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Modeling and Performance Simulation of Gas Cooler for CO2 Heat Pump System

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ABSTRACT

The outlet temperature of gas cooler has a great effect on the efficiency of carbon dioxide heat pump system. In order to obtain a small approach temperature difference at gas cooler, near-counter flow type heat exchanger has been proposed, and larger heat transfer area is demanded. The optimum design of gas cooler involving the analysis of trade-offs between heat transfer performance and cost is desirable. In this study, a simulation model has been developed for fin-tube type gas cooler, and the air-side heat transfer correlation has been proposed. A developed model was confirmed by experimental results. The effects of geometric parameters, such as the tube arrangement, transverse tube spacing, longitudinal tube spacing and the number of tube rows and fin spacing on the performance of heat transfer were investigated using the developed model. This study suggested various simulation results for optimum designs of gas cooler.

1. INTRODUCTION

Largely due to the recent outbreak of environmental problems caused by global warming, many studies are currently on-going to adopt heat pump system using carbon dioxide, a natural refrigerant. As shown in Figure 1, the critical temperature of carbon dioxide is lower than that of air, which acts as heat sink. Thus, this refrigeration system is different from others using conventional refrigerants, the condensing process does not exist and hot gas refrigerant is cooled as single phase gas cooling process. In this study, a performance analysis model is proposed, using fin-tube heat exchanger as gas cooler of carbon dioxide heat pump system.

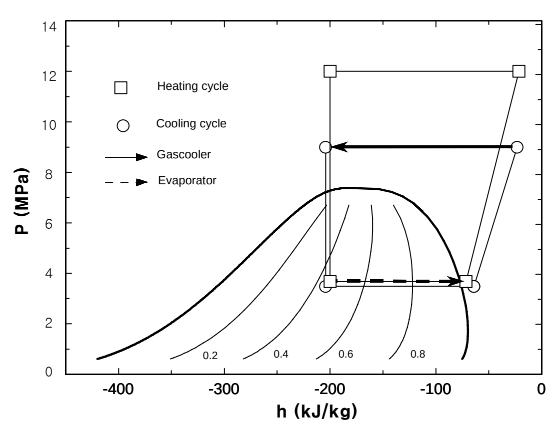


Figure 1: The p-h diagram of heat pump cycle using carbon dioxide

Hot carbon dioxide is introduced through in-tube of fin-tube heat exchanger, and air-cooling process takes place. For a fin-tube heat exchanger, air-side thermal resistance has a great portion among total thermal resistances. Thus, in order to improve performance of the heat exchanger, it is crucial to enhance performance of air-side heat transfer. Consequently, it is very important to understand air-side heat transfer characteristics for the design of a fin-tube gas cooler. Overall heat transfer coefficient is calculated from heat exchanger performance test data and ε-NTU relation considering the shape of heat exchanger and operating condition, and then air-side heat transfer coefficient can be obtained from overall heat transfer coefficient.

In this study, tube-by-tube method was used as the gas cooler model. This method is relatively simple and accurate results can be obtained considering variation of heat transfer characteristics of each tube. Measured air-side heat transfer coefficient can be expressed by the correlation of Reynolds number and Prandtl number. The effects of geometric parameters, such as the tube arrangement, transverse tube spacing, longitudinal tube spacing, the number of tube rows and fin spacing on the performance of heat exchanger were investigated using the developed model.

2. SIMULATION MODEL

Cross-flow type fin-tube heat exchanger widely used in air-conditioning system. It is composed of circular tubes made of copper and aluminum fins and adhered to each other by mechanical expansion of the tube. While the refrigerant flows through the copper tubes, air travels in cross-direction through the fins. There are several ways of analyzing the fin-tube heat exchanger; one involves using ε-NTU relation deduced by Hiller and Glicksman(1976) and Fischer and Rice(1981). Another is the tube-by-tube method developed from the study of Domanski(1989). It is assumed that the air between steps is unmixed, and each tube is handled as an independent component. This method is simple and allows obtaining accurate results. The model is widely applied in the analysis of air-cooled heat exchanger.

Using mean temperature difference to each independent tube which is cross flow type, heat transfer equation is shown as below.

$$Q = U \cdot A \cdot \Delta T_m \tag{1}$$

Enthalpy change of refrigerant in heat transfer process can be expressed as below.

$$Q = \dot{m}_r \cdot (i_i - i_e) \tag{2}$$

As micro-fin tube is used in this study, heat transfer correlation by Han et al.(2005) is used for refrigerant-side. By summing up heat resistances between the refrigerant and the air, overall heat transfer coefficient for a unit fin-tube can be expressed as follows.

$$U = \left[\frac{A_o}{h_r A_r} + \frac{A_o t}{A_{p,m} k_p} + \frac{A_o}{A_{p,o} h_c} + \frac{1}{h_o \left(1 - \frac{A_f}{A_o} (1 - \phi) \right)} \right]^{-1} \tag{3}$$

Here, ϕ , fin efficiency, is calculated from the method proposed by Schmidt(1945) and McQuiston and Parker(1982).

3. AIR-SIDE HEAT TRANSFER COEFFICIENT

3.1 Experiments

Experimental apparatus measuring the performance of the heat exchanger is shown in Figure 2. It is placed inside a thermal environment chamber where the temperature and humidity of air are carefully controlled. RTDs are installed to measure the air temperature and the humidity at the inlet and outlet of the heat exchanger. Air flow rate is measured using multi-nozzles. The schematic diagram of water circulation loop is shown in Figure 3. The loop is composed of a water tank, a mass flow meter, pumps, a constant temperature bath and the heat exchanger sample.

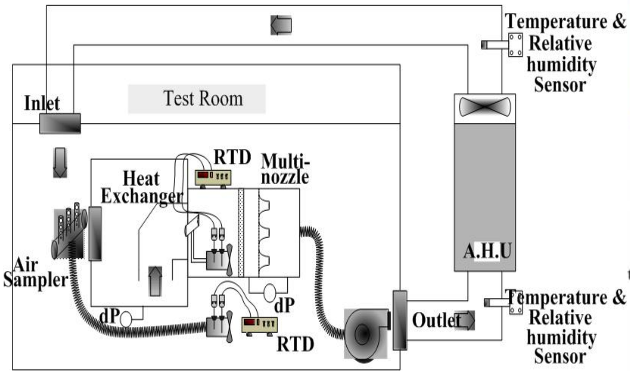


Figure 2: Heat exchanger performance test apparatus

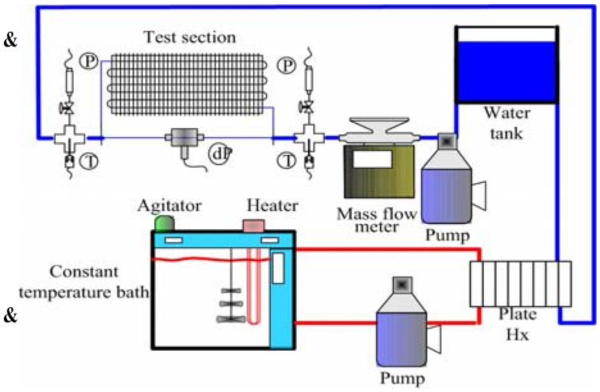


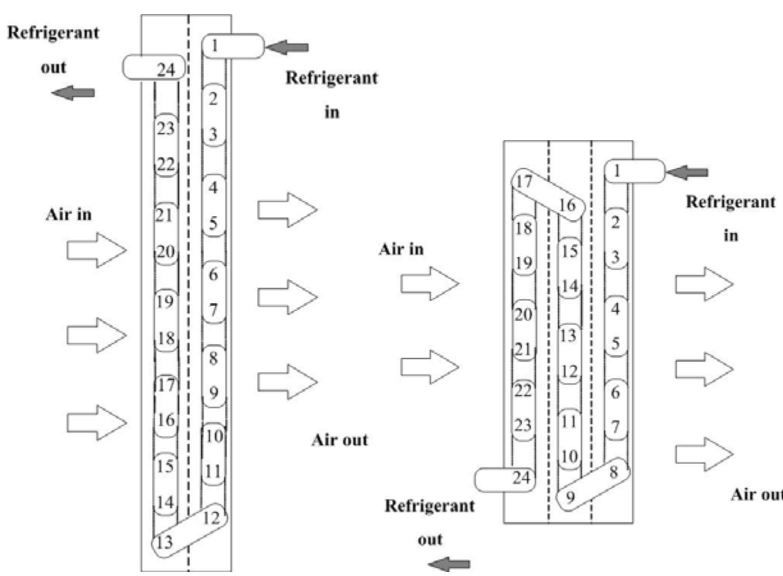
Figure 3: Water circulation loop

Table 1: Experimental test conditions

Inlet conditions		Value
Air	Temperature(℃)	15
	Relative Humidity (%)	60
	Frontal velocity (m/s)	0.5-2.5
Water	Temperature(℃)	35
	Mass flow rate(g/s)	15-100

The inlet and outlet temperatures of the heat exchanger are measured with T-type thermocouples. The test condition are shown in Table 1. Experiment has been conducted varying the frontal velocity of air and the mass flow rate of water while other conditions of air and water maintain same as Table 1. Experimental data are recorded on the PC using a data acquisition apparatus in a steady state.

Fin-tube heat exchanger has one row or more according to the applications. Fins used for fin-tube heat exchangers have various types, such as plate fin, wavy fin, louver fin, slit fin, and so on. Among them, slit fin and louver fin are used commonly due to their high performance. In this study, two types of fin-tube heat exchangers are used; one has two rows and the other has three rows which are shown in figure 4. The outer diameter of the tube is 7 mm. The specification of the fin-tube heat exchangers is shown in Table 2.



(a) 2 rows (b) 3 rows

Figure 4 : Fin-tube heat exchangers

Table 2: Geometric dimensions of heat exchangers

	2 rows	3 rows
Tube outside diameter [mm]	7	7
tube thickness [mm]	0.32	0.32
number of rows	2	3
fin material	Al	Al
fin type	louver	louver
tube type	micro fin tube	micro fin tube

Air-side heat transfer rate is calculated from the inlet and outlet temperatures and mass flow rate of the air. Water-side heat transfer rate is calculated using the equation (5).

$$Q_a = \dot{m}_a \cdot (i_{a,e} - i_{a,i}) \tag{4}$$

$$Q_w = \dot{m}_w \cdot (i_{w,i} - i_{w,e}) \tag{5}$$

Heat transfer rate of gas cooler is calculated by the arithmetic means of air and water-side heat transfer rates. Air-side heat transfer coefficient is obtained by performance simulation using the tube-by-tube model presented beforehand.

3.2 Results

The deviations of the heat transfer rates of water and air are within 5%. Fin efficiency obtained from the model is about 80%, and the air-side heat transfer correlation according to air velocity change is expressed as shown in the equation (6) and Figure 5.

$$Nu = C \, \mathrm{Re}_D^{\,m} \, \mathrm{Pr}^{1/3} \tag{6}$$

Here, Re_D is Reynolds number for tube diameter.

In order to verify the accuracy of the developed heat transfer correlation, performance simulation is carried out and whose results are compared to experimental results as shown in Figure 6. Simulation model predicts the heat transfer rate within 10% error.

To verify whether simulation model in this study could be applied to CO_2 gas cooler, performance simulation is carried out using experimental results of previous study for CO_2 gas cooler. Comparison of simulation results to experimental results is shown in Figure 7. The developed model describes the experimental results well and is suitable for the performance analysis of CO_2 gas cooler.

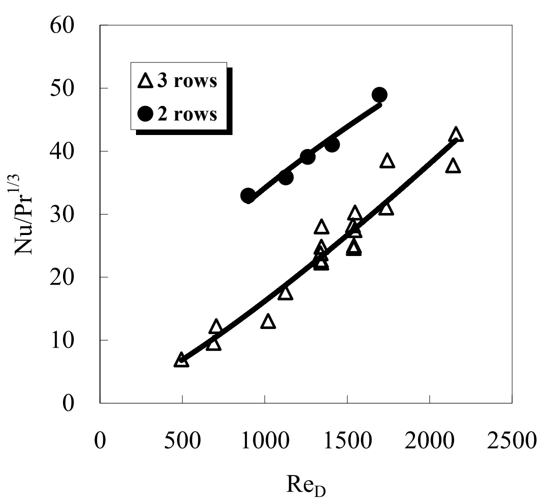


Figure 5: Developed air-side heat transfer correlation for 2 rows and 3 rows heat exchanger

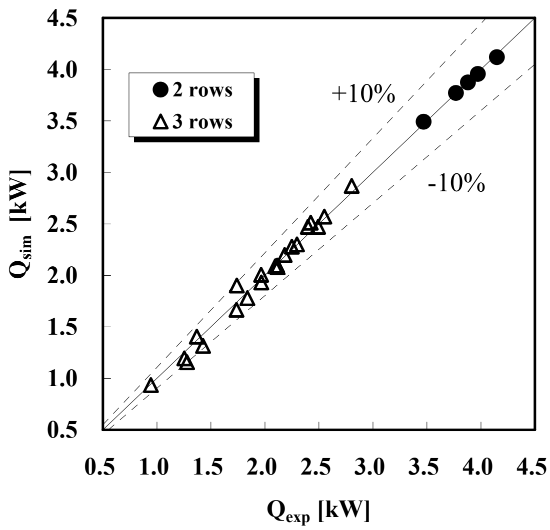


Figure 6: Comparison of heat transfer rate of simulation and experiment for water

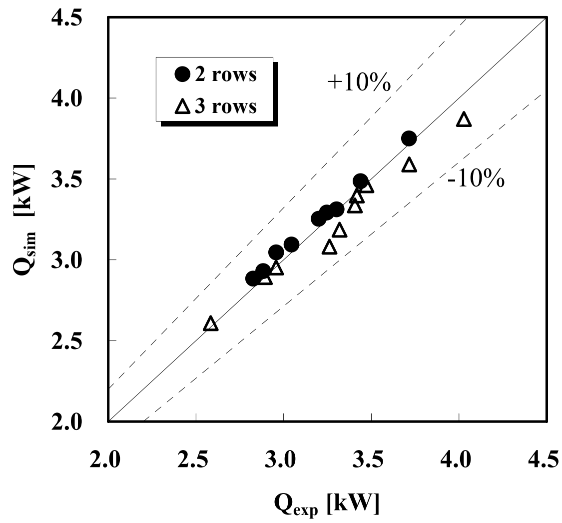


Figure 7: Comparison of heat transfer rate of simulation and experiment for CO_2 gas coolers

4. PERFORMANCE SIMULATION

4.1 Variations in heat transfer coefficient and fin efficiency

Air-side heat transfer coefficient is mainly affected by the air conditions and fin shape and by the changes in fin pitch, transverse tube pitch and longitudinal tube pitch of heat exchanger. In Figure 8, according to the design parameters of heat exchanger, variations in the heat transfer coefficient and fin efficiency are predicted using heat transfer model of Wang(1999). Heat transfer coefficient decreases when either the fin pitch or the transverse tube pitch increases. The heat transfer coefficient decreases by 4% and 10% when there is a 30% increase in the fin pitch and the transverse tube pitch respectively. The variation in heat transfer coefficient by the longitudinal tube pitch is very little. Fin efficiency is not much affected by the fin pitch or longitudinal tube pitch, but decreases largely when transverse tube pitch increases.

Because the air-side heat transfer coefficient proposed in this study can be applied only to a given fin shape, fin pitch, longitudinal tube pitch, and transverse tube pitch, the correction factor, *f* is used to calculate the air-side heat transfer coefficient with various geometric parameters.

$$h = f \times h^* \tag{7}$$

Here, upper suffice * stands for the reference condition and the h^* is the heat transfer coefficient of the tested heat exchanger in Figure 4 at a given air and refrigerant condition.

4.2 Performance with frontal area

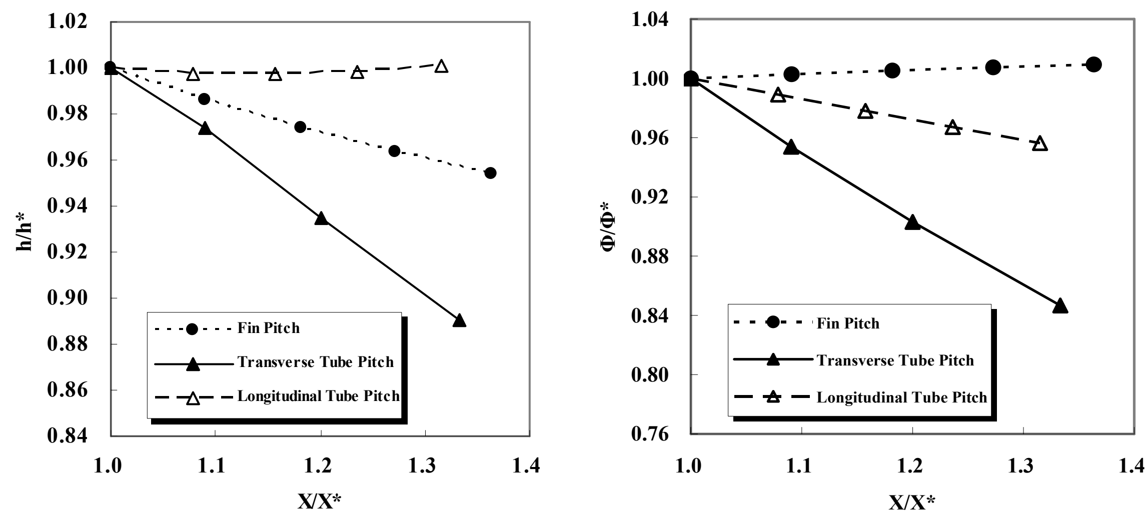
When frontal area is changed, heat transfer rate and minimum temperature difference of refrigerant and air at refrigerant outlet is shown in Figure 9. At reference condition, minimum temperature difference is very small as 2℃. The decrease of frontal area brings increase of minimum temperature difference and decrease of heat transfer rate due to reductions of heat transfer area and air flow rate.

4.3 Performance with number of circuits

The performance analysis result with number of circuits is shown in Figure 10 when the frontal area is constant. Refrigerant that is supplied to gas cooler is distributed to each circuit. If the number of circuits is 4, refrigerant that flows in each circuit is 1/4 of total refrigerant flow rate. Total mass flow rate of CO_2 is 70g/s. If the number of circuits decreases, the mass flow rate of refrigerant in each tube increases and heat transfer coefficient increases and pressure drop of refrigerant also increases. When refrigerant circuit is decreased from 4 to 3, performance is decreased about 1%, when number of circuits is 2, pressure drop is increased about 10 times, and heat transfer rate is decreased dramatically by 10% compared to that of 4 circuits.

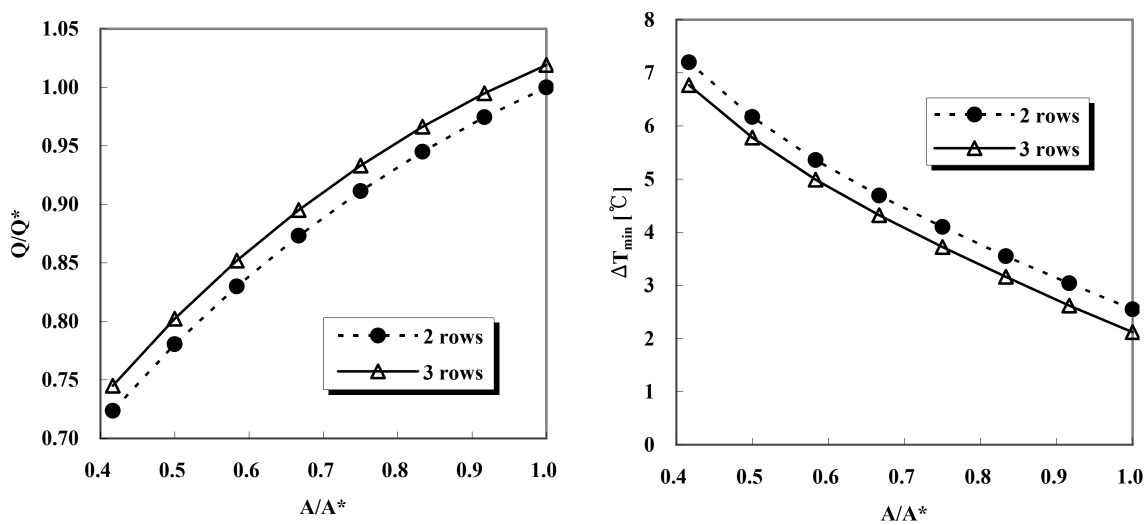
4.4 Performance with geometric parameters

The change of heat transfer rate and air-side pressure drop according to variations of fin pitch, transverse tube pitch, and longitudinal tube pitch are shown in Figure 11. When fin pitch or transverse tube pitch increases, heat transfer rate decreases due to reduction of the air-side heat transfer coefficient as shown in Figure 8. Contrarily, when longitudinal tube pitch increases, heat transfer rate and pressure drop increase due to increased heat transfer area.



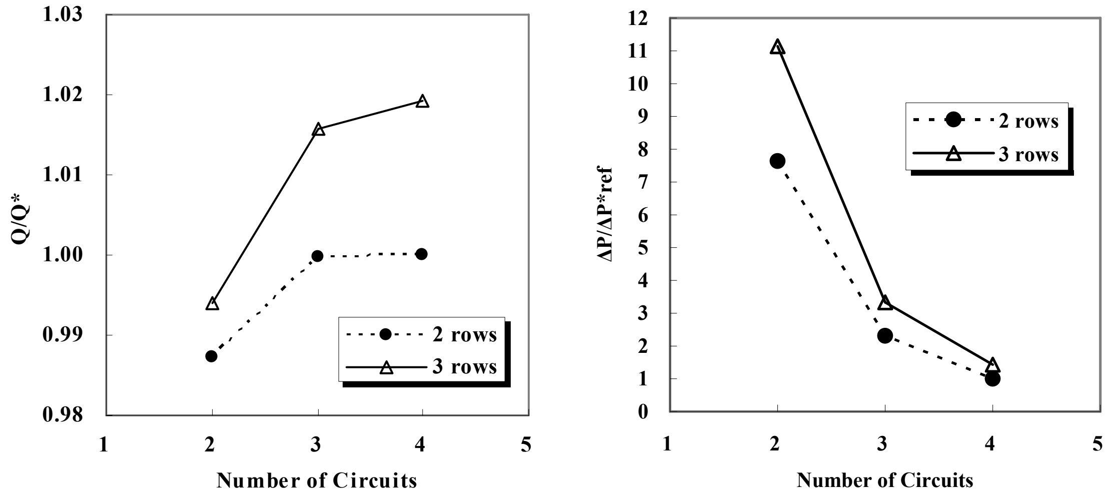
(a) heat transfer coefficient (b) fin efficiency

Figure 8: Variation in heat transfer coefficient and fin efficiency



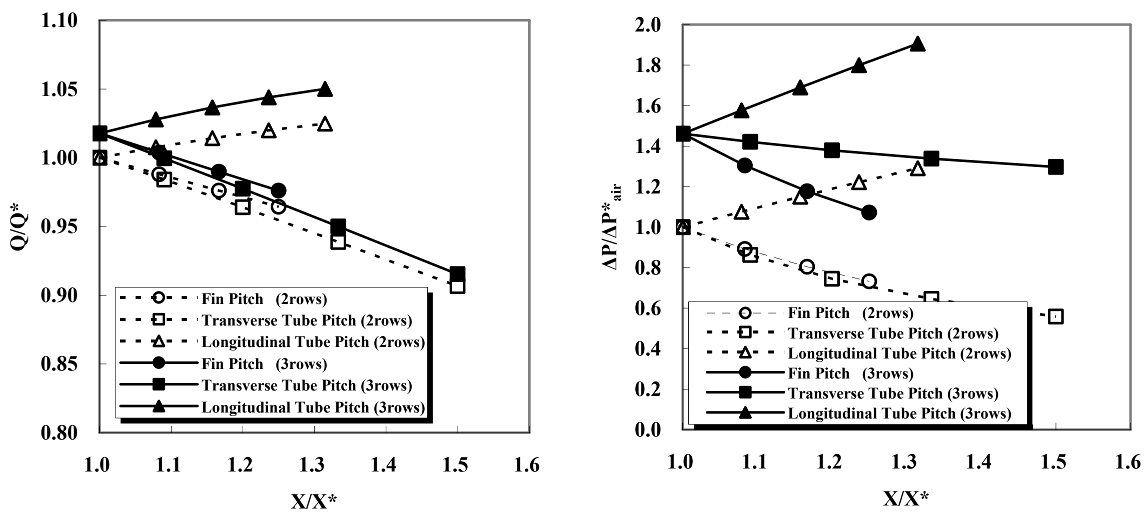
(a) Heat Transfer Rate (b) Minimum Temperature Difference

Figure 9: Simulation results with changes in frontal area



(a) Heat Transfer Rate (b) Refrigerant Pressure Drop

Figure 10: Simulation results with changes in number of circuits



(a) Heat Transfer Rate (b) Air-side Pressure Drop

Figure 11: Simulation results with changes in fin pitch, transverse tube pitch, and longitudinal tube pitch

6. CONCLUSIONS

Through experiment using water, air-side heat transfer correlation is developed and applied to the simulation of gas cooler with various design parameters. The results are as follows:

- The decrease of frontal area brings increase of minimum temperature difference and decrease of heat transfer rate.
- When the number of circuits is decreased, heat transfer rate is decreased and pressure drop is increased
- When fin pitch or transverse tube pitch increases, heat transfer rate and pressure drop decrease. When longitudinal tube pitch increases, heat transfer rate and pressure drop increased.

NOMENCLATURE

A	area	(m^2)
f	correction factor	(-)
h	heat transfer coefficient	(kW/m^2K)
i	enthalpy	(kJ/kg)
k	thermal conductivity	(kW/m-K)
$\dot{m}$	mass flow rate	(kg/s)
NTU	number of transfer unit	(-)
Nu	Nusselt number	(-)
Pr	Prandtl number	(-)
Q	heat transfer rate	(kW)
Re	Reynolds number	(-)
t	tube thickness	(m)
U	overall heat transfer coefficient	(kW/m^2K)
ΔT_m	mean temperature difference	(℃)
ΔP	pressure drop	(Pa),(kPa)
ε	effectiveness	(-)
φ	fin efficiency	(-)

Subscripts

a	air
D	diameter
exp	experiment
f	fin
i	inlet
o	outlet
r, ref	refrigerant
sim	simulation
w	water

Superscripts

*	reference condition

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