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Performance Analysis of an Indoor Heat Exchanger with R-410A for GHP Application

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Abstract

The objectives of this paper are to study the effects of thermal and geometric conditions on the performance of indoor heat exchangers with R-410A for Gas Engine Driven Heat Pump (GHP) application and to find the optimum design conditions of indoor heat exchangers by parametric analysis for the key parameters. The key parameters are number of tube row, number of tube pipe, fin pitch and transverse tube pitch. In the air side, moisture out of the humid air condenses on the fin surface while the refrigerant (R-410A) boils inside the smooth tube. Therefore this study uses Log Mean Enthalpy Difference (LMHD) method to analyze the heat transfer from the humid air to the refrigerant. This study determines the heat exchanger size, air side/refrigerant side pressure drop and overall heat transfer coefficient. Optimum design conditions for the key parameters are also determined by the parametric analysis. The results show that number of rows and pipes, fin pitch have significant effect on the heat exchanger size. It is also found that the tube length of the louver fin is 17~30% shorter than that of the plate fin.

Key words: R-410A, GHP, LMHD, Parametric analysis, Optimum design

Nomenclature

A_o	: total contact area[m^2]
A_{po}	: outer surface area of tube[m^2]
b	: ratio between temperature and enthalpy [kJ/kg℃]
d	: diameter [m]
D_c	: tube diameter including fin collar[m]
D_h	: hydraulic diameter [m]
f	: friction factor
H	: enthalpy [kJ/kg]
h	: heat transfer coefficient[W/m^{2}K]
j	: Colburn factor
k	: thermal conductivity [W/mK]
L	: length [m]
$\dot{m}$	: mass flow rate [kg/s]
N	: number of tube row
P	: pressure [kPa]
Q	: heat transfer rate(W)
T	: temperature [℃]
T_p	: temperature of the pipe [℃]
U	: overall heat transfer coefficient [W/m^{2}K]
W	: relative humidity [%]
Fr	: Froude number
Re	: Reynolds number
We	: Weber number

Greek symbols

θ	: angle with horizontal
Φ	: two phase flow friction coefficient

Subscripts

a	: air
cal	: calculation
f	: fluid
i	: inlet
LO	: liquid only
o	: outlet
p	: pipe
R	: refrigerant
tp	: two phase

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r : rows
w : water film
wn : mean water film

1. Introduction

Recently, gas-engine driven heat pump (GHP) has been paid attention to the air conditioning purpose. Refrigerant R22 currently used in GHP is supposed to be abandoned entirely by 2020 according to the Montreal Protocol. R410A is now considered as a proposing candidate for GHP application as it is equivalent in performance with R22.

In general, the performance of heat exchanger is greatly influenced by the heat transfer coefficient of the air side which is much smaller than that from the refrigerant side. In order to improve the heat transfer coefficient on the air side, the contact area between the air and the heat exchanger should be increased. For the purpose, the number of fins of the heat exchanger may be increased and the shape may be changed, but those changes result in the pressure drop on the air side and increases noise and vibration.

In the present study, an optimum design program for heat exchanger, which is essential in enhancing the performance of air-conditioning system, has been developed based on the combined heat and mass transfer analysis. In the program, the performance analysis of evaporator is conducted using the LMHD method. In order to satisfy the required capacity, the optimal design is carried out through calculation of the length of heat exchanger, and the pressure drops on the air side and refrigerant side.

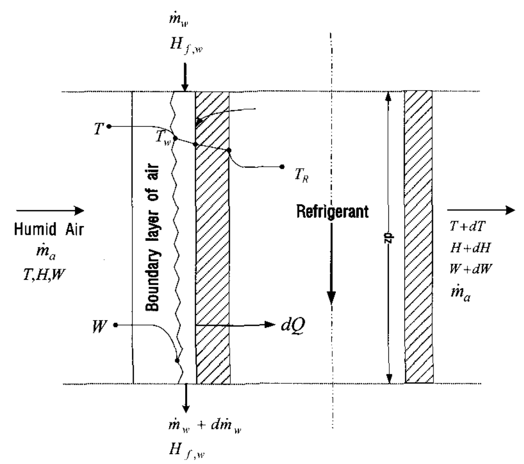

Fig. 1. Cooling and dehumidification process of moisture air around a tube.

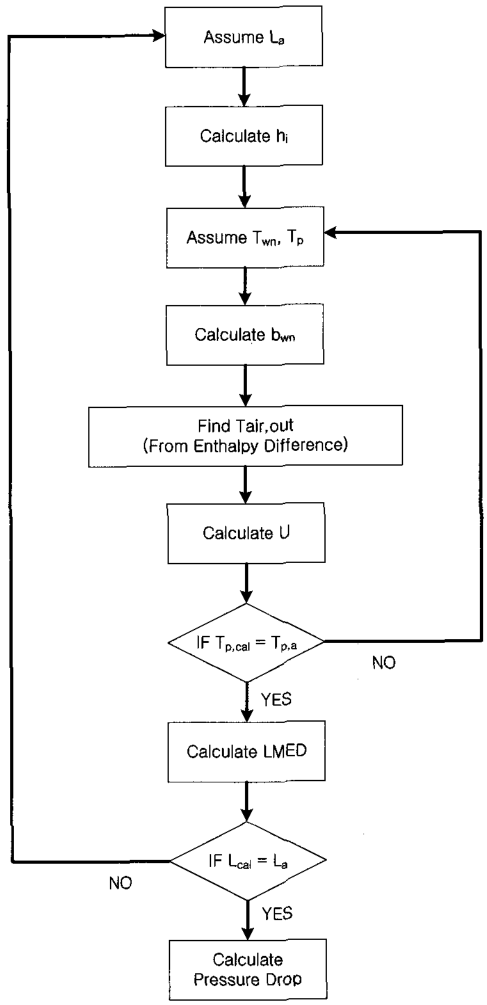

Fig. 2. Procedure of heat exchanger design.

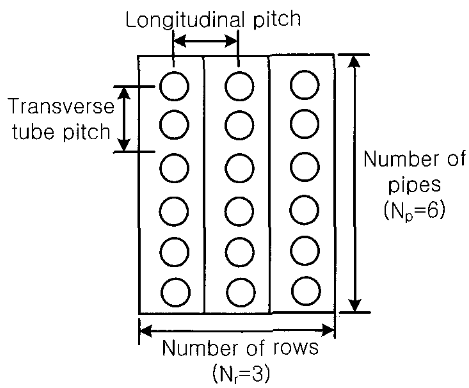

Fig. 3. Heat exchanger configuration.

Figs. 3 and 4 show the heat exchanger configuration and the schematic diagrams of the louver and the plate fins, respectively.

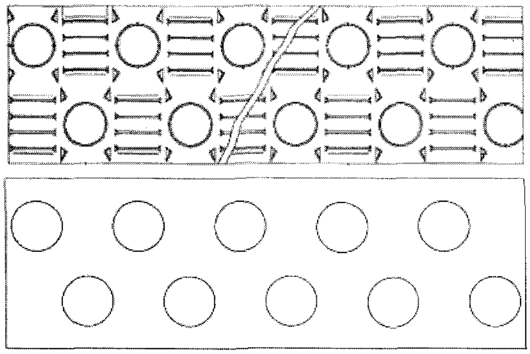

Fig. 4. Schematic diagrams of louver fin (top) and plate fin (bottom).

2. Heat exchanger design and performance analysis

2.1 Theoretical background

Due to the difficulty in analyzing fluid movement inside tubes, the formulas based on the experiments have been mainly used. However, such an approach is valid only in specific domains, and there does not exist generalized formula for refrigerant R410A. In this study, Eq. (1) provided by Kattan et al[1] for the two-phase flow is used for the analysis of heat exchanger. The heat transfer coefficient of Kattan et al[1] agrees quite well with the experimental result by Kim et. al[2] for boiling heat transfer characteristics of R22 and R410A in micro-fins.

$$h_{tp} = \frac{d_i \theta_{dry} h_{vapor} + d_i (2\pi - \theta_{dry} h_{wet})}{2\pi d} \tag{1}$$

For the pressure drop on the refrigerant side, the Friedel's correlation[3], Eq. (2) is used.

$$\Phi_{LO}^2 = E + \frac{3.24 F \cdot H}{Fr_{tp}^{0.045} We^{0.035}} \tag{2}$$

where Φ_{LO}^2 is the two phase frictional multiplier. E, F, and H are parameters defined in the Friedel's correlation[3]

Vigorous research activities have been reported on the heat transfer coefficient and the pressure drop on the air side of fin-tube heat transfer[4,5]. In the present design program, the heat transfer coefficient and the pressure drop on the air side are calculated using the Wang's correlations[6], Eqs. (3) and (4).

$$j = 19.39 Re_{Dc}^{j1} \left(\frac{F_p}{D_c}\right)^{1.352} \left(\frac{P_l}{P_t}\right)^{0.6795} N^{-1.291} \tag{3}$$

$$f = 16.55 Re_{D_c}^{f1} (10 Re_{D_c})^{f2} \left(\frac{A_o}{A_{p,o}}\right)^{f3} \left(\frac{P_l}{P_t}\right)^{f4} \left(\frac{F_p}{D_h}\right)^{-0.5} \tag{4}$$

where F_p, P_l, and P_t, are fin pitch, longitudinal pitch, and transverse tube pitch respectively.

2.2 Design method and procedure

In the indoor heat exchanger, the temperature on the fin-tube surface is lower than that of the atmospheric air, resulting in the moisture condensation on the fin surface. In this case, the difference between the enthalpies of the interfacing air and the saturated moist air at the surface temperature becomes an important factor. Accordingly, analysis on the humid condensing air is required on wetted surface. Fig. 1 shows the cooling and dehumidification process of moisture around a tube. Condensed water exists as a water film with a certain thickness in a minute interval. The air and the refrigerant move with uniform average velocity, and the thermal conditions of air at the interface between the water film and the moist air are the same as those of saturated moist air at the representative temperature.

As for heat exchanger design procedure firstly an initial heat exchanger length is assumed. Then the heat transfer coefficient on the refrigerant side is found using the Kattan's correlation[1], and b_{wn} is found under the assumptions of T_{wn} and T_p. $T_{air,out}$ is found using the difference in the enthalpy, and then the heat transfer coefficient is found. After then $U_{o,w}$, the overall heat transfer coefficient is finally found using Eq. (5).

$$U_{o,w} = \frac{1}{\frac{b_R' A_o}{A_{p,i} h_i} + \frac{b_{w,m}(1-\Phi_w)}{h_{o,w}(A_{P,o}/A_F + \Phi_w)} + \frac{b_{w,m}}{h_{o,w}}} \tag{5}$$

If the assumed values for T_{wn} and T_p satisfy the thermal conditions, the length of the heat exchanger and pressure drop on the moist air side are found using the LMHD. Fig. 2 shows the flow chart for the design procedure.

The present design program is developed using MFC(Microsoft foundation class) of Visual C, and Refprop 7.0[7] is used to calculate the physical properties on the refrigerant side. The physical properties of the moist air is obtained from the ASHRAE Handbook[8].

2.3 Parametric analysis

Base input data needed in heat exchanger design program are listed in Table 1. Capacity of heat exchanger, diameter and thickness of the tubes, the number and the shape of the tubes, air velocity on the air side, and fin configuration are based on the conventional indoor heat exchanger. The inlet temperature of the air and the relative humidity are also based on the indoor units' standard experimental conditions. Table 2 summarizes the calculated values for the base input data conditions.

Table 1. Input data.

Refrigerant side	Cooling capacity(kWh)	5.6
	Refrigerant inlet Temp.(℃)	11
	Refrigerant outlet Temp.(℃)	12.6
Tube side	Tube outer dia.(mm)	7.1
	Tube thickness(mm)	0.5
	Tube conductivity(W/mK)	390
	Number of tube rows(Nr)	3
	Number of tube pipe(Np)	6
	Longitudinal pitch(mm)	21
	Transverse pitch(mm)	12.7
Air Side	Flow rate(m^3/min)	15
	Input temp(℃)	27
	Relative humidity(%)	50
Fin	Fin type	Louver
	Number of fin	14
	Fin thickness(mm)	0.127
	Fin conductivity(W/mK)	200

Table 2. Calculation results.

Results	Outlet air temperature(℃)	14.15
	Overall heat transfer coefficient (W/mK)	88.37
	Pipe Length(m)	0.91
	Refrigerant Pressure drop(kPa)	9.74
	Air Pressure drop(kPa)	4.26
Heat Exchanger Size	Length(mm)	906
	Hight(mm)	105
	Width(mm)	25

The most important factors in evaluating the indoor heat exchanger's performance are its size and pressure drop on the air side. Accordingly, it is judged that the number of rows is the most influencing factor for the pressure drop on the air side. Fig. 5 shows the pressure drop and the length of the heat exchanger as a function of the number of rows (1~5). As the number of rows increases, the tube length decreases while the pressure drop in the air side increases. It is also found that the pressure drop in the refrigerant side decreases substantially as the number of rows increases due to the reduction of length the tube.

Fig. 6 shows the pressure drop in the air and the refrigerant sides and the length of tube with respect to the number of pipes. As the number of pipes increases, the tube length decreases, and the pressure drops on both sides decrease minutely. This is because of a fact that, when the number of pipes increases, the length is reduced, thus reducing the area over which the air passes through, and then affecting the pressure drop on the air side. However, in this case, the height of the heat exchanger increases in proportion to the number of pipes.

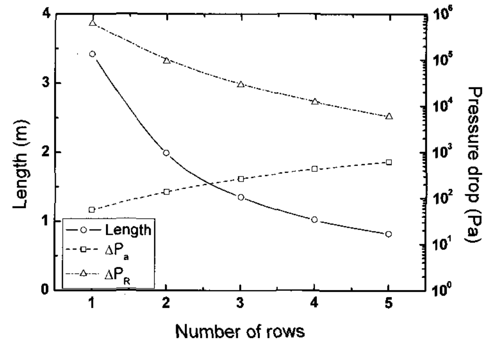

Fig. 5. Pressure drop and length with number of row.

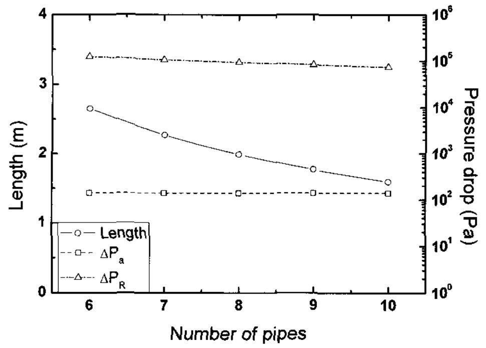

Fig. 6. Pressure drop and length with number of pipe(row=2).

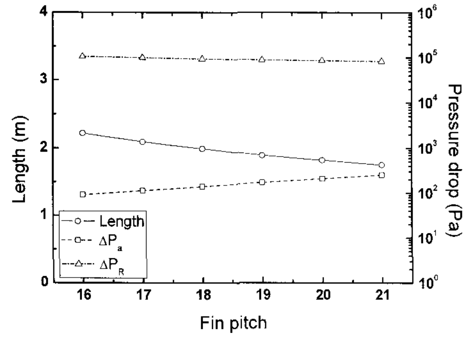

Fig. 7. Pressure drop and length with fin pitch.

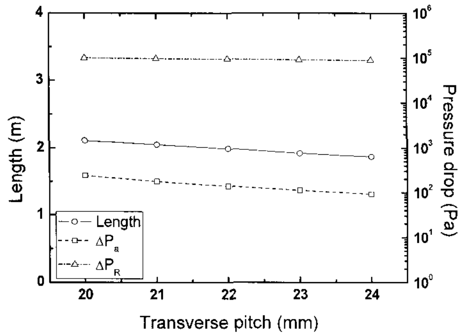

Fig. 8. Pressure drop and length with transverse tube pitch.

Fig. 7 shows the change in pressure drop and the tube length with respect to the fin pitch. It is found that the pressure drop of the moist air greatly increases in comparison with the reduction of tube length. When the fin pitch increases, the tube length decreases because the heat transfer area gets larger.

Fig. 8 shows the pressure drop and the tube length variations with respect to the transverse pitch. When the transverse pitch increases, the pressure drop on the air side and the tube length are reduced. When the pitch increases, the air can pass through the heat exchanger more easily, reducing the pressure drop. In addition, the interface area between the air and the tubes increases, shortening the tube length. However, if transverse pitch increases, the height of heat exchanger increases, making the overall size of the heat exchanger problematic.

Fig. 9 shows the pressure drop and the tube length variations with respect to the fin's configuration and the number of rows. It is found that the louver fin is 17~30% shorter than the plate fin. The pressure drop of the air side is higher for the louver fin while that of the refrigerant side is slightly higher for the plate fin side. Actually, for a same the tube length, the air side flow resistance is larger for the louver fin, and the

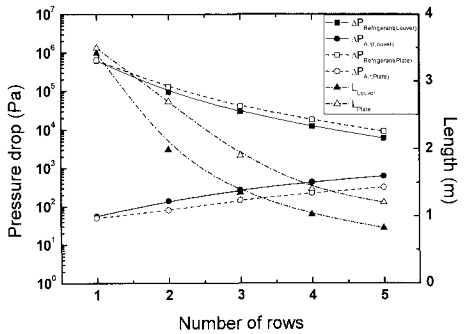

Fig. 9. Pressure drop and the tube length for each fin configuration.

pressure drop would be larger for the case of the louver fin. Therefore, it can be concluded that the pressure drop for the air side is the flow resistance-dominant while that for the refrigerant side is the tube length-dominant.

3. Conclusions

In this study, heat and mass transfer analysis for the moist air is conducted for the indoor heat exchangers using R410A. Standard values are input to each inlet condition in order to evaluate air outlet temperature, overall heat transfer coefficient, tube length, and pressure drops on the refrigerant and the air sides. The following conclusions are obtained from the present study.

(1) As the number of the row increases, the tube length decreases, and the pressure drop on the air side increases while the pressure drop on the refrigerant side decreases.

(2) It is found that when the number of the pipes increases, the tube length decreases, and the pressure drops on both sides decrease slightly. When the fin pitch increases, the tube length decreases while the pressure drop on the air side increases.

(3) It is found that the louver fin is 17~30% shorter than the plate fin. Is is also concluded that the pressure drop for the air side is the flow resistance-dominant while that for the refrigerant side is the tube length-dominant.

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