INFLUENCE OF FIN THICKNESS ON HEAT TRANSFER AND FLOW PERFORMANCE OF A PARALLEL FLOW EVAPORATOR

by

Haijun LI[a*], Enhai LIU[a], Guanghui ZHOU[a], Fengye YANG[a], Zhiyong SU[a], Yunting XU[b], Chenguang JIAN[a], and Shuailing SHI[a]

[a] School of Energy and Environmental, Zhongyuan University of Technology, Zhengzhou, China
[b] Henan Zhongke Architectural Design and Planning co., LTD, Zhengzhou, China

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This paper studies numerically the influence of the louver's fin thickness on heat transfer and flow performance of a parallel flow evaporator, a comprehensive evaluation and analysis of the five structures at different Reynolds numbers are systematically carried out. Comparison of the numerical results with the experimental data shows good agreement with maximal errors of 12.16% and 5.29% for the heat transfer factor and the resistance factor, respectively. The results show that the heat transfer coefficient and the pressure drop increase with the increase of the thickness of the louver fins when the Reynolds number is a constant. The analysis of the comprehensive evaluation factor shows that the A-type fin is the best, and it can effectively strengthen the heat exchange on the air side and improve the heat transfer capacity of the system. The research results can provide reference for the structural optimization of the louver fins.

Key words: *automotive air conditioning, parallel flow evaporator, louver fin, numerical simulation*

Introduction

With the fast development of refrigeration and air conditioning industry, the parallel flow heat exchanger has been widely used in automotive air conditioning, commercial air conditioning and other refrigeration and air conditioning fields because of its good heat transfer performance, compact structure and many other advantages.

At present, many scholars have made a lot of research on the factors affecting the heat transfer on the air side and flow characteristics of the heat exchanger, and have put forward the measures to strengthen the heat transfer performance of the louver fin and have optimized its structure [1-3]. Yang *et al.* [4] carried out a 3-D numerical simulation through four kinds of louver fins with variable angles, indicating that the overall performance of louver fins with four variable angles is better than that with uniform angle. Kim *et al.* [5] carried out experimental study of 45 kinds of louver fins with different louver angles and fin spacings, and obtained the experimental correlation of heat transfer and friction factor.

In recent years, the study on optimization of the heat transfer of the louver fin structure has been mainly focused on the width of the louvers, the angle of louvers, the spacing of fins and louvers. However, there is little report on the study of the influence of louver fin thickness on heat transfer and flow performance. This study is based on the multivariate paral-

* Corresponding author, e-mail: 13653866002@126.com

lel flow evaporator inside an automotive air-conditioning, and takes its louver fin as the object of study. The 3-D numerical simulation of the evaporator's louver fin model is carried out by using ANSYS15.0 software, the influence of the louver fin thickness on the fin heat transfer and flow performance of parallel flow evaporator is studied, the heat transfer and flow performance of the five structures at different Reynolds numbers, Re_{LP}, are analysed and compared, and the louver fin thickness with the best overall performance is obtained. This is important for the structural optimization of parallel flow heat exchangers.

Model establishment

Physical model

Figure 1 is the physical model of the louver fins established for the parallel flow evaporator inside the automobile with the size of 930 mm × 597 mm × 36 mm. The louver fin structure of the parallel flow evaporator is listed in tab. 1.

The 3-D numerical simulation of the louver fins with different thicknesses of 0.06 mm, 0.08 mm, 0.10 mm, 0.12 mm, and 0.14 mm, respectively, is carried out by using software ANSYS15.0, the five samples are referred as A-type, B-type, C-type, D-type, and E-type fins, respectively.

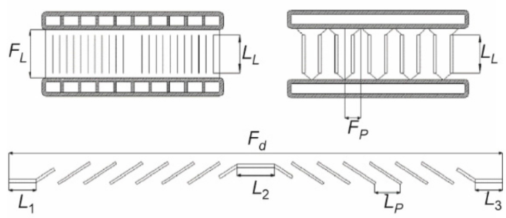

Figure 1. The structure of the louver fins

Table 1. Basic structural parameters of the model

Parameter name	Parameter value	Parameter name	Parameter value
Fin spacing, F_P, [mm]	1.4	Louver angle, θ, [°]	30
Fin length, L_L, [mm]	6.4	Inlet fin length, L_1, [mm]	1.4
Fin height, F_L, [mm]	7.4	Steering zone length L_2, [mm]	2.0
Fin width, F_d, [mm]	36	Outlet fin length L_3, [mm]	1.4

Governing equations

The governing equations include the continuity equation, the momentum equation, and the energy equation. Considering that the temperature change along the length of the louver fins is small, it is considered that the air side of the parallel evaporator is a steady 3-D incompressible and laminar flow, the basic governing equations are:

– Continuity equation

$$\frac{\partial \rho}{\partial t}+\frac{\partial}{\partial x_i}(\rho u_i)=S_m \tag{1}$$

– Momentum conservation equation

$$\frac{\partial}{\partial t}(\rho u_i)+\frac{\partial}{\partial x_i}(\rho u_i u_j)=\frac{\partial p}{\partial x_i}+\frac{\partial \tau_{ij}}{\partial x_j}+\rho g_i+F_i \tag{2}$$

– Energy equation

$$\frac{\partial}{\partial t}(\rho E)+\frac{\partial}{\partial x_i}[u_i(\rho E+P)]=\frac{\partial}{\partial x_i}\left(k_{\text{eff}}\frac{\partial T}{\partial x_i}-\sum_{j'}h_{j'}J_{j'}+u_j(\tau_{ij})_{\text{eff}}\right)+S_h \tag{3}$$

In the equations, S_m is the source item, P – the static pressure, τ_{ij} – the stress tensor, ρg_i and F_i – the weight volume force and the external volume force in the direction of i, respectively, k_{eff} – the effective heat transfer coefficient, $J_{j'}$ – the diffusion flow of combination j', and S_h for the volume heat source item.

Calculation model and boundary conditions

The calculation domain is established based on the length F_d, the spacing F_P and the height F_L of the louver fins. The calculation domain is composed of a fluid domain and a solid domain. The fluid domain of the model is the air flow path on the air side of the parallel flow evaporator. The model's louver fin wall and the flat wall are solid domains, and the simplified 3-D calculation model of the louver fin is as shown in fig. 2.

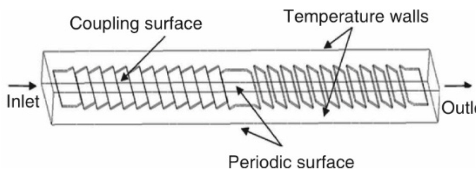

Figure 2. The boundary conditions of the louver fins

As shown in the model boundary conditions in fig. 2, the left inlet is set as the condition for velocity to flow into the boundary, the inlet temperature, *T,* is equal to 300 K. The outlet is set as the pressure outlet boundary condition. The upper and lower side of the fluid domain perpendicular to the flow direction are set as the periodic boundary condition. The front and back sides of the fluid domain are the boundary conditions for the constant temperature wall. The tube wall and fin wall are constant temperature walls with temperature of 286 K, and the contact surface between the fluid and the fin is the coupling surface.

Data processing

The 3-D model is established by using ANSYS15.0 software. The heat transfer and resistance of the louver fins with five different thicknesses in tab. 1 are analysed when the Reynolds number, Re_{LP},is equal to 273. This paper uses the following formula for data processing:

$$\text{Re}_{LP}=\frac{\rho_{\text{a}}u_{\text{a}}d_{\text{e}}}{\mu} \tag{4}$$

$$\text{Nu}=\frac{h_{\text{a}}d_{\text{e}}}{\lambda_{\text{a}}} \tag{5}$$

In the formulas, d_{e} [m] is the equivalent diameter of the model, u_{a} [ms^{-1}] – the velocity, ρ_{a} [kgm^{-3}] – the density, λ_{a} [Wm^{-1}K^{-1}] – the air thermal conductivity coefficient, and h_{a} [Wm^{-2}K^{-1}] – the heat transfer coefficient of the air side on dry surface. The expression of the thermal factor *j* reads:

$$j = \frac{h_a}{\rho_a u_a C_p} \mathrm{Pr}^{2/3} \tag{6}$$

In the formula, C_p [Jkg^{-1}K^{-1}] is the specific heat at constant pressure, Pr is 0.7 in our experiment. The calculation method of pressure drop Δp is:

$$\Delta p = P_i - P_o \tag{7}$$

where P_i is the pressure at the air inlet and P_o is the pressure at the air outlet. The calculation method of resistance factor f is:

$$f = 2\frac{\Delta p}{\rho_a u_a^2}\frac{d_e}{L} \tag{8}$$

where d_e [m] is the equivalent diameter of the model and L [m] – the length of the calculated domain. The weight of the louver fins is:

$$m_1 = Nn\rho V \tag{9}$$

In the formula, N is the row number of louver fins n – the number of louver fins on a single row, ρ [kgm^{-3}] – the density of the aluminium fins of the louver, and V [m^{-3}] – the volume of the louver fins. The cold-to-weight ratio is:

$$\psi = \frac{Q}{m_1 + m_2} \tag{10}$$

where m_1 [kg] is the weight of the parallel flow evaporator's louver fins, m_2 [kg] – the weight of the parallel flow evaporator flat tube and the collecting pipe, and Q [kW] – the heat transfer capacity of the parallel evaporator.

Result analysis

Model validation

The verification model of calculation correlation of heat transfer factor, j, given by Chang *et al.* [6] and calculation correlation of resistance characteristic factor, f, given by Davenport [7] are used to fit with the curves of heat transfer factor, j, and resistance characteristic factor, f, with the change of Reynolds number, Re_{LP}, the numerical simulation value is compared with the empirical correlation to verify the accuracy of the simulation results.

Figures 3 and 4 are, respectively, the curves of heat transfer factor, j, and resistance characteristic factor, f, with the change of Reynolds number. It can be seen from figs. 3 and 4 that the variation trend of the calculation simulation results is consistent with that of the empirical correlation given in [6, 7]. By comparison, the maximum error of heat transfer factor, j, is 12.16% and the maximum error of resistance factor, f, is 5.29%, both less than 15%, within the allowable error range of the project. Thus, the reliability of the simulation method is proved.

Analysis of simulation results

According to the numerical results given in fig. 5, with the increase of Re_{LP}, the heat transfer coefficients of five type structures increase. Figure 6 shows the pressure drop of

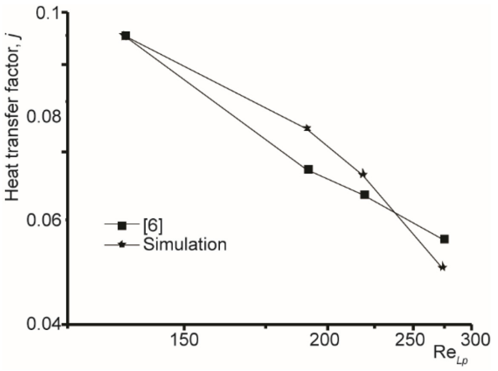

Figure 3. Comparison of the numerical result of the heat transfer factor with that by Chang *et al.* [6]

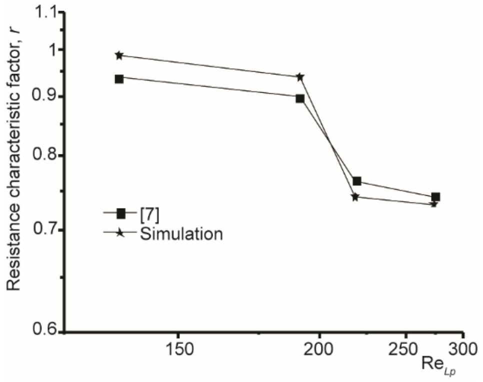

Figure 4. Comparison of the numerical result of the resistance characteristic factor with that by Davenport [7]

the five fins from A-type to E-type structures increases with the increase of Re_{LP}. From the perspective of cold-to-weight ratio index of energy-saving air conditioning and production cost, the A-type structure among five fins is the best.

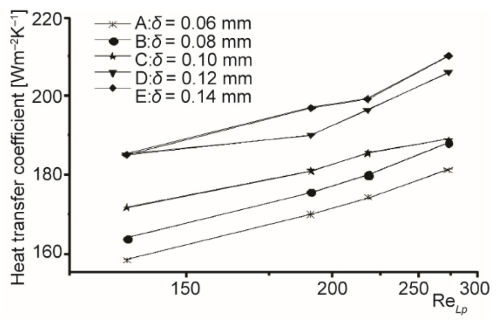

Figure 5. Heat transfer coefficients of five kinds louvered fins

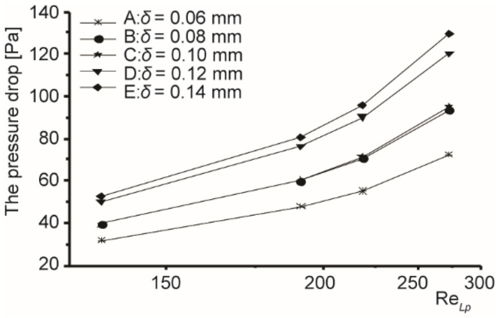

Figure 6. The pressure drop of the five kinds of louver fins

It can be seen from fig. 7, as the Reynolds number increases, the cold-to-weight ratio of the fins increases. The thicker the fin, the smaller the cold-to-weight ratio. From the perspective of cold-to-weight ratio index of energy-saving air conditioning and production cost, the A-type structure among five fins is the best.

Analysis of comprehensive performance

When the Reynolds number is 136, 191, 218, and 273, respectively, the comprehensive performance evaluation factor, E_{JF}, of the louver with five different thicknesses is calculated, and the comprehensive performance of five kinds of louver fin thicknesses is compared. In order to better evaluate the heat transfer and flow performance of the evaporator, the comprehensive performance evaluation factor E_{JF} is defined, which describes the heat transfer capacity of the louver fins under the same resistance. The larger the value, the better the heat transfer capacity of the louver fins [8].

The comprehensive performance evaluation factor E_{JF} is:

$$E_{\mathrm{JF}} = \frac{j}{f^{1/3}} \tag{11}$$

In the formula, j is the heat transfer factor and f – the resistance characteristic factor.

Figure 8 shows the curve of the comprehensive evaluation factors of the louver fins with 5 different thicknesses changing with the Reynolds numbers. It can be seen from the figure that the enhanced heat transfer effect of the louver A-type fin is the best, the enhanced heat transfer effects of the louver B-type fin and C-type fin are the closest, and the enhanced heat transfer effects of the louver E-type fin is the worst. It can be explained that the louver of A-type fin has the most significant enhanced heat transfer effect.

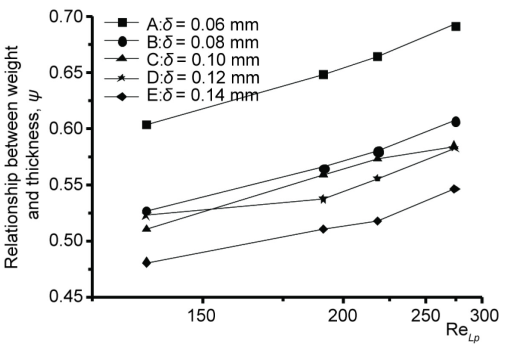

Figure 7. Relationship between weight and thickness of louver fins

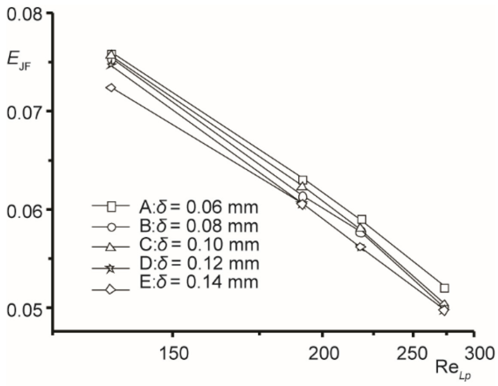

Figure 8. Curve of louver fin E_{JF} changing with Re_{LP}

Conclusion

After the model is verified, the heat transfer factor, j, is consistent with the variation trend of the resistance characteristic factor, f, the maximal error of the heat transfer factor, j, is 12.16%, the maximal error of the resistance characteristic factor, f, is 5.29%, and both errors are less than 15%, within the allowable range of the project. With the increase of Reynolds number, the heat transfer coefficient and pressure drop of the louver fins in five kinds of structures increase. When the Reynolds number, is constant, the heat transfer coefficient and pressure drop increase with the increase of thickness of the louver fins.

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