論 文 の 内 容 の 要 旨

論文題目　Heat transfer enhancement in a parallel, finless heat exchanger
（フィンレス熱交換器における熱伝達向上に関する研究）

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For reducing the refrigerant charge amount in residential air conditioner system, therefor the low GWP but flammable refrigerant can be used safely, the aluminum parallel heat exchangers are supposed to be adopted instead of the conventional tube-fin ones. In this dissertation, the research focus on three aspects of the parallel aluminum heat exchanger. The first part is the up-flow boiling performance including heat transfer coefficient and pressure drop in aluminum flat tubes with various cross sections. The second part is the air side heat transfer coefficient and pressure drop of parallel heat exchangers under dry, wet and frosting/defrosting conditions. The fin-tube parallel heat exchanger may has deficiency on drainage, therefore we proposed the finless heat exchanger, whose heat transfer performance was further enhanced by a vortex generator. And the third part is a new design of manifold for parallel heat exchangers, which is expected to enhance the distribution uniformity of two phase flow among parallel flat tubes. The summary of each part is shown below.

1. Refrigerant side

The characteristics of local heat transfer and pressure drop were investigated experimentally for the vertical up-flow boiling of refrigerant R1234yf in two types of aluminum multi-port extruded tubes having 16 channels with a cross-section of 0.91 × 0.21 mm (tube A) and 40 channels with a cross-section of 0.34 × 0.21 mm (tube B). The heat transfer performance was compared with that of multi-port extruded tubes having 16 channels with a cross-section of 0.9 × 0.9 mm (tube C), and the pressure drop was compared with calculation results of three different correlations. From the results, the following conclusions can be drawn.

a. Under low heat and mass flux conditions, the heat transfer coefficient almost linearly decreased as a function of vapor quality in tubes A and B. As the vapor quality increased, the area of dry patches became lager within these rectangular channels, which deteriorated the heat transfer performance dramatically.

b. The channel dimensions significantly influenced the heat transfer performance. At a mass flux of 60 kg/m^{2}s, tube C showed better heat transfer performance than tubes A and B for all measured vapor quality regions because the shape of tubes A and B is not square, so the dry-out occurs more easily on the wider surfaces, thereby compromising the heat transfer performance. When the mass flux increased to 120 kg/m^{2}s, the dry-out was alleviated, enabling better performance in low vapor quality regions. In high vapor quality regions, the dry patches reoccupied the heat transfer area, leading to reduced heat transfer performance in tubes A and B. Moreover, tube A was worse than tube B because of its lower aspect ratio.

c. The correlations of Lee−Lee, Lockhart−Martinelli, and Mishima−Hibiki did not precisely predict the experimental pressure drops of tube A and B. Relatively, the Lockhart−Martinelli correlation showed better prediction. The data trend of decreasing heat transfer coefficient with increasing vapor quality was well predicted by the Saitoh et al. correlation and the Agostini−Bontemps correlation, but the value shows large deviation.

2. Air side

Firstly, the research focuses on the performance of fin-tube parallel heat exchangers, as shown figure 1a. The air-side pressure drops and overall transport coefficients of three louvered-fin all-aluminum parallel multi-port heat exchangers with a fin pitch of 1.2 mm, 1.4 mm and 1.6 mm, and two with slit-fins with a pitch of 1.2 mm and 1.4 mm were experimentally measured under dry, wet and frosting/defrosting conditions. The effects of fin pitch and type on the pressure drop and heat transfer performance were discussed.

a. In the dry condition, as the air velocity increased from 1.5 m/s to 3 m/s, the pressure drops and overall heat transfer coefficients of each heat exchanger increased. At the same air velocity, the pressure drops and overall heat transfer coefficients of louvered-fin heat exchangers were higher than those of slit-fin heat exchangers with same fin pitch.

b. In the wet condition, louvered- and slit-heat exchangers with fin pitch of 1.2 mm and 1.4 mm showed similar pressure drops and overall transport coefficients, which reflected that the effects of fin type and pitch were alleviated by the condensate on the fins. However, when the louvered-fin pitch increased to 1.6 mm, it showed lower pressure drops and overall transport coefficients.

c. In the frosting/defrosting condition, the pressure drop of a single heat exchanger was not identical in each frosting cycle, because of the different initial conditions due to the irregular retention distribution, whereas the overall heat transfer coefficients were relatively stable. The heat exchangers with a lower fin pitch showed higher overall transport coefficients. Nonetheless, the effect of fin type was not obvious on the overall transport coefficients.

d. For the heat exchangers of indoor units, which operate under dry and wet conditions, the louvered-fin with a fin pitch of 1.2 mm is recommended. However for the heat exchangers of outdoor units, which may operates under the frosting/defrosting condition, the slit-fin with a fin pitch of 1.4 mm is suggested.

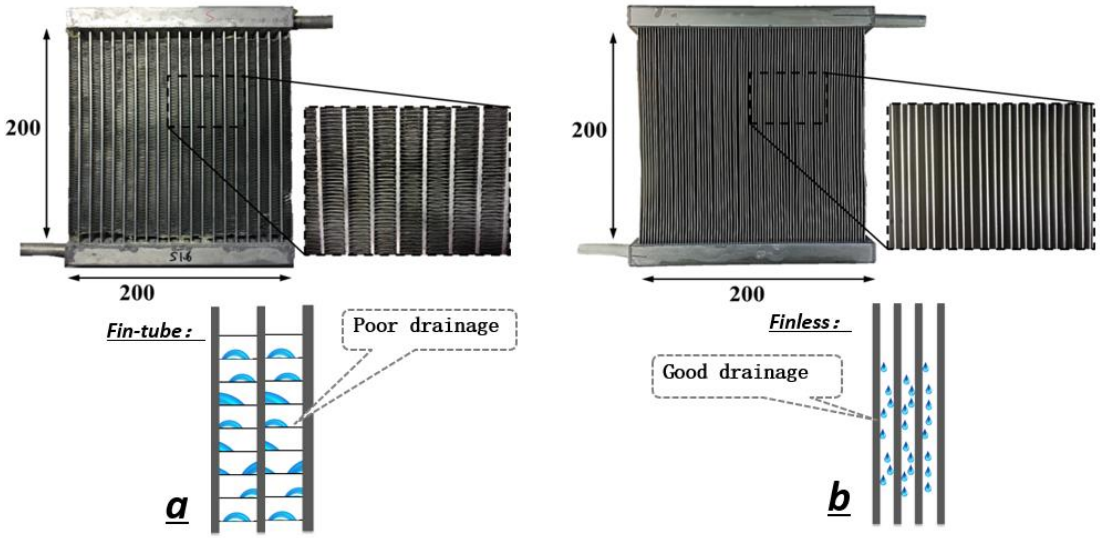

Figure 1 Drainage of fin-tube and finless heat exchangers

Considering the poor drainage performance of fin-tube parallel heat exchangers, we proposed the finless heat exchangers with excellent drainage performance, as shown in figure 1b. However, the heat transfer performance of the finless heat exchanger is much lower than the fin-tube ones, due to the less heat transfer area and lower heat transfer coefficient. Hence, we proposed a method for the heat transfer enhancement of the finless heat exchanger, that in front of which a vortex generator is set, as shown in figure 2. In the heat transfer domain between the flat tubes, the longitudinal vortex can disrupt the thermal boundary, as well as enhance the mixing of hot and cold air, thus the heat transfer performance can be sharply enhanced. With numerical simulation, a double triangle vortex generator is designed, which can offer more heat transfer enhancement comparing to conventional triangle and rectangular vortex generators. Moreover, some parameters analysis which affect the performance of vortex generator is conducted using numerical simulation. Finally, the heat transfer performance and pressure drops of finless heat exchangers with and without vortex generator were experimentally measured under dry, wet and frosting/defrosting conditions while the air velocity varied from 1.5 m/s to 3 m/s, and compared with the fin-tube parallel heat exchangers. Some conclusions are summarized below.

e. The heat transfer coefficient enhancement brought by the double triangle vortex generator is 63% when the air velocity is 2 m/s. Comparing with the triangle and rectangular vortex generator, the double triangle vortex generator bought 20% higher heat transfer coefficient, but also doubled the pressure drop, which achieved 34 Pa. However, the pressure drop is still acceptable.

f. Some parameters can significantly influence the performance of vortex generator. The smaller gap between the finless heat exchanger and the vortex generator, the shorter flat tube width, and the higher air velocity can offer higher heat transfer performance. The tube pitch deviation can sharply decrease the performance of vortex generator, a deviation of 0.8 mm can bring a decrease of 19.4% on heat transfer coefficient.

g. The heat transfer performance enhancement by the vortex generator is verified by experiments. Under the dry condition, comparing to the fin-tube heat exchangers, the heat transfer performance of finless heat exchanger is still about 40% less, while the pressure drop is similar. Under the wet condition, due to the excellent drainage performance, the heat transfer coefficient reached the same level with the fin-tube heat exchanger, while the pressure drop is lower. Under the frosting/defrosting condition, the heat transfer coefficient is similar, but the pressure drop is much lower, which only reach 50 Pa at each end of frosting period. Totally, comparing to the fin-tube heat exchanger, the performance of finless heat exchanger with vortex generator is still lower under dry condition, but it shows superiority under wet and frosting/defrosting conditions.

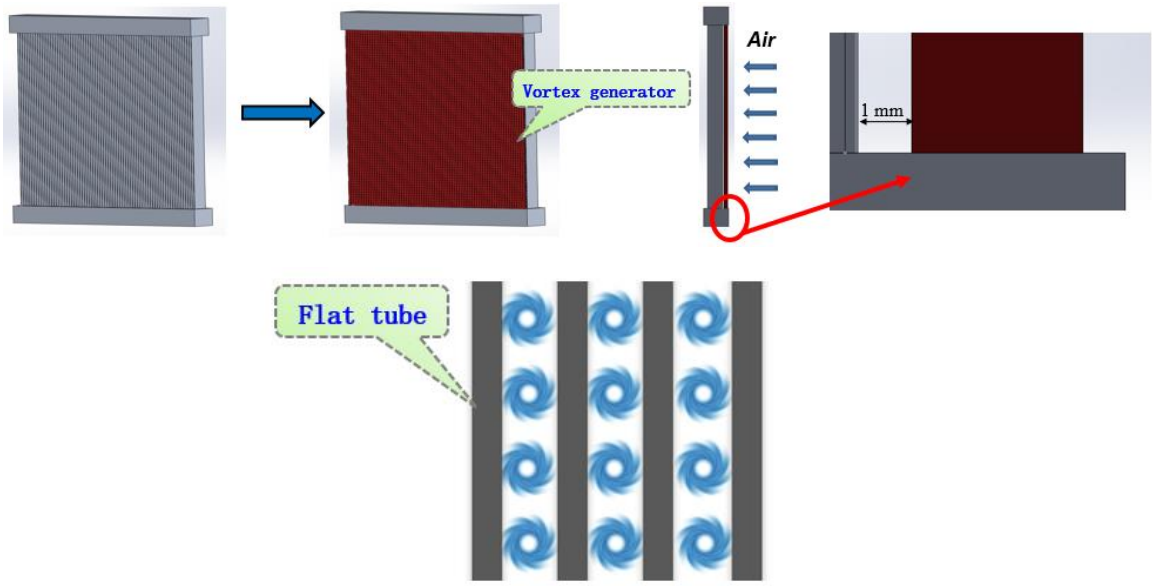

Figure 2 Concept of finless heat exchanger with LVG

3. Distribution of two phase flow in parallel tubes

The maldistribution of two phase flow in the parallel flat tubes is another main problem preventing the parallel heat exchanger to be used in AC system. Because of the density difference and gravity, the liquid separates from the gas and leads to a stratified flow, which causes some tubes receive little liquid refrigerant. In an outlet header of the maldistributed evaporator, vapor from some tubes must be significantly superheated to compensate for two phase refrigerant coming from the other tubes, and provide sufficient bulk superheat for superheat controlled flow. Hence, the superheating vapor in some tubes causes significant reduction in evaporator performance.

The gas and liquid tend to separate from each other in the manifold because of their different densities and initial force. These phenomenon inspire an idea to achieve using FBG without additional inner volume. That is, using the manifold as separator. The detail design of header is shown in figure 3. After two phase flow enter the inlet which set on one side of the manifold, because of lower density, gas changes its flow direction to the manifold length direction immediately, while the liquid impact on the manifold wall and accumulated at the bottom because of gravity. Since the inlets of flat tubes are set at the bottom of manifold, they will only receive liquid but not two phase flow. Moreover, the part of manifold, which combines the flat tubes, is inclined to ensure the reverse bubble converging to the main gas flow due to buoyance. Therefore, the bubbles will not accumulate in the bottom. The gas flow will be discharged out and converging to the evaporated gas at the out let of heat exchanger.

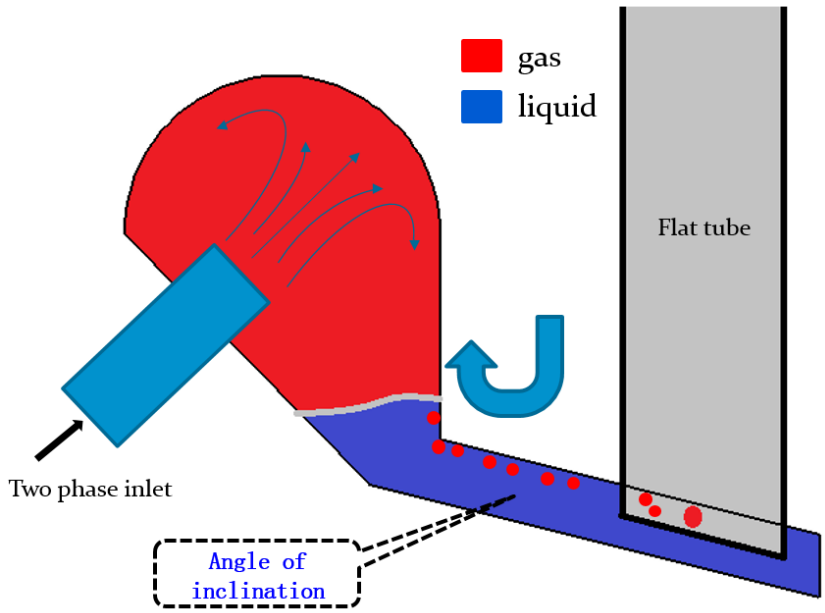

Figure 3. Design of manifold