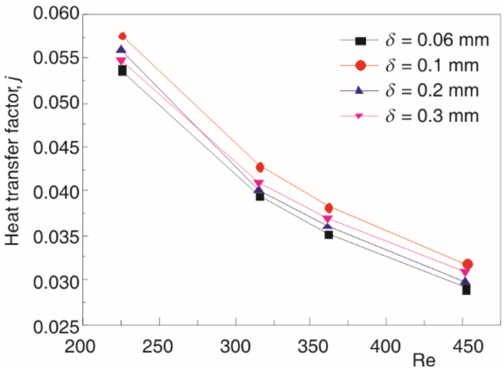

Figure 7. Influence of louver fins thickness on heat transfer factor, *j*

The reason is that the space between the adjacent louvers will be reduced by increasing the thickness of the fins, it will increase the flow velocity when the air-flows through the channel, its surface will be compressed and the boundary-layer will be destroyed, so the thickness of the boundary-layer becomes thinner, it can enhance the heat transfer effect. We can conclude that the increase in the heat transfer effect of the louver fins comes at the cost of increased resistance. Therefore, it is necessary to comprehensively evaluate the heat transfer performance of the louver fins.

The curves of the comprehensive performance factor of an elliptic louver fin in different thickness are shown in fig. 8. It can be seen from the figure that the change trend of the comprehensive heat transfer performance factor curve is basically the same, and the comprehensive performance factors of different thickness decrease with the increase of the Reynolds number. When δ = 0.1 mm, the comprehensive performance of the elliptical louver fins is the best, but when δ = 0.3 mm, the comprehensive performance is the worst, the difference ranges from 23.4% to 28.13%. Therefore, it can be concluded that as the thickness of the fins increases, the comprehensive performance of the elliptical louver fins becomes better. If the thickness of the fins continues to increase, the comprehensive performance will decrease, so when the fin thickness is δ = 0.1 mm, the elliptical shape louver fins has the best comprehensive performance.

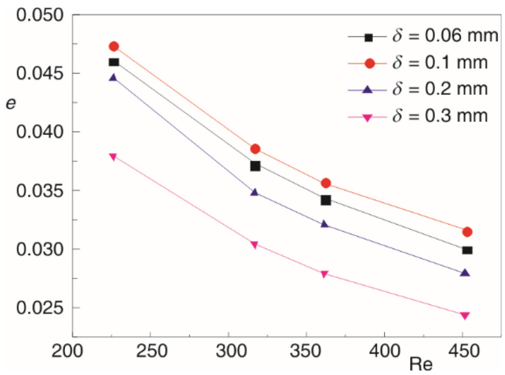

Figure 8. Influence of louver fins thickness on the comprehensive performance

Influence of louver space

The Reynolds number ranges from 225.7 to 451.3, louver fin thickness, δ, is 0.1 mm (the previous calculation result), louver fin space is 0.9 mm, 1.1 mm, 1.3 mm, 1.5 mm, and

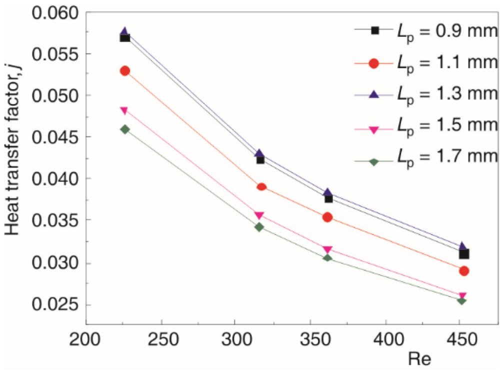

Figure 9. Influence of the space of louver on heat transfer factor, *j*

1.7 mm, respectively, the elliptical louver fins heat transfer and flow resistance characteristics are researched and analyzed.

The variation curves of the heat transfer factor, *j*, of the louver fins with the Reynolds number at different louver spaces are shown in fig. 9. It can be seen from the figure that the heat transfer factor, *j*, gradually decreased as the Reynolds number increases, and the change tendency of the different louver fin space of the heat transfer factor, *j*, is basically the same, and within the calculated Reynolds number range, when the L_P = 1.3 mm, the comprehensive heat transfer performance of the elliptical louver fins is best, but when L_P = 1.7 mm, the result reverses, the difference between the two curves reaches a maximum of 26.09%. Moreover, the values of the heat transfer factor, *j*, are very close and the change is small when the L_P is 1.3 mm and 0.9 mm.

The variation curves of the resistance factor, *f*, of louver fins with the Reynolds number at different louver space are shown in fig. 10. It can be seen from the figure that the resistance factor, *f*, increases with the increase of the Reynolds number. When the louver distance L_P is 1.3 mm, 1.5 mm, and 1.7 mm, respectively, the resistance factor, *f*, is very close to each other, when L_P = 0.9 mm and L_P =1.1 mm, the values of the resistance factor, *f*, are quite different within the range of calculation, the difference range is 10.95%~16.13%. The maximum difference between 0.9 mm and 1.7 mm is 28.69%. We can see that when the distance increases to 1.3 mm, the resistance performance curve is basically stable. If we continue to increase the louver space, the resistance factor, *f*, values are not significantly difference.

The curves of the comprehensive performance factor of the elliptical louver fins at different louver space are shown in fig. 11. It can be seen from the figure that the comprehensive performance factor curves of louver space decreases with the increase of Reynolds number, and the change trend is consistent. The comprehensive evaluation factor is the best when the L_P = 1.3 mm, but the comprehensive evaluation factor is the worst when the L_P = 1.7 mm, the difference between Reynolds number ranges from 225.7 to 451.3 is 14.89%, 17.95%,

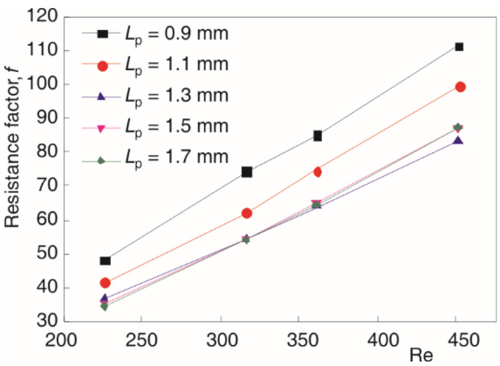

Figure 10. Influence of the space of louver on resistance factor, *f*

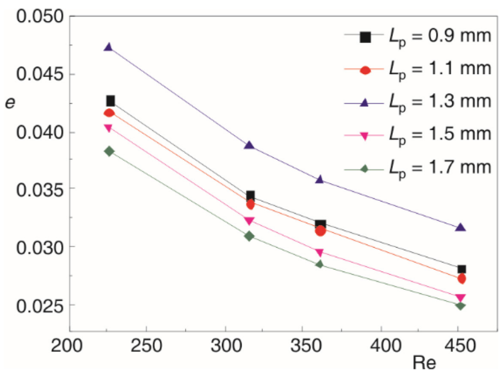

Figure 11. Influence of the space of louver on comprehensive performance

16.67%, and 18.75%, respectively. It can be concluded that the comprehensive evaluation of the elliptical louver fins has the best performance when $L_P = 1.3$ mm.

Conclusions

In this study, the performance of heat transfer and flow resistance of the elliptical louver fins are studied and analyzed numerically. The 3-D simulation models are well validated by experimental data from open literature. The influences of the louver thickness and space on the louver fin performance are investigated at different Reynolds numbers ranging from 225.7 to 451.3. The analysis and discussion of the numerical simulation give the following conclusions.

- The numerical simulation results of the heat transfer factor, j, and the resistance factor, f, are contrasted with the experimental data. It is found that the errors for the heat transfer factor, j, and the resistance factor, f, are 6% and 11%, respectively, all results within the allowable range of error. It is proved that the simulation method in this paper is correct and credible.
- The effects of louver thickness and louver space on the heat transfer performance and flow resistance are analyzed systematically. Finally, the micro-channel structure of the fin is optimized to give the best comprehensive performance evaluation. The results show that the best comprehensive performance is achieved when the Reynolds number ranges from 225.7 to 451.3, the fin thickness is 0.1 mm and the louver space is 1.3 mm.

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Nomenclature

A – cross section, [m^2]
c_p – specific heat at constant pressure, [kJkg^{-1}K^{-1}]
d_e – equivalent diameter, [m]
h – heat transfer coefficient, [Wm^{-2}K^{-1}]
L – length, [m]
$\dot{m}$ – mass-flow rate of fluid, [kgm^{-2}s^{-1}]
P – pressure, [Pa]
Pr – Prandtl number, [–]
ΔP – pressure drop, [Pa]
Re – Reynolds number, [–]
T_w – wall temperature, [K]
ΔT – temperature drop, [K]
u – velocity, [ms^{-1}]

Greek symbols

ρ – density, [kgm^{-3}]
μ – dynamic viscosity of the fluid, [Pa·s]

Subscripts

i – intlet
o – outlet
w – wall

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