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PRINCIPLE OF DESIGNING FIN-AND-TUBE HEAT EXCHANGER WITH SMALLER DIAMETER TUBES FOR AIR CONDITIONER

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ABSTRACT

The principle of designing fin-and-tube heat exchanger with smaller tubes is proposed in this study. The principle includes designing of fin configuration and designing of refrigerant circuits. In the design principle, the suitable fin configuration for 5 mm diameter tubes is designed by Computational Fluid Dynamic-based method, and the refrigerant circuit with 5 mm diameter tubes is designed by simulation-based method. To verify the results of designing, experiments on air conditioner unit are carried out. The experimental results confirm the design principle, and indicate that the optimal air conditioner with smaller diameter tubes has better performance and lower refrigerant charge which will promote the application of inflammable nature refrigerants, such as R290.

1. INTRODUCTION

For the lower material cost and less refrigerant charge, fin-and-tube heat exchangers with smaller diameter tubes (smaller than or equal to 5 mm) gradually replace those with 7 mm or large diameter tubes. When tube diameter decreases from 7 mm to 5mm, the tube cross-sectional area will reduce by 49%, and the refrigerant charge can be decreased accordingly. Due to the lower refrigerant charge, the explosion risk of using inflammable nature refrigerants which have lower GWP and ODP (e.g. R290) can be obviously decreased, which promotes the application of small diameter tubes in R290 systems.

However, the application of smaller diameter tubes will affect the performance of heat exchanger at both of air side and refrigerant side (Ding *et al*., 2009). On air side, the fin size is related to the balance of heat transfer resistance between fin and tube, so fin size for smaller diameter tubes may be smaller than that for large diameter tubes. The fin pitch, which depends on the tube diameter, is also decreased. These may decrease the heat transfer capacity and increase air side pressure drop. On refrigerant side, smaller tube may increase the refrigerant pressure drop, and decrease the heat transfer area of tube. These will decrease the heat transfer capacity if only the tube diameter is reduced. In order to have a good performance of the air conditioner with smaller diameter tubes, it is necessary to propose a principle of designing fin-and-tube heat exchangers, including designing of fin configuration and tube circuits.

For fin configuration designing, many researchers employed empirical equations, which were developed on the experimental data of heat exchanger with 7 mm or large diameter tubes (Wang *et al*., 2002; Ma *et al*., 2007; Ma *et al*., 2009). However, the deviations will occur between predicted values and experimental data, when the empirical equations for larger diameter tubes are used for predicting performance of heat exchanger with smaller diameter tubes. Until now, no data or empirical equation for heat exchanger with 5 mm or smaller diameter tubes is published on literatures. Thus, the design method of fin configuration for heat exchanger with smaller diameter tubes should be proposed.

For tube circuits designing, the simulation based design method with short time consumption and less resource requirements has been widely used to replace the traditional trial and error approaches for heat exchanger design and optimization (Ding, 2007), and it is considered as the most effective way to obtain the required performance in the heat exchanger designing. For tube circuits optimization, the optimization algorithm is used to control the optimization process and generate initial solutions, and the distributed-parameter model is applied to evaluate solutions generated by the optimization algorithm (Domanski, 1991; Liu *et al*., 2004; Domanski and Yashar, 2007; Wu *et al*., 2008a, 2008b). However, the current design method is based on empirical equation for 7 mm or larger diameter tubes. Thus, it is necessary to develop the suitable tube circuits design method for 5 mm or smaller diameter tubes.

In this study, a principle of designing fin-and-tube heat exchanger with smaller tube is proposed, including designing of fin configuration and designing of tube circuits. In order to obtain a set of fins with higher heat exchange capacity and lower air-side pressure, the principal of determining the fin configuration, including the fin size and the fin pattern, is presented. In order to optimize the tube circuits, a simulation-based design method is used, in which a three-dimensional distributed parameter model is used for simulation, and a knowledge-based evolution method optimizer was applied. Experiments on air conditioner unit with 5 mm diameter tubes are carried out to verify the design principle.

2. DESIGN PRINCIPLE

The design principle includes two parts: fin configuration designing and tube circuits designing. Because the fin size determines tube pitch which influents tube distribution, the designing of fin configuration should be firstly carried out and followed by the designing of tube circuits. The scheme of design principle is shown in Fig. 1.

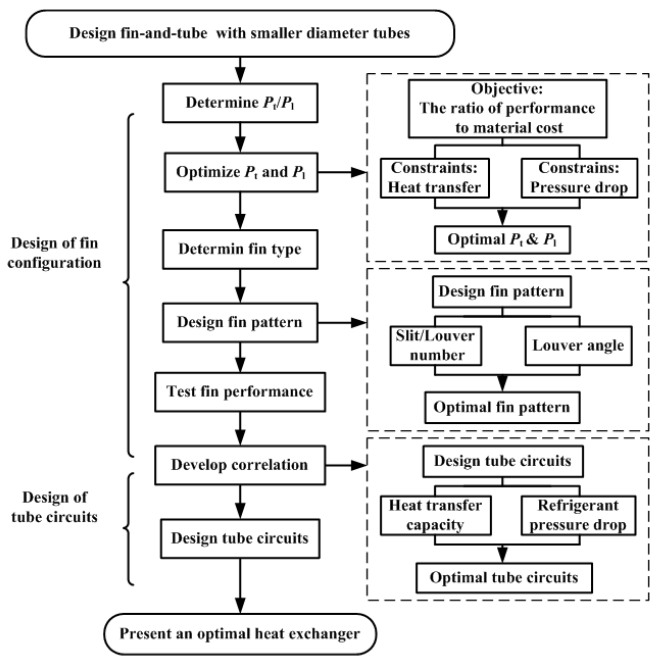


Figure 1: The scheme of design principle of heat exchanger with smaller diameter tubes

2.1 Fin Configuration Designing

In fin configuration designing, the fin size and fin pattern which mainly affect heat transfer capacity and air pressure drop are designed by Computational Fluid Dynamic (CFD)-based method. The design steps shown as follow:
Step 1: Determine the best ratio of transverse tube pitch to longitudinal tube pitch (P_t/P_l) by fin efficiency analysis;
Step 2: Optimize transverse tube pitch and longitudinal tube pitch by analysis of performance and material cost;
Step 3: Optimize fin pattern by comparing performances of fins with different patterns through CFD-based method;
Step 4: Test the performance of heat exchangers with smaller diameter tubes;
Step 5: Develop empirical equation for predicting performance of heat exchanger with smaller diameter tubes.
The details of each step will be illustrated as follow:

Step 1: Determine the best P_t/P_l

In this study, the best P_t/P_l refers to the P_t/P_l of fin with the highest fin efficiency among the fins with the same area. The fin efficiency, in Equation (1), is defined as the ratio of the actual fin heat transfer capacity ($Q_{actual,fin}$) to the maximum possible heat transfer capacity ($Q_{ideal,fin}$) if the entire fin were at the base temperature.

$$\eta = \frac{Q_{actual,fin}}{Q_{ideal,fin}} \tag{1}$$

Both of $Q_{actual,fin}$ and $Q_{ideal,fin}$ are calculated by CFD method. In CFD calculation, the values of P_t/P_l are selected in the range of commonly used P_t/P_l of heat exchangers in air conditioners. The geometrical models are selected as the fin-and-tube with 2-rows tubes. The boundary conditions are defined by the working conditions of air conditioner. The fin is coupled with tube wall in the model of actual fin, and the fin temperature is set as the same with that of tube wall in the model of ideal fin. The upper and under air surface are defined as periodic surface without pressure drop. From the CFD calculation, the best P_t/P_l of fin with high fin efficiency and within the limits of manufacturing can be easily determined.

Step 2: Optimize P_t and P_l

For evaporator, a thin condensate film will form in refrigeration conditions, so the louver or slit will be blocked by condensate film which will make the louver or slit like the plate fin. Thus, this method employs the correlations for plate fin to determine fin size.

In the fin size optimization, one objective function is used to analyze the ratio of performance to material cost as Equation (2) shown. And two constraint functions, in Equations (3) and (4), are that the *UA* of fin for smaller tubes should be equal to or larger than requirement, and the air pressure drop should be equal to or lower than requirement.

$$\max w = \frac{UA}{C} \tag{2}$$

$$UA > UA_{require} \tag{3}$$

$$\Delta P > \Delta P_{require} \tag{4}$$

where, *C* refers to the material cost of copper and aluminum, and *UA* can be calculated from Equations (5) to (7),

$$UA = \frac{1}{\dfrac{1}{n\eta h_0 A_{fu}} + \dfrac{1}{nh_r A_{ru}}} \tag{5}$$

$$A_{fu} = 2\left[P_t P_l - 3.14(D/2)^2\right] + 3.14F_p D \tag{6}$$

$$A_{fu} = 3.14F_p(D - 2\delta) \tag{7}$$

ΔP can be calculated by Equation (8),

$$\Delta P = \frac{G_c^2}{2\rho_1}\left[\frac{A_0\rho_1}{A_c\rho_m} f + (1+\delta^2)\left(\frac{\rho_1}{\rho_2} - 1\right)\right] \tag{8}$$

Step 3: Optimize fin pattern

In fin pattern optimization, because no empirical correlation for predicting the performance of enhanced fin-and-tube with smaller diameter tubes is published, the CFD method is used to simulate the heat transfer capacity and air pressure drop of heat exchangers in this study.

For louver fin, the louver angle and louver number are independent variables, while louver height and louver pitch are determined by two independent variables. Thus, the effects of louver angle and louver number on fin performance are needed to be discussed in designing. For slit fin, the slit height is determined as the half of fin pitch, and the slit pitch is determined by slit number. Thus, the effect of slit number on fin performance will be discussed in designing. Based on the CFD result, the optimal fin pattern with high heat transfer capacity and low air pressure drop will be recommended.

Step 4: Test the performance of heat exchanger
The fin-and-tube heat exchangers with 5 mm diameter tubes are manufactured by air conditioner manufacturer and tested in an experimental system which is shown in Figure 2. The details and uncertainties of experimental system were illustrated on the previous papers (Ma, 2007). In experiments, the working conditions are selected by the evaluation standard for room air conditioners, in which the evaporation temperature is 283 K, the air inlet temperature is 300 K, the range of air humidity is from 40% to 60%, and the range of wind velocity is from 0.5 m/s to 1.5 m/s. The details of data reduction process, which considered full wet and partially wet conditions, were introduced on previous papers (Ma, 2007; Ma, 2009).

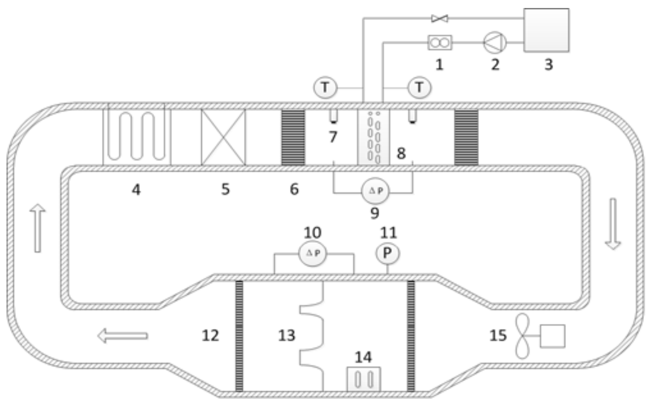


Figure 2: Scheme of experimental system

Step 5: Develop prediction correlation
Based on the experimental data, a multiple linear regression technique in a practical range of experimental data is used to develop prediction correlation for heat exchanger with smaller diameter tubes.

2.2 Tube Circuits Designing

The simulation-based design method, which bases on the reliable model of heat exchanger and combines with the optimization algorithm, is used to design tube circuits. Figure 3 shows the scheme of simulation-based design method for air conditioner with smaller diameter tubes. The potential geometry investigation is used to analyze the design constraints of space and manufacturability. The first priority analysis is used to determine the potential geometries with first priority of whether tube or fin should be first increased by considering the performance and material cost of heat exchanger. The fixed inputs are used to fix potential geometries of heat exchangers after the first priority analysis. The multi-objective optimizer knowledge based evolution method, is used to control the optimization process and obtain optimal solution of heat exchangers.

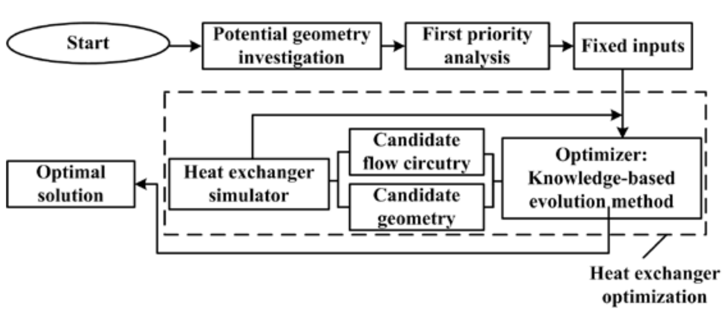


Figure 3: Scheme of simulation-based design method

2.2.1 The heat exchanger simulator

The three dimensional distributed-parameter model based on graph theory (Liu *et al*., 2004) is employed to predict heat exchanger performance with different circuits. The predicted heat exchange capacity of Liu's model agrees with experimental ones within a maximum error of ±10% (Liu *et al*., 2004). Liu's model divides heat exchanger into several control volumes along length, width and height direction of heat exchanger. The control volume includes refrigerant, air and fin-and-tube heat exchanger. The governing energy equation and momentum equation of refrigerant and air are shown in Equations (9) to (13), respectively.

$$Q_r = M_r(h_{r,out} - h_{r,in}) = \alpha_r A_i\left(\frac{T_{r,in}+T_{r,out}}{2} - T_{wall}\right) \tag{9}$$

$$\Delta p_r = \Delta p_{r,f} + \Delta p_{r,acc} + \Delta p_{r,g} \tag{10}$$

$$Q_a = M_a(h_{a,out} - h_{a,in}) = \alpha_a A_o\left(\frac{T_{a,in}+T_{a,out}}{2} - T_{wall}\right) \tag{11}$$

$$\Delta p_a = \frac{G_{a,\max}^2}{2\rho_{a,i}}\left[\frac{A_o\rho_{a,in}}{A_c\rho_{a,m}} f_a + (1+\sigma^2)\left(\frac{\rho_{a,in}}{\rho_{a,m}} - 1\right)\right] \tag{12}$$

$$Q_r + Q_a + Q_{front} + Q_{back} + Q_{top} + Q_{bottom} = 0 \tag{13}$$

where, A_i is the heat transfer area of refrigerant side; T_{wall} is the tube wall temperature; $\Delta p_{r,f}$ is the frictional pressure drop; $\Delta p_{r,acc}$ is acceleration pressure drop; $\Delta p_{r,g}$ is the pressure drop caused by gravity; A_o is heat transfer area of air side; $T_{a,in}$ and $T_{a,out}$ are the air inlet and outlet dry bulb temperature, respectively; $G_{a,max}$ is the air mass flux at minimum cross-sectional area; f_a is the friction factor of air; σ is the contraction ratio of cross-sectional area; $h_{r,in}$ and $h_{r,out}$ are the enthalpy of refrigerant at inlet and outlet of heat exchanger; $h_{a,in}$ and $h_{a,out}$ are the enthalpy of air at inlet and outlet of heat exchanger; Q_{front}, Q_{back}, Q_{top} and Q_{bottom} are the heat conductions through fins from front row, back row, upper column and bottom column, respectively.

The accuracy of distributed-parameter model is affected by the accuracy of correlations for predicting heat transfer coefficient and pressure drop. In this study, the correlations for smaller diameter tubes heat transfer coefficient and pressure drop are carefully chosen as Table 1 shows, and the slip ratio model developed by (Zivi, 1964) is used to predict mass inventory of heat exchanger.

Table 1: Heat transfer and pressure drop correlation for two phase refrigerant

Tube diameter	Items	Correlations	
		Evaporator	Condenser
7 mm and 9.52 mm	Heat transfer	Kandlikar et al. 1997	Yu and Koyama 1998
	Pressure drop	Kuo and Wang 1996	Smith et al. 2001
5 mm	Heat transfer	Hu et al. 2009	Yu and Koyama 1998
	Pressure drop	Ding et al. 2009	Huang et al. 2009

2.2.2 The knowledge-based evolution method

The knowledge-based evolution method (KBEM) (Wu *et al*., 2008a, 2008b) is used to optimize heat exchanger. It consists of two parts: the improved genetic algorithm (IGA) and the knowledge-based optimization module (KOM).

The IGA in the KBEM is the improved version of the conventional genetic algorithm. The improvements include the tailor-made coding method, specific initialization method, and improved genetic operators (crossover, mutation and correction). Crossover operator is used to generate new heat exchanger named child from the existed two heat exchangers named parents. Mutation operator is used to perform a stochastic mutation based on child heat exchanger to accelerate optimization process. And correction operators are added in the IGA to avoid infeasible solutions. The IGA is the basis of the KBEM because it generates the initial solutions and controls the whole optimization process.

The knowledge-based search methods are applied to increase the optimization efficiency by reducing search space according to the domain knowledge without losing optimal solutions. The main reason for the low efficiency of the conventional genetic operators is that they neglect the inner characteristics of the optimization object and only use random operations to blindly search the solutions on a wide solution space. Applying the domain knowledge to generate better solutions by considering the inner characteristics of the optimization object may help to make up the deficiency of the conventional genetic operators, and then improve the efficiency of the optimization process. In the KBEM, the knowledge-based search methods are merged into one module which is used after the genetic operations of the IGA during the optimization process.

3. THE STUDY CASE OF DESIGNING

In this section, a case of designing air conditioner with 5 mm diameter tubes is carried out by the design principle proposed above. The design result is verified by the experiments on air conditioner unit.

3.1 Designing Result

In this case, the indoor heat exchanger chooses the fin-and-tube with 5 mm diameter tubes, and the outdoor heat exchanger chooses the fin-and-tube with 7 mm diameter tubes which has larger fin pitch to prevent performance degradation due to frost in heat pump condition. Because fin-and-tube with 7 mm diameter tubes is already used in current air conditioner, this section will present the designing of fin-and-tube with 5 mm diameter tubes.

3.1.1 Fin configuration designing

Step 1: Determine the best P_t/P_l

In P_t/P_l designing, air inlet temperature is set as 300 K, the tube wall temperature is set as 280 K. Other boundary conditions of CFD are set as above section. From the CFD result in Figure 4, the best P_t/P_l can be easily determined as 1.23 where the fin efficiency reaches the highest value.

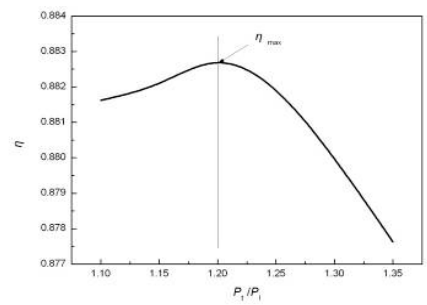
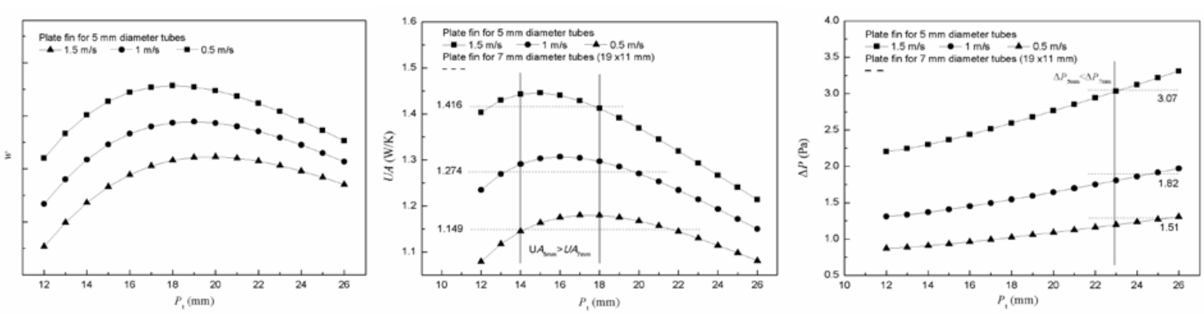


Figure 4: Variations of fin efficiency as function of wind velocity

Step 2: Optimize P_t and P_l

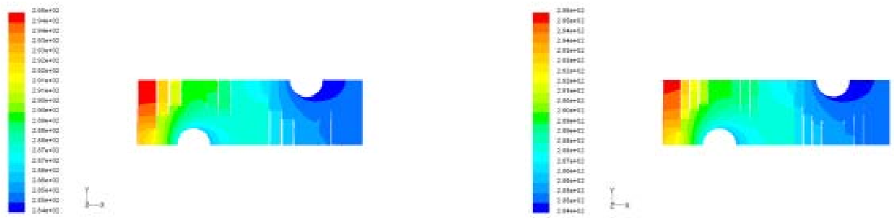
In fin size designing, the *UA* of fin for 5 mm diameter tubes should be higher than that of plate fin for 7 mm diameter tubes, and the ΔP of fin for 5 mm diameter tubes should be smaller than that of plate fin for 7 mm diameter tubes. Based on the above design principle, the variations of heat transfer coefficient, air pressure drop and the ratio of performance to material cost as function of the P_t are calculated and shown in Figure 5 (a) to (c). From the results, the best P_t can be determined as 18 mm which has high *w* and also satisfies the constraints of *UA* and ΔP. From the best P_t/P_l, the optimal fin size is determined as 18×14.7 mm.



(a) (b) (c)

Figure 5: Variations of *w*, *UA* and ΔP as function of P_t: (a. Ratio of performance to material cost-*w*; b. Heat transfer performance-*UA*; c. Air pressure drop-ΔP)

Step 3: Optimize fin pattern
Based on the optimal fin size, the performances of fins with 3 louvers and 4 louvers are calculated by CFD method. The temperature distributions on fins surface are shown in Figure 6. The results of heat transfer capacity and air pressure drop of heat exchanger are shown in Table 2. From results, the fin with 4 louvers has higher heat transfer capacity, and smaller air pressure drop than those of fin with 3 louvers which are caused by the more louver number but lower louver height. Thus, the louver fin with 4 louvers is selected as the optimal fin pattern.



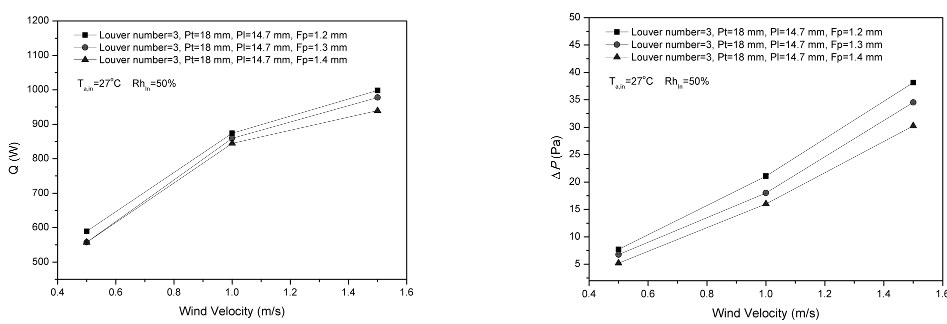
(a) Louver fin with 3 louvers (b) Louver fin with 4 louvers

Figure 6: CFD results of temperature distribution on louver fins with different pattern

Table 2: Heat transfer capacity and air pressure drop of louver fins with different pattern

Louver angle	Louver number	Heat transfer capacity (W)	Air pressure drop (Pa)
25^{o}	3	16.35	17.10
	4	16.53	16.75

Step 4: Test the performance of heat exchanger
Figure 7 (a) and (b) show the testing results of heat transfer capacity and air pressure drop of optimal fin-and-tube heat exchangers with 5 mm diameter tubes. This study presents the testing results of optimal fin-and-tube heat exchangers with 5 mm diameter tubes. For developing prediction correlation, the fins with other geometrical parameters, such as fin size and louver number, are also manufactured and tested.



(a) Heat transfer capacity (b) Air pressure drop

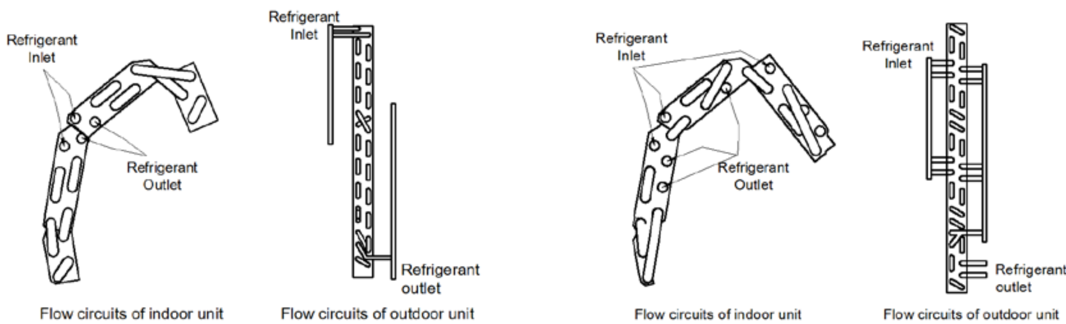
Figure 7: Experimental results of variations of heat transfer capacity as function of wind velocity

Step 5: Develop prediction correlation
Based on the large number of experiments on fin-and-tube with 5 mm diameter tubes, a multiple linear regression technique in a practical range of experimental data is used to develop prediction correlation for heat exchanger with smaller diameter tubes. The correlation for fin-and-tube with 5 mm diameter tubes is added into heat exchanger simulator for tube circuits designing.

3.1.2 Tube circuits designing
Based on the design result of fin configuration, the structural parameters and flow circuits of original and optimal air conditioners are shown in Table 3 and Figure 8 (a) and (b). For optimal indoor heat exchanger with 5 mm diameter tubes, the fins pitch is 1.2 mm and the flow circuitry is 3 paths. For optimal outdoor heat exchanger with 7 mm diameter tubes, the fin pitch is 1.4 mm and the flow circuitry is 4 paths.

Table 3: Structural parameters of original and optimized air conditioner

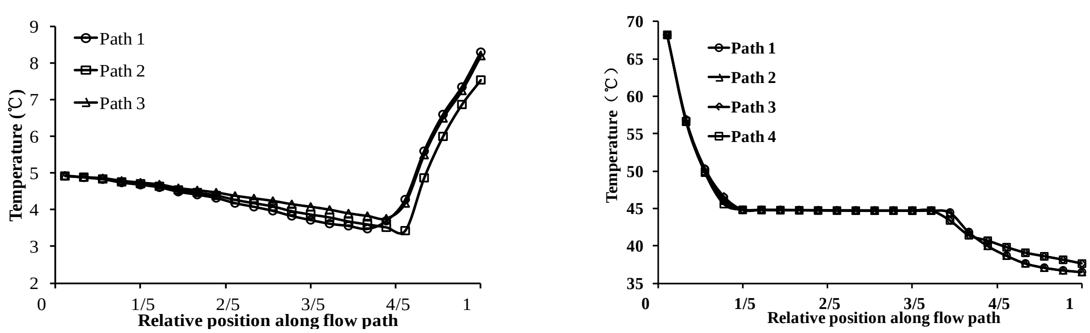
Structural parameters	**Original air conditioner**		**Optimal air conditioner**	
	Indoor unit	**Outdoor unit**	**Indoor unit**	**Outdoor unit**
Diameter of tube, mm	7	9.52	5	7
Length/Width/Height, mm	228/22/320	708/43.3/480	228/27.2/320	706/36/462
Row number/Column number per row	2/12	2/20	2/12	2/22
Row space/Column space, mm	11.0 /19.0	21.6/25.4	15/18	18/21
Bottom boundary space of each row, mm	4.75, 14.25	6.5, 18.9	4.75, 14.25	5.25, 15.75
Path number	2	2	3	4
Fin thickness, mm/Fin pitch, mm	0.105/1.6	0.105/1.8	0.105/1.4	0.105/1.4
Fin type	Louver fin	Wavy fin	Louver fin	Wavy fin
Tube thickness, mm/tube Diameter, mm	0.25/7.0	0.28/9.52	0.20/5.0	0.25/7.0



(a) Flow circuit of original air conditioner (b) Flow circuit of optimal air conditioner

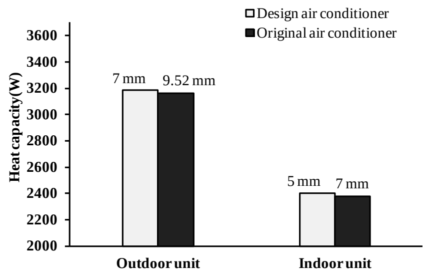
Figure 8: Flow circuits of air conditioners

Through above simulation-based design method, the temperature distributions along flow path of optimal heat exchangers are uniform distributed, as shown in Figure 9. The heat transfer capacities of optimal indoor and outdoor heat exchanger with smaller diameter tubes are shown as Figure 10 (a). From Figure 10 (a), the heat transfer capacities of optimal indoor and outdoor heat exchanger with 5 mm diameter tubes are almost the same as those of original indoor and outdoor heat exchanger. From Figure 10 (b), the refrigerant charge of outdoor unit and indoor unit of air conditioner with 5 mm diameter tubes is decreased by 50% and 30% than those of air conditioner with large diameter tubes, respectively. The total refrigerant charge of air conditioner is reduced by 27%.

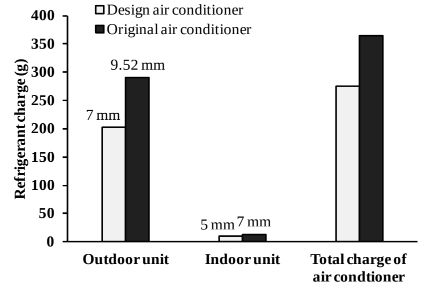


(a) Heat exchanger of indoor unit (b) Heat exchanger of outdoor unit

Figure 9: Temperature distributions along flow path of heat exchangers with 5 mm diameter tubes



(a)



(b)

Figure 10: (a) Heat transfer capacity of original optimal air conditioner
(b) Refrigerant charge of original optimal air conditioner

3.2 Experimental Result

Both of optimal air conditioner with 5 mm diameter tubes and original air conditioner are tested in the same working conditions. The performances of these two air conditioners are measured by the enthalpy potential method, and the experimental results are shown as Figure 11. From Figure 11, the *EER* and Cooling capacity is slightly increase while refrigerant charge is larger than 275 g. Because the refrigerant charge is the less the better due to the flammability of R290, the best refrigerant charge is 285 g.

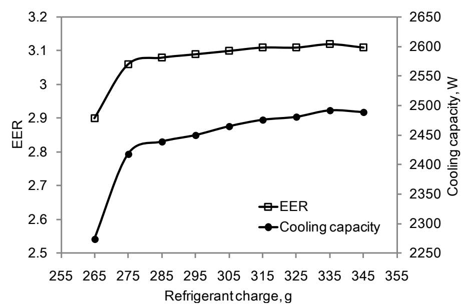


Figure 11: Variation of air conditioner performance as function of refrigerant charge

The main performance data of optimal air conditioner is listed in Table 4. In Table 4, the experimental results confirm the simulation results, and they also shows that 5 mm diameter tubes are suitable for the safe room air conditioners with R290.

Table 4: Experimental result of optimal air conditioner and original air conditioner

Items	Simulation result	Experimental result
Refrigerant charge	275 g	285 g
Cooling capacity	2403 W	2439 W
EER	3.05	3.08
Indoor unit heat capacity	2403 W	2439 W
Outdoor unit heat capacity	3183 W	3117 W
Condensing temperature	46.5 ℃	45.8℃
Evaporating temperature	7.9 ℃	7.8℃

4. CONCLUSION

(1) This study proposes a design principle for heat exchanger with smaller diameter tubes. The design principle includes fin configuration designing which bases on Computational Fluid Dynamic method and tube circuits designing which bases on simulation-based design method;

(2) Through design principle, fin-and-tube heat exchanger with 5 mm diameter tubes is designed, and an air conditioner with 5 mm diameter tubes is developed;

(3) The experimental result of air conditioner unit shows that the air conditioner, which used recommended heat exchanger with 5 mm diameter tubes, has lower refrigerant charge and also good performance. This can promote to widely and safely use inflammable nature refrigerants, such as R290.

NOMENCLATURE

A	heat transfer area	(m^2)
C	material cost	(kg)
D	diameter	(m)
F_p	fin pitch	(m)
G_c	mass flus	(kg/m^2s)
h	heat transfer coefficient	(W/m^2K)
P_t	transverse tube pitch	(m)
P_l	longitudinal tube pitch	(m)
ΔP	pressure drop	(Pa)
Q	heat transfer capacity	(W)
UA	heat transfer efficiency	(W/K)
δ	fin thickness	(m)

Subscripts

a	air
f	fin
i	inner
o	outer
r	refrigerant

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