

2002

Comparison Of Air-Side Heat Transfer Coefficients Of Several Types of Evaporators Of Household Freezer/Refrigerators

T. H. Lee
LG Electronics

J. S. Lee
LG Electronics

S. Y. Oh
LG Electronics

M. Y. Lee
LG Electronics

K. S. Lee
Hanyang University

Follow this and additional works at: <http://docs.lib.purdue.edu/iracc>

Lee, T. H.; Lee, J. S.; Oh, S. Y.; Lee, M. Y.; and Lee, K. S., "Comparison Of Air-Side Heat Transfer Coefficients Of Several Types of Evaporators Of Household Freezer/Refrigerators" (2002). *International Refrigeration and Air Conditioning Conference*. Paper 611. <http://docs.lib.purdue.edu/iracc/611>

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.

Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at <https://engineering.purdue.edu/Herrick/Events/orderlit.html>

Comparison of Air-Side Heat Transfer Coefficients of Several Types of Evaporators of Household Freezer/Refrigerators

Tae-Hee Lee, Jang-Seok Lee, Se-Yoon Oh, Myung-Yul Lee, Research Engineers, Digital Appliance Research Lab.
 LG Electronics Inc. 327-23, Gasan-Dong, Geumchun-Gu, Seoul, 153-023 Korea;
 Tel.: +82-2-818-7871; Fax: +82-2-856-0313
 E-Mail: thl@lge.com

*Kwan-Soo Lee, School of Mechanical Engineering, Hanyang University, 17,
 Heangdang-Dong, Seongdong-Gu, Seoul, 133-791 Korea; Tel.: +82-2-2290-0426; Fax: +82-2-2295-9021
 E-Mail: ksleehy@hanyang.ac.kr *Author for Correspondence

ABSTRACT

In this study, the air-side heat transfer coefficients of several types of heat exchangers used as the evaporators in the household freezer/refrigerators are investigated. The types considered in this work are: spine finned tube type(in-lined tube array), continuous flat-plate fin-tube type(staggered tube array), and discrete flat-plate fin-tube type(in-lined tube array). The computer programs were coded using the section-by-section method similar to the tube-by-tube method to calculate the heat transfer coefficients from the experimental results of the heat exchangers. The input data for these programs are the flow rate and inlet temperature of air, the flow rate and inlet temperature of brine, and the heat transfer rate obtained from the experiment. The heat transfer coefficient of each experiment can be found by comparing the calculated heat transfer rate with the experimental heat transfer rate iteratively. The heat transfer correlations for each heat exchanger were driven from the calculated heat transfer coefficients using the least square method, and the error between the correlations and the calculated heat transfer coefficients were less than 5%. The result indicates that the air-side heat transfer performance of spine finned-tube type(in-lined tube array) heat exchangers shows the highest value among these three heat exchangers at dry condition.

NOMENCLATURE

A : Heat transfer area	NTU : Number of transfer unit
A _c : Cross sectional area	P : Perimeter
C : Heat capacity ratio	Q : Heat transfer rate
c _p : Isobaric specific heat	R : Thermal resistance
D : Diameter	T : Temperature
h : Heat transfer coefficient	U : Overall heat transfer coefficient
k : Thermal conductivity	V : Air velocity
L : Length	
<i>m</i> : Mass flow rate	

GREEK SYMBOLS

ε : Effectiveness	η : Efficiency
-------------------	----------------

SUBSCRIPTS

1 : Heat exchanger # 1	app : Approaching velocity
2 : Heat exchanger # 2	cor : Value by correlation
3 : Heat exchanger # 3	exp : Experimental results
a : Air-side	F : Fin area

i : Inlet
max : Maximum velocity
o : Outlet

T : Total area
t : Tube area
w : Water-side

INTRODUCTION

World-widely, about 80 million sets of household freezer/refrigerators are produced every year. About 40% of these household freezer/refrigerators adopt indirect cooling type using finned-tube type evaporator with a fan. Several types of finned-tube heat exchangers are used as an evaporator, and Table 1 illustrates the usage with photographs. The table shows that many companies use discrete flat plate finned tube type heat exchanger with in-lined tube array(heat exchanger #1), continuous flat plate finned tube type heat exchanger with staggered tube array(heat exchanger #2) or spine finned tube type heat exchanger(heat exchanger #3). Unfortunately, there exists only limited data about these heat exchangers, especially about on comparison of thermal performances.

The aim of this study is to compare the air-side heat transfer coefficients of the above-mentioned three types of heat exchangers. Experiments were conducted to evaluate the heat transfer performance of each heat exchanger. The heat transfer coefficients were numerically calculated by utilizing the experimental data and analysis program. The program used in this study is based on section-by-section method, a similar scheme with Domanski's tube-by-tube method[1].

Table 1. Heat exchangers used in a domestic freezer/refrigerator

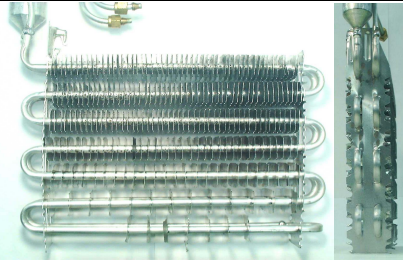
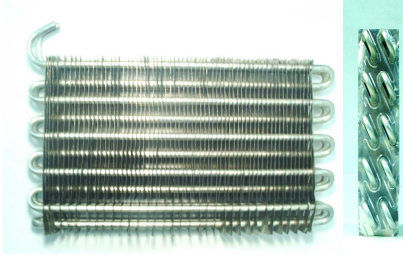

Heat exchanger No.	Type	Photo	Company
1	Discrete Flat plate-finned Tube		<ul style="list-style-type: none"> - LG - Samsung - Matsushita - Fujitsu - Westing House - BPL - Frigidaire
2	Continuous Flat plate-finned Tube		<ul style="list-style-type: none"> - Whirlpool - MayTag - Liebherr - Amana
3	Spine finned Tube		<ul style="list-style-type: none"> - GE

Table 2 Geometric parameters of the test heat exchangers

Heat exchanger No.		1	2	3
Fin type		Discrete flat plate	Continuous flat plate	Spine
Tube array		In-lined	Staggered	In-lined
Number of tube rows		7	10	5
Number of tube steps		2	←	←
Tube outer diameter(D _o)		8.5mm	7.9mm	9.4mm
Size of heat exchanger (mm)	Width	258	320	300
	Depth	60	50	65
	Height	210	190	153
Air-side heat transfer area (m ²)	Tube	0.0965	0.159	0.0881
	Fin	0.696	0.892	0.478
	Total	0.793	1.05	0.566

EXPERIMENT

Experimental set-up and procedure

The detailed specifications of the test heat exchangers illustrated in Table 1 are shown in Table 2. The schematic diagram of the experimental set-up is shown in Fig. 1. This apparatus is constructed as an open wind tunnel. The air flow rates were controlled by a centrifugal fan with inverter and measured by the nozzle and the differential manometer with uncertainty of $\pm 1.5\%$ FS. The inlet and outlet temperature of the air across the sample heat exchanger were measured by 5 resistance temperature detectors (RTD), respectively. The uncertainty of these RTDs are $\pm 0.05^\circ\text{C}$. The air mixer was provided for accurate measurement of the outlet air temperature at the end of the test section. The inlet air temperatures were varied from 22°C to 28°C . The mass flow rate of brine (ethylene glycol 100%) contained in a reservoir was controlled by a gear pump with inverter and maintained about 50kg/h through all the tests. The inlet temperatures of the brine are maintained at 35°C or 40°C . The test section was made of 30mm thick acrylic plates and the 80mm thick thermal insulators (styrene form) were attached on the plates.

The heat transfer rate of the test heat exchanger is taken as an arithmetic average of the air-side and water-side heat transfer rates, namely;

$$Q_a = (\dot{m}c_p)_a (T_{a,o} - T_{a,i}) \quad (1a)$$

$$Q_w = (\dot{m}c_p)_w (T_{w,i} - T_{w,o}) \quad (1b)$$

$$Q = \frac{Q_a + Q_w}{2} \quad (2)$$

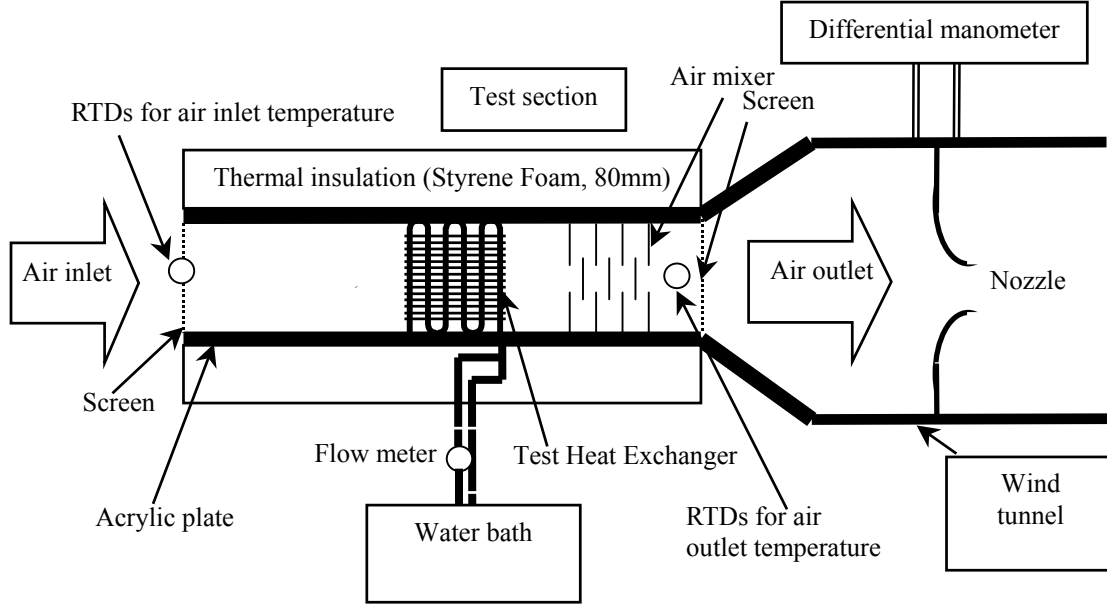


Fig. 1 Schematic diagram of experimental apparatus

In the experiments, the water temperatures were higher than the air temperature to avoid water condensing.

Data Analysis with computer program

In this study, the computer program based on the section-by-section method was used to calculate the air-side heat transfer coefficient from the test results. The test condition and result and the geometry of the test heat exchanger are the input data of the program. The air-side heat transfer coefficient is the output data of this program.

The section-by-section method is similar to the tube-by-tube method of Domanski [1]. In the tube-by-tube method, each tube with associated fins is treated as a small heat exchanger. While in the section-by-section method, each tube is divided into several sections and these sections have their own identity. The ε -NTU relation to analyze the sectional heat exchanger is as follows;

$$\varepsilon = \frac{1}{C} \left[1 - e^{-C \cdot (1 - e^{-NTU})} \right] \quad (3)$$

where,

$$C = \frac{\left(\dot{m} c_p \right)_w}{\left(\dot{m} c_p \right)_a} \quad (4)$$

and,

$$NTU = \frac{UA}{\left(\dot{m} c_p \right)_a} \quad (5)$$

where, UA is as follows ;

$$UA = \left[\frac{1}{h_w A_w} + R_t + \frac{1}{\eta_s h_a A_T} \right]^{-1} \quad (6)$$

h_w is water side heat transfer coefficient and calculated using the Dittus-Boelter's correlation, A_w is the water side heat transfer area. R_t , η_s , h_a and A_T are the thermal resistance of the tube wall, the surface efficiency of the air-side heat transfer area, the air-side heat transfer coefficient and the air-side heat transfer area respectively. η_s is defined as follows ;

$$\eta_s = \frac{A_t + \eta_F A_F}{A_T} \quad (7)$$

To calculate the fin efficiency in the equation (7), η_F , the approximation method described by Schmidt [2] was used ;

$$\eta_F = \frac{\tanh(m_f r \phi)}{m_f r \phi} \quad (8)$$

where,

$$m_f = \sqrt{\frac{2h_a}{k_f t_f}} \quad (9)$$

$$\phi = \left(\frac{R_{eq}}{r} - 1 \right) \left[1 + 0.35 \ln \left(\frac{R_{eq}}{r} \right) \right] \quad (10)$$

For staggered tube array ;

$$\frac{R_{eq}}{r} = 1.27 \frac{X_M}{r} \sqrt{\frac{X_L}{X_M} - 0.3} \quad (11)$$

for in-lined tube array ;

$$\frac{R_{eq}}{r} = 1.28 \frac{X_M}{r} \sqrt{\frac{X_L}{X_M} - 0.2} \quad (12)$$

The fin efficiency of the spine fin can be calculated as follows;

$$\eta_F = \frac{\tanh(m_p L)}{m_p L} \quad (13)$$

where,

$$m_f = \sqrt{\frac{h_a P}{k_f A_c}} \quad (14)$$

and, L is the length of the spine fin. P and A_c in equation (14) are perimeter and cross sectional area of the spine fin respectively.

Data reduction

The procedure to calculate the air-side heat transfer coefficient is as follows;

- 1) Input the test condition and heat transfer rate to the computer program.
- 2) Assume an air-side heat transfer coefficient
- 3) Perform the thermal performance analysis with the input data and assumed air-side heat transfer coefficient.
- 4) Calculate a new air-side heat transfer coefficient using the secant method, as follows;

$$h_a^{n+1} = h_a^n - (Q^n - Q_{exp}) \frac{h_a^n - h_a^{n-1}}{Q^n - Q^{n-1}} \quad (15)$$

- 5) Repeat 3) and 4) until the error between the calculated heat transfer rate and measured value satisfies the convergence criterion.

The study on the effect of fin spacing of flat plate finned-tube heat exchanger to the heat transfer coefficients was performed by Rich [3]. By his study, the heat transfer coefficients are not a function of the fin spacing, and the average heat transfer coefficient is a function of Reynolds number. In the calculation, the authors assumed that the air-side heat transfer coefficient is not a function of the fin spacing.

RESULTS AND DISCUSSION

Fig. 2 shows the variation of air-side heat transfer coefficients with respect to approaching air velocity. The heat transfer coefficient of the heat exchanger #2 and that of the heat exchanger #3 show similar values each other, while these heat exchangers record 14% and 15% higher value than the heat exchanger #1. The authors observed that the air-side pressure drop of the heat exchanger #2 showed about 260% of that of the heat exchanger #1, and the heat exchanger #1 showed about 150% of the heat exchanger #3 at the same approaching air velocity. This result implies that the heat exchanger #3 has the higher air-side performance than the others at a same air flow rate at a dry condition.

On the other hand, the heat exchanger #1 shows higher Nusselt number than the heat exchanger #2 at a same Re_D , as shown in Fig. 3. This figure shows the variation of air-side Nusselt number with respect to Reynolds number. The definition of the Nusselt number and Reynolds number are as follow;

$$Nu_D = \frac{h_a D_o}{k_a} \quad (16)$$

$$Re_D = \frac{\rho_a V_{a,\max} D_o}{\mu_a} \quad (17)$$

where, the maximum air velocity is calculated as follows;

$$V_{a,\max} = \frac{m_a}{\rho_a A_{\min}} \quad (18)$$

The heat exchanger #3 shows the highest Nusselt number among these three heat exchangers.

In figures 2 and 3, one can know that the spine finned tube heat exchanger always shows higher air-side performance than the others in the dry condition. This is because the airflow length over the fin is too short for the thermal boundary layer to grow enough.

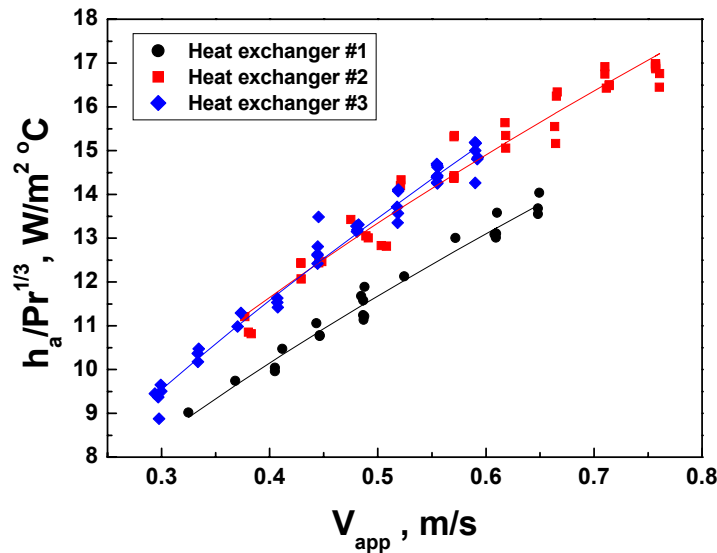


Fig. 2 The variation of heat transfer coefficients with approaching air velocity

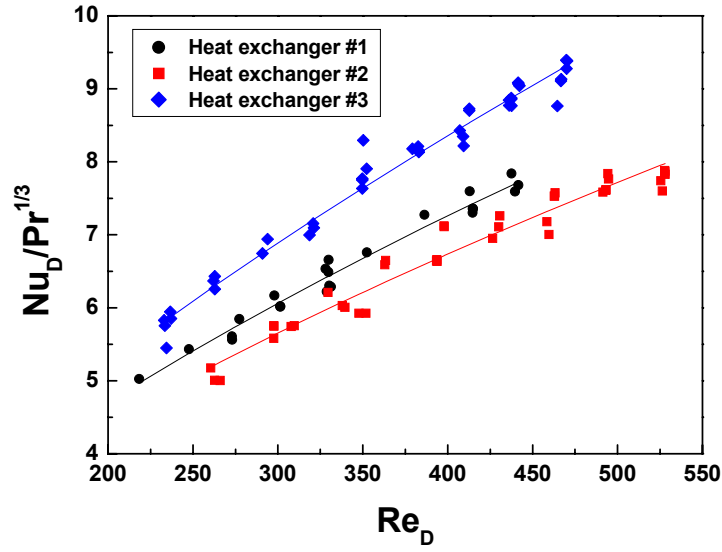


Fig. 3 The variation of the air-side heat transfer coefficient with Reynolds number.

The correlations between the air-side heat transfer coefficients of these heat exchangers and Reynolds number as follow;

$$Nu_{D,1} = \frac{h_{a,1}D_o}{k_a} = 0.170 \cdot Re_D^{0.63} \cdot Pr^{1/3} \quad (19)$$

$$Nu_{D,2} = \frac{h_{a,2}D_o}{k_a} = 0.162 \cdot Re_D^{0.61} \cdot Pr^{1/3} \quad (20)$$

$$Nu_{D,3} = \frac{h_{a,3}D_o}{k_a} = 0.148 \cdot Re_D^{0.67} \cdot Pr^{1/3} \quad (21)$$

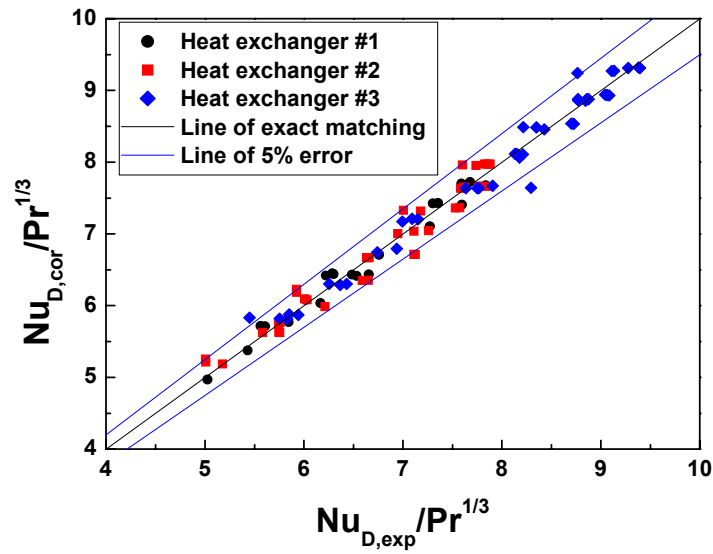


Fig. 4 Comparison of Nusselt numbers between correlations and experimental data.

The comparison of the heat transfer correlations, Eqs. (19), (20), and (21) developed in this study, with experimental data is shown in Fig. 4. The proposed heat transfer correlations can describe 95% of the heat transfer coefficients within $\pm 5\%$.

CONCLUSIONS

The air-side heat transfer performance of the three types of heat exchanger used as evaporators in household freezer/refrigerators are investigated. The spine finned tube heat exchanger has the best air-side heat transfer performance at a dry condition among the heat exchangers considered in this study. The air-side heat transfer correlations for these three types of finned tube heat exchangers are proposed and these correlations can describe 95% of the experimental data within $\pm 5\%$.

REFERENCES

- Domanski, P.A. , "Simulation of an evaporator with nonuniform one-dimensional air distribution", ASHRAE Transactions : Symposia, pp. 793-802, 1991
- Schmidt, T. E., "Heat transfer calculations for extended surfaces", Refrigerating Engineering, Vol. 57, pp. 351-357
- Rich. D. G., "The effect of fin spacing on the heat transfer and friction performance of multi-row, smooth plate fin-and-tube heat exchangers" ASHRAE Transactions, Vol. 79. No. 2, pp.137-145