Optimal Analysis of Domestic Air-Conditioning Heat Exchanger

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Abstract. The energy efficiency improvement of air conditioning systems is very important. The relevant equations of air-conditioning heat exchanger are established based on the basic thermophysical laws in this paper. While R22 is the working fluid, the heat transfer equations of condenser and evaporator in gas and liquid two-phase region were solved with two different diameters 9.52mm and 7mm. The influence of diameter on fluid and heat transfer performance of air-conditioning heat transfer is analyzed based on calculated results. It shows that decreasing the diameter of the heat exchanger tube will be beneficial for the optimal design of heat exchanger if the increasing degree of the average heat transfer coefficient is higher than that of the total pressure drop. The conclusions of this study could provide theoretical basis to optimize the design of domestic air-conditioning heat transfer.

1. Introduction
Condenser and evaporator are air-conditioning heat exchangers. They are key components of domestic air-conditioning system. There are many academic researches about the characteristics of air-conditioning system. Luo et al. [1] simulated the dynamic behavior of one-dimensional flow in multi-stream heat exchangers. Chen et al. [2] compared a heat exchanger with vapor–liquid separation to a conventional heat exchanger and proposed the mechanisms of how the vapor–liquid separation enhances heat transfer coefficient during evaporation.

Basing on the calculation of conservation equations, this study is focused on the analysis of the changes of pressure drop and heat transfer coefficient along the tube length of condenser and evaporator in gas and liquid two-phase region. Because the total pressure drop and average heat transfer coefficient along the tube length of the heat exchanger will influence the performance of air-conditioning system. The results could help optimize the structure of domestic air-conditioning heat exchanger.

2. Conservation Equations
To simplify the calculating process, there are some assumptions for the conservation equations according to Chen et al. [3]:

(1) The air-conditioning system is in steady state;
(2) Liquid phase and gas phase of refrigerant are in thermal equilibrium;
(3) No heat conduction in the axial direction of the tube and the fluid flow is one-dimensional;
(4) No mass diffusion of refrigerant in the axial direction of the tube and refrigerant in the tube is in a homogeneous distribution.

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Continuity equation:
\[
\frac{d}{dz}(\rho u) = 0
\]  
(1)

Momentum equation:
\[
\frac{d}{dz}(\rho u^2) = -\frac{dp}{dz} - \frac{f \rho u^2}{2d}
\]  
(2)

Energy equation:
\[
\frac{d}{dz}(\rho u h) + \frac{d}{dz}\left(\rho u \frac{1}{2} u^2\right) = \frac{\pi d}{A} \dot{q}
\]  
(3)

where \(z\) is the axial length of tube, m; \(\rho\) is the density of refrigerant, kg/m\(^3\); \(u\) is the axial velocity of refrigerant, m/s; \(p\) is the pressure of refrigerant, Pa; \(h\) is the enthalpy of refrigerant, kJ/kg; \(d\) is the inside diameter of tube, m; \(A\) is the section area of tube, m\(^2\); \(f\) is the friction factor, and for single phase flow, \(f = 0.3164 \text{Re}^{-0.25}\) while for two-phase flow, \(f = 3.49 \text{Re}^{-0.47}\), where Re is the Reynolds number.

The heat flux \(\dot{q}\) (W/m\(^2\)) is given as:
\[
\dot{q} = K(T_w - T)
\]  
(4)

where \(K\) is the heat transfer coefficient, W/m\(^2\)K; \(T\) is the temperature of refrigerant, K; \(T_w\) is the temperature of tube wall in heat exchangers, K.

3. Calculation of the Equations

To calculate the single phase flow of superheated gas region and supercooled liquid region in the heat-exchangers, we usually adopt the Dittus and Boelte correlation [4]. Neglecting the dehumidifying effect of the air side, the Wang correlation [5] is used to calculate the two-phase flow heat transfer coefficient of refrigerant in the evaporator. The Shah correlation [6] is proposed to calculate the two-phase flow heat transfer coefficient of refrigerant in the condenser.

Two-phase flow heat transfer of the heat exchanger was calculated and analyzed by homogeneous model in this paper. Equations (1)-(3) were solved separately by the fourth order Runge-Kutta method, along the \(z\) axis. For two-phase fluid, the velocities of liquid flow and gas flow are equal, and the thermal physical property parameters of two-phase state refrigerant are defined as:
\[
\rho_{tp} = \alpha \rho_g + (1 - \alpha) \rho_l
\]  
(5)

\[
\mu_{tp} = x \mu_g + (1 - x) \mu_l
\]  
(6)

\[
\lambda_{tp} = x \lambda_g + (1 - x) \lambda_l
\]  
(7)

where \(\alpha\) is the void fraction of two-phase refrigerant; \(x\) is the mass quality of two-phase refrigerant; \(\rho_g\) is the gas density of refrigerant, Kg/m\(^3\); \(\rho_l\) is the liquid density of refrigerant, Kg/m\(^3\); \(\mu_g\) is the gas dynamic viscosity of refrigerant, kg/ms; \(\mu_l\) is the liquid dynamic viscosity of refrigerant, kg/ms; \(\lambda_g\) is the gas heat conductivity of refrigerant, W/mK; \(\lambda_l\) is the liquid heat conductivity of refrigerant, W/mK; and subscript \(tp\) represents two-phase.

For homogeneous model, the void fraction \(\alpha\) is defined as:
\[
\alpha = \frac{1}{1 + \left(\frac{1}{x} - 1\right) \frac{\rho_g}{\rho_l}}
\]  
(8)

Calculating conditions are as follows:
The working fluid is R22. Condensation temperature $t_k$ is 54.4°C; evaporation temperature $t_0$ is 7.2°C; subcooled temperature $t_{sc}$ is 46.1°C; superheated temperature $t_{sh}$ is 18.3°C; mass flow rate of refrigerant $\dot{m}$ is 320.58kg/h; and exhaust temperature of compressor is 95°C. The heat transfer equations of condenser and evaporator in gas and liquid two-phase region were solved with two different diameters $\phi$ 9.52mm and $\phi$ 7mm. The temperature differences between the tube wall and refrigerant in condenser and evaporator are proposed as 5°C for two-phase region in the calculation.

4. Calculating Results and Discussion

Figure 1 to Figure 4 are the calculating results which show the changes of pressure drop and heat transfer coefficient along the tube length of condenser and evaporator in gas and liquid two-phase region.

For both condenser in Fig.1 and evaporator Fig.3, the refrigerant pressure drop along the tube length increases with the diameter of the heat exchanger tube decreasing, since the fluid has to overcome much bigger flow resistance. However, the heat transfer coefficient along the tube length also increases with the diameter of the heat exchanger tube decreasing, as Fig.2 and Fig.4 shown. Meanwhile, when mass flow rate of refrigerant in the heat exchanger is invariable, the total length of
the tube in condenser or evaporator becomes to shorten with the decreasing diameter. Thus the structure of the heat exchanger will become more compact. It will be benefit to cut the cost of producing heat exchanger.

Table 1 is the comparison of calculating results in two-phase region. It shows that the total length of the condenser tube becomes to shorten and the average heat transfer coefficient increases with the diameter of condenser tube decreasing. Although the total pressure drop along the tube length increases, its extent is lower. Therefore, it is beneficial for the optimal design of a condenser. With the diameter of heat exchanger tube decreasing, the variable degrees of average heat transfer coefficient and total pressure drop in the evaporator are higher than those in the condenser. The enhancing heat transfer is beneficial for the optimal design of an evaporator, but the higher increasing pressure drop along the tube length is not beneficial for an evaporator, since it is harmful to system performance.

Table 1. Comparison of parameters of domestic air-conditioning heat exchanger.

|                  | Condenser (d=9.52mm) | Condenser (d=7mm) | Evaporator (d=9.52mm) | Evaporator (d=7mm) |
|------------------|-----------------------|-------------------|-----------------------|-------------------|
| Total length of the tube (m) | 21.6                  | 16.8              | 18                    | 14                |
| Total pressure drop (kPa) | 8.36                  | 27.09             | 22.39                 | 76.9              |
| Average heat transfer coefficient (W/m²k) | 2297.98              | 4158.25           | 5716.38               | 8613.97           |

5. Conclusion
For both condenser and evaporator, the total length of heat exchanger tube will become to shorten with the diameter of the heat exchanger tube decreasing. Tube length decreasing is benefit to decrease total pressure drop. It will be beneficial for the optimal design of heat exchanger if the increasing degree of the average heat transfer coefficient is higher than that of the total pressure drop.

The calculating results show that it is beneficial for the optimal design of a condenser to use the tube with diameter ø 7mm instead of ø 9.52mm. But for an evaporator, since the increasing degree of total pressure drop is higher, it should be considered carefully if use the tube with diameter ø 7mm instead of ø 9.52mm. The calculating results in this paper are helpful to the optimal design of domestic air-conditioning heat exchanger.

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