Available online at www.sciencedirect.com
SCIENCE DIRECT®
International Journal of Heat and Fluid Flow 25 (2004) 1006–1014
International Journal of HEAT AND FLUID FLOW
www.elsevier.com/locate/ijhff

A study on the correlation between the thermal contact conductance and effective factors in fin–tube heat exchangers with 9.52 mm tube

Jin Jeong [a], Chang Nyung Kim [b,*], Baek Youn [c], Young Saeng Kim [c]

[a] Department of Mechanical Engineering, Graduate School of Kyunghee University, South Korea
[b] College of Mechanical and Industrial System Engineering, Kyunghee University, College of Advanced Technology, 1 Seochon, Kihung, Yongin, Kyunggi 449-701, South Korea
[c] System Appliances Division, Samsung Electronics Co., LTD, South Korea

Received 16 October 2003; accepted 17 March 2004
Available online 17 June 2004

Abstract

The thermal contact resistance is a principal parameter interfering with heat transfer in a fin–tube heat exchanger. However, the thermal contact resistance in the interface between tubes and fins has not been clearly investigated. The objective of the present study is to examine the thermal contact conductance for various fin–tube heat exchangers with tube diameter of 9.52 mm and to find a correlation between the thermal contact conductance and effective factors such as expansion ratio, fin type, fin spacing and hydrophilic coating. In this study, experiments have been conducted only to measure heat transfer rate between hot and cold water. To minimize heat loss to the ambient air by the natural convection fin–tube heat exchangers have been placed in an insulated vacuum chamber. Also, a numerical scheme has been employed to calculate the thermal contact conductance with the experimental data. As a result, a new correlation including the influences of expansion ratio, slit of fin and fin coating has been introduced, and the portion of each thermal resistance has been estimated in the fin–tube heat exchangers with 9.52 mm tube.

Keywords: Fin–tube heat exchanger; Heat transfer rate; Thermal resistance; Thermal contact resistance; Thermal contact conductance; Overall heat transfer coefficient

1. Introduction

In modern industrial society, fin–tube heat exchangers play an important role in power stations, chemical plants, refrigerating industries, aircrafts, automobiles, etc. So far, many researchers have studied to enhance the efficiency of the fin–tube heat exchanger. The thermal contact resistance, however, has not been focused deeply and occasionally has been overlooked owing to complexities of heat transfer through interfaces and difficulties in measurements. Generally, fin–tube heat exchangers are manufactured through mechanical expansion to tighten the contact between tubes and fins. Yet, the features of heat transfer through interfaces have not been clarified because of the irregular contact of interface.

A study on the effect of thermal contact resistance in a fin–tube heat exchanger was first attempted by Dart (1959). In his study, thermal contact resistances were tested with several samples with two passages, which were one for cold and the other for hot water. To minimize the influence of the natural convection, the tube was placed in an adiabatic chamber. The results were compared to that in soldered fins. Eckels (1977) and, Eckels and Rabas (1987) predicted the thermal contact conductance, varying the number of fin, the fin thickness, and the diameter of tube in the wet and the dry fin–tube heat exchangers. In addition, they improved the empirical method, based on Dart's method, including error analysis. Abuebid (1984) investigated the thermal contact resistance with plate-finned tube heat exchangers placed in a vacuum. He performed an error

* Corresponding author. Tel.: +82-31-201-2578/2869; fax: +82-31-202-9715/8106.
E-mail address: cnkim@khu.ac.kr (C. Nyung Kim).

0142-727X/$ - see front matter © 2004 Elsevier Inc. All rights reserved.
doi:10.1016/j.ijheatfluidflow.2004.03.005

Nomenclature

A_c	contact area between the fin collar and tube surface (m^2)	Pr	Prandtl number
A_f	cross-sectional area of the fin (m^2)	$\dot{Q}_c$	heat transfer rate of the cold water (W)
A_i	inside area of the tube (m^2)	$\dot{Q}_h$	heat transfer rate of the hot water (W)
A_m	log mean area of the tube (m^2)	$\delta\dot{Q}_h$	inaccuracy of heat transfer rate from hot to cold water (W)
c	thermal contact conductance (W/m^2 °C)	R_c	thermal contact resistance (°C/W)
C_f	the constant of hydrophilic fin coating (W/m^2 °C)	Re	Reynolds number
C_p	specific heat (J/kg °C)	R_f	thermal conduction resistance of the fin (°C/W)
D	nominal tube diameter (m)	R_{hc}	thermal convection resistance of cold water (°C/W)
D_{ball}	diameter of expansion ball (m)	R_{hh}	thermal convection resistance of hot water (°C/W)
D_i	inner diameter of the tube (m)	R_t	thermal conduction resistance of the tube (°C/W)
D_{min}	minimum diameter of the tube (m)	R_{total}	sum of the all thermal resistances (°C/W)
D_o	outer diameter of the tube (m)	R_{xh}	axial convective resistance in association with advection of thermal energy (°C/W)
E	expansion ratio (%)	R_{xt}	axial conduction resistance in tube material (°C/W)
h_c	convective heat transfer coefficient of the cold water (W/m^2 °C)	S_f	the constant of hydrophilic fin coating
h_h	convective heat transfer coefficient of the hot water (W/m^2 °C)	t_f	thickness of the fin (m)
k_f	thermal conductivity of the fin (W/m °C)	t_t	thickness of the tube (m)
k_t	thermal conductivity of the tube (W/m °C)	T_c	temperature of the cold water (°C)
l_{eq}	equivalent length of the fin (m)	T_h	temperature of the hot water (°C)
$\dot{m}_c$	mass flow rate of the cold water (kg/s)	ΔT_h	temperature drop of hot water (°C)
$\dot{m}_h$	mass flow rate of the hot water (kg/s)	U	overall heat transfer coefficient (W/m^2 °C)
n_f	number of fins (per unit length)	w	width of the fin (m)
N_f	fin number		
Nu	Nusselt number		
P_f	fin pitch (spacing) (m)		

analysis, which is similar to the Eckels' method. But the error band was narrower. Shah (1986) investigated the effect of pressure distribution on the collar analyzing the temperature distribution in the fin and collar. Sheffield et al. (1987) considered the contact pressure as the significant factor of the thermal contact resistance. They introduced the influence of surface hardness and studied the correlation between contact pressure and expansion interference (hardness and roughness). Stubblefield et al. (1996) investigated the heat loss by thermal contact resistance using the insulated pipe and presented a simple method to predict the effect of contact resistance. Salgon et al. (1997) theoretically predicted the thermal contact conductance which was presented as a function of contact pressure and compared to experimental data. But, the previous studies were insufficient to identify factors affecting the thermal contact conductance because several factors were not considered, and many assumptions were used in the mathematical models.

In this study, we evaluate the thermal contact conductance using experimental–numerical method. Here, thermal contact conductances have been evaluated on various fin–tube heat exchangers with 9.52 mm tube. And a new correlation between the thermal contact conductance and effective factors such as expansion ratio, fin type, fin spacing, and hydrophilic coating has been developed. Additionally, the portions of the thermal resistances to total resistance have been computed in all samples.

2. Theory of the experimental–numerical method

The fin–tube heat exchangers manufactured through mechanical expansion of tubes are composed of aluminum fins and grooved copper tubes with diameter of 9.52 mm, and have eight tubes in a row as depicted in Fig. 1. The fin–tube heat exchanger is set in a vacuum chamber ensuring that the aluminum fins function only as conduction media with a minimum of natural convection. To minimize the heat transfer between hot and cold water through the chamber wall, inlet and outlet tubes of cold water are neighbored in the upper part of the chamber wall and those of hot water are placed in the lower part.

Fig. 1. A fin–tube heat exchanger with 9.52 mm tube used in the experiment.

To evaluate influences by variance of the features of tube and the fin, the experiment is conducted on the fin–tube heat exchangers with various expansion ratios of tubes, fin spacings, fin coatings (hydrophilic and non-hydrophilic coating fins), and fin types (wave fin, wave-slit fin, and D-fin) as shown in Fig. 2. The configuration of the fin–tube heat exchanger is presented in Table 1. To improve the reliability of the experiment, a pair of heat exchangers of the same specification were manufactured and tested, respectively, and 10 pairs (total 20 samples) were tested repeatedly.

As shown in Fig. 3, the experimental apparatus consists of vacuum chamber, vacuum pump, pair of constant temperature reservoirs, pair of water pumps, pair of mass flow meters, vacuum gauge, thermo sensors, etc. The procedure of the experiment is as follows. With the vacuum of about 3 mmHg, hot and cold water are circulated through the fin–tube heat exchanger. As the heat transfer reaches steady state, the recorder saves all the experimental data. The obtained data in the experiment are used as input data for the numerical calculation. The detailed explanation about the experimental method has been suggested in our previous research (Kim et al., 2003).

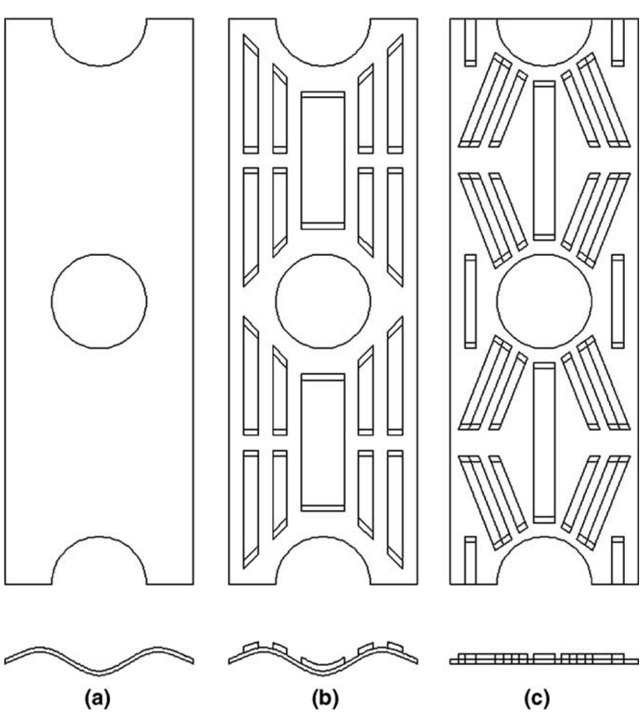

Fig. 2. Schematic of fins. (a) Wave fin, (b) wave-slit fin and (c) D-fin.

As shown in Table 2, the inlet temperatures of hot water have ranged from 51.1 to 56.3 °C, and those of the cold water have ranged from 17.8 to 23.7 °C. The flow rates of hot and cold water have, respectively, ranged 1.38–1.76 kg/min and 1.36–1.98 kg/min. The measurement errors of the temperatures and the flow rates are about ±0.1 °C and ±0.1%, respectively. The differences between heat transfer rates based on the hot and cold water have been estimated about 1–5%. These small differences may be due to heat loss through the chamber wall, errors in properties evaluation, and measurement errors of instrumentation.

The thermal circuit of a hot and cold tube is shown in Fig. 4(a). If the hot and cold tube is one path, respectively, a radial heat transfer rate ($\mathrm{d}\dot{Q}$) through only an infinitesimal area ($\mathrm{d}A_\mathrm{i}$) between hot and cold water is as follows.

$$\mathrm{d}\dot{Q} = U(T_\mathrm{h} - T_\mathrm{c})\mathrm{d}A_\mathrm{i} \tag{1}$$

$$\frac{1}{U\mathrm{d}A_\mathrm{i}} = R_\mathrm{total} = R_\mathrm{hh} + 2R_\mathrm{t} + 2R_\mathrm{c} + R_\mathrm{f} + R_\mathrm{hc} \tag{2}$$

where

$$R_\mathrm{hh} = 2/(h_\mathrm{h}\,\mathrm{d}A_\mathrm{i}) \tag{3}$$

$$R_\mathrm{t} = 2t_\mathrm{t}/(k_\mathrm{t}\,\mathrm{d}A_\mathrm{m}) \tag{4}$$

$$R_\mathrm{c} = 2/(c\,\mathrm{d}A_\mathrm{c}) \tag{5}$$

$$R_\mathrm{f} = l_\mathrm{eq}/(k_\mathrm{f}\,\mathrm{d}A_\mathrm{f}) \tag{6}$$

$$R_\mathrm{hc} = 2/(h_\mathrm{c}\,\mathrm{d}A_\mathrm{i}) \tag{7}$$

To calculate the thermal conductance using only the previously mentioned mathematical model (Eqs. (1) and (2)) is not adequate in this fin–tube heat exchanger problem because the heat exchangers are composed of several paths which are connected with fins each other as shown in Fig. 1. Therefore, the numerical calculation using the global energy balance is considered in the present study, and the heat exchanger is divided into small elements as shown in Fig. 4(b).

Correlations for convective heat transfer of water flow in a 9.52 mm grooved tube was presented by Park et al. (1997) as;

$$Nu = 0.00172 \times Re^{1.12} \times Pr^{0.3} \quad (3000 < Re < 21,000) \tag{8}$$

Therefore, the convective heat transfer coefficients (h_h and h_c) of the hot and cold water can be determined

Table 1
Configuration of fin–tube heat exchangers with 9.52 mm tubes

<table>
<tr><th></th><th>D [mm]</th><th>D_{ball} [mm]</th><th>Fin type</th><th>Hydrophilic coating</th><th>P_f [mm]</th><th>D_o [mm]</th><th>D_i [mm]</th><th>w [mm]</th><th>t_f [mm]</th><th>N_f</th></tr>
<tr><td>Case 1</td><td rowspan="20">9.52</td><td rowspan="6">9.04</td><td rowspan="2">Wave</td><td rowspan="2">without</td><td rowspan="2">1.5</td><td rowspan="2">9.98</td><td rowspan="2">9.38</td><td rowspan="2">21.65</td><td rowspan="2">0.11</td><td>277</td></tr>
<tr><td>Case 2</td><td>276</td></tr>
<tr><td>Case 3</td><td rowspan="4">D-fin</td><td rowspan="4">without</td><td rowspan="2">1.3</td><td rowspan="2">9.98</td><td rowspan="2">9.38</td><td rowspan="2">21.65</td><td rowspan="2">0.11</td><td>309</td></tr>
<tr><td>Case 4</td><td>312</td></tr>
<tr><td>Case 5</td><td rowspan="2">1.7</td><td rowspan="2">9.98</td><td rowspan="2">9.38</td><td rowspan="2">21.65</td><td rowspan="2">0.11</td><td>235</td></tr>
<tr><td>Case 6</td><td>232</td></tr>
<tr><td>Case 7</td><td rowspan="12">9.17</td><td rowspan="8">Wave</td><td rowspan="4">without</td><td rowspan="2">1.3</td><td rowspan="2">10.12</td><td rowspan="2">9.52</td><td rowspan="2">21.65</td><td rowspan="2">0.11</td><td>314</td></tr>
<tr><td>Case 8</td><td>314</td></tr>
<tr><td>Case 9</td><td rowspan="2">1.5</td><td rowspan="2">10.12</td><td rowspan="2">9.52</td><td rowspan="2">21.65</td><td rowspan="2">0.11</td><td>277</td></tr>
<tr><td>Case 10</td><td>275</td></tr>
<tr><td>Case 11</td><td rowspan="2">with</td><td rowspan="2">1.5</td><td rowspan="2">10.12</td><td rowspan="2">9.52</td><td rowspan="2">21.65</td><td rowspan="2">0.11</td><td>285</td></tr>
<tr><td>Case 12</td><td>282</td></tr>
<tr><td>Case 13</td><td rowspan="2">without</td><td rowspan="2">1.7</td><td rowspan="2">10.12</td><td rowspan="2">9.52</td><td rowspan="2">21.65</td><td rowspan="2">0.11</td><td>248</td></tr>
<tr><td>Case 14</td><td>247</td></tr>
<tr><td>Case 15</td><td rowspan="4">Wave -Slit</td><td rowspan="4">without</td><td rowspan="2">1.3</td><td rowspan="2">10.12</td><td rowspan="2">9.52</td><td rowspan="2">21.65</td><td rowspan="2">0.11</td><td>307</td></tr>
<tr><td>Case 16</td><td>309</td></tr>
<tr><td>Case 17</td><td rowspan="2">1.7</td><td rowspan="2">10.12</td><td rowspan="2">9.52</td><td rowspan="2">21.65</td><td rowspan="2">0.11</td><td>244</td></tr>
<tr><td>Case 18</td><td>242</td></tr>
<tr><td>Case 19</td><td rowspan="2">9.30</td><td rowspan="2">wave</td><td rowspan="2">without</td><td rowspan="2">1.5</td><td rowspan="2">10.14</td><td rowspan="2">9.54</td><td rowspan="2">21.65</td><td rowspan="2">0.11</td><td>274</td></tr>
<tr><td>Case 20</td><td>276</td></tr>
</table>

from $h_h = Nu(k_h/D_i)$, $h_c = Nu(k_c/D_i)$, and Eq. (8). The ranges of h_h and h_c in the present study are about 2800–3700 and 2100–3400 W/m^2 °C, respectively. The Reynolds number is about 4500–8500.

As presented in the Eq. (6), to simplify the calculation of the thermal resistance of fin (R_f) we introduced an effective conduction length (equivalent length) instead of a real conduction length through a fin. While the distance between the centers of the tubes (l) in the real geometry of a fin is 25.0 mm, the calculated equivalent length (l_{eq}) is 19.8 mm. The specific procedure of numerical calculation has been suggested in our previous research (Kim et al., 2003).

As presented in Table 3, the results of numerical calculation include the Reynolds number, the convective heat transfer coefficients, the overall heat transfer coefficient, the thermal contact conductance, and the portions of all thermal resistances between the hot and the cold water. However, other values except the thermal contact conductance should not be used as the characteristics comparing to each case because these values change to each test condition. The thermal contact conductances range about from 6000 to 17,000 W/m^2 °C depending on the features of heat exchangers. In the two heat exchangers of the same specification, the differences of the thermal contact conductances have been calculated to be about 1–20%. It may be due to the differences of the number of fins and to the experimental error. Therefore, the influences of the effective factors have been evaluated using the arithmetic mean values of the two thermal contact conductances.

The sensitivity of the thermal contact conductance to the temperature drop in the numerical calculation, $\frac{\partial c}{\partial(\Delta T_h)}$, is about 1.26×10^4 W/m^2 °C in the first case (the symbol,

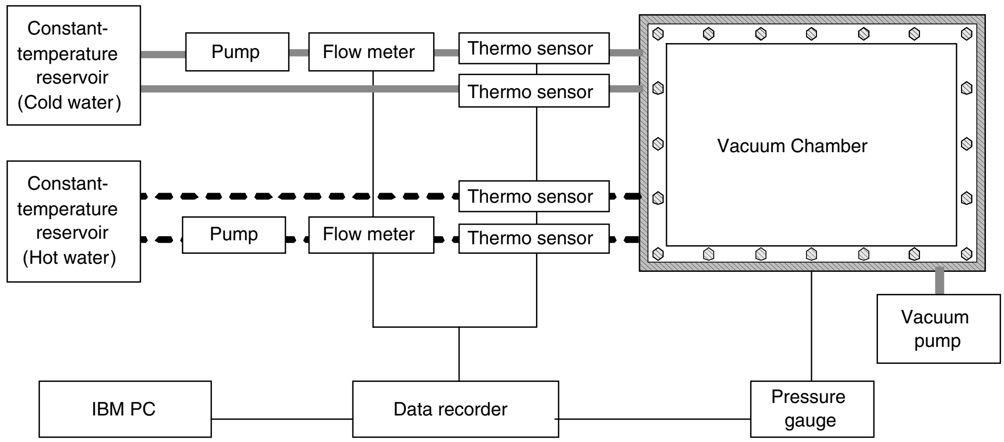

Fig. 3. Schematic diagram of the experimental apparatus.

Table 2
Experimental data for energy balance between hot and cold water

	T_{ci} (°C)	T_{co} (°C)	T_{hi} (°C)	T_{ho} (°C)	$\dot{m}_c$ (kg/min)	$\dot{m}_h$ (kg/min)	$\dot{Q}_c$ (W)	$\dot{Q}_h$ (W)	$\frac{(\dot{Q}_h-\dot{Q}_c)}{\dot{Q}_h} \times 100$ (%)
Case 1	19.8	25.5	52.4	45.9	1.60	1.46	635.12	661.82	4.03
Case 2	18.0	23.9	51.1	44.9	1.57	1.56	645.08	674.51	4.36
Case 3	22.8	28.8	54.8	48.4	1.50	1.47	626.76	656.10	4.47
Case 4	21.5	27.8	53.9	47.5	1.48	1.50	649.32	669.49	3.01
Case 5	22.6	27.8	56.3	51.0	1.73	1.76	624.48	650.52	4.00
Case 6	19.8	25.9	55.9	48.6	1.47	1.25	624.46	636.36	1.87
Case 7	21.8	26.8	51.7	45.1	1.98	1.55	689.44	713.42	3.36
Case 8	20.1	26.9	52.2	45.8	1.36	1.51	644.03	673.95	4.44
Case 9	22.8	29.1	55.5	49.2	1.43	1.49	627.39	654.63	4.16
Case 10	23.7	29.8	55.2	49.5	1.48	1.63	628.91	647.94	2.94
Case 11	19.9	26.3	54.3	47.7	1.44	1.46	641.80	671.99	4.49
Case 12	22.9	28.1	56.1	49.8	1.77	1.53	640.97	672.21	4.65
Case 13	19.4	25.4	53.8	47.2	1.53	1.46	639.30	672.00	4.87
Case 14	17.8	23.6	52.2	45.8	1.55	1.47	626.06	656.10	4.58
Case 15	20.7	26.7	52.5	46.2	1.53	1.52	639.30	667.81	4.27
Case 16	22.5	28.3	53.6	47.0	1.60	1.47	646.26	676.60	4.48
Case 17	19.3	25.2	53.8	47.5	1.52	1.47	624.53	645.85	3.30
Case 18	18.6	24.3	53.8	47.3	1.64	1.49	650.99	675.42	3.62
Case 19	18.8	25.8	52.3	45.5	1.30	1.38	633.72	654.42	3.16
Case 20	21.9	27.9	53.2	46.8	1.57	1.49	656.01	665.02	1.35

∂, means the infinitesimal change in the numerical calculation). The major sources of the uncertainty of the thermal contact conductance are the measurement errors of the temperature and the mass flow rate. Therefore, the instruments of high accuracy have been used in the current experiment. Consequently the heat transfer rate from the hot to cold water has been estimated to be accurate within ±2%. Also, the thermal contact conductance has contained an uncertainty of about ±20%.

3. Discussion

The analyses of data have been conducted with the confidence level of 95% according to statistical method. As depicted in Fig. 5, it can be said that the thermal contact conductance increases as the tube expansion ratio (E) increases. This behavior can be reasoned by the fact that the contact pressure between the tubes and the fins increases as the expansion ratio increases. Here, the tube expansion ratio has been defined as $E = (D_{ball}/D_{min} - 1) \times 100$, and they are 4.15, 5.65, and 7.14, respectively.

As shown in Fig. 6, the thermal contact conductance decreases as the fin spacing increases. This is attributable to the fact that the applied pressure during the process of tube expansion increases as the fin spacing decreases because fins with smaller fin spacing are more likely to resist against the tube expansion caused by the movement of the ball in a tube. Here, the decrease of fin spacing is compatible with the increase of the number of fins. Therefore, it can be said that the thermal contact conductance increases as the number of fins increases.

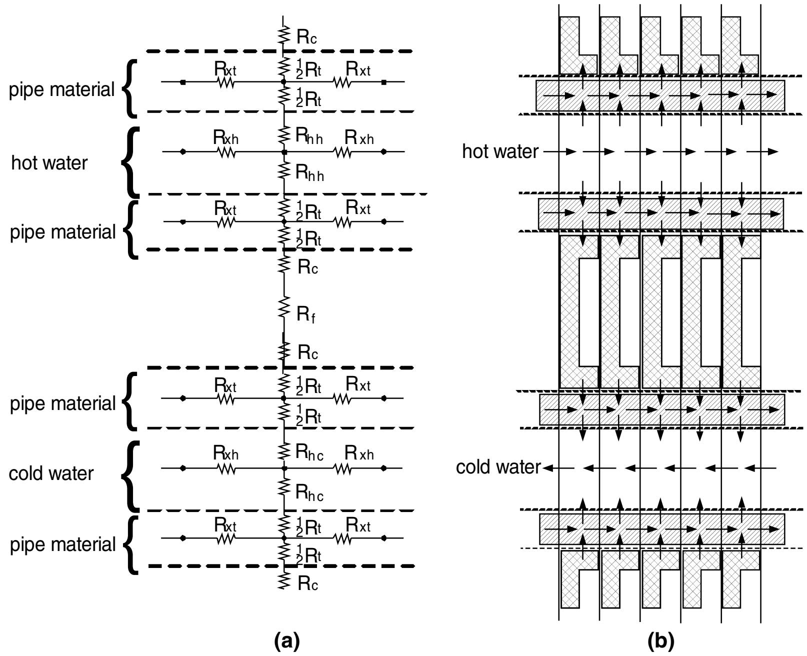

Fig. 4. Thermal resistance circuit and control volumes in a part of heat exchanger (dots mean the nodes for temperature calculation).

Table 3
Results of numerical calculation for thermal contact resistances

	Re_c	Re_h	h_c [W/m^2°C]	h_h [W/m^2°C]	U [W/m^2°C]	c [W/m^2°C]	Ave. of c [W/m^2°C]	Dev. of c [%]	R_c [%]
Case 1	5531	7012	2782.76	3374.83	293.31	8420	8244.0	2.13	13.10
Case 2	5427	7492	2724.38	3634.76	293.96	8068			13.70
Case 3	5185	7060	2588.71	3400.73	309.31	8532	8499.0	0.39	13.63
Case 4	5116	7204	2550.08	3478.55	308.35	8466			13.73
Case 5	5980	8453	3037.19	4160.55	278.87	6534	8161.0	19.94	16.05
Case 6	5082	6003	2530.79	2836.06	266.61	9788			10.24
Case 7	6744	7335	3423.62	3497.10	347.74	14114	13951.5	1.17	9.27
Case 8	4632	7145	2247.94	3396.18	312.06	13789			8.52
Case 9	4871	7051	2377.92	3345.84	286.60	10055	10168.5	1.12	10.73
Case 10	5041	7713	2471.23	3699.87	294.82	10282			10.79
Case 11	4905	6909	2396.54	3270.48	271.14	6648	6117.5	8.67	15.35
Case 12	6029	7240	3019.61	3446.60	278.38	5587			18.75
Case 13	5211	6909	2564.93	3270.48	272.91	10067	8853.0	13.71	10.20
Case 14	5279	6956	2602.51	3295.58	265.45	7639			13.08
Case 15	5211	7193	2564.93	3421.38	313.91	10216	11918.0	14.28	11.56
Case 16	5450	6956	2696.71	3295.58	324.91	13620			8.98
Case 17	5177	6956	2546.16	3295.58	262.80	7326	7640.0	7.90	13.50
Case 18	5586	7051	2772.33	3345.84	270.90	7954			12.82
Case 19	4419	6517	2127.67	3056.81	286.65	17533	17150.0	2.23	6.15
Case 20	5336	7036	2628.43	3330.99	306.24	16767			6.87

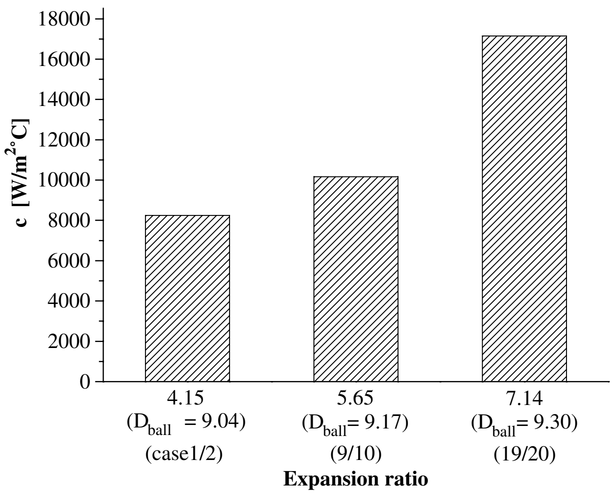

Fig. 5. Thermal contact conductance as to expansion ratio in fin–tube heat exchangers with 9.52 mm tube.

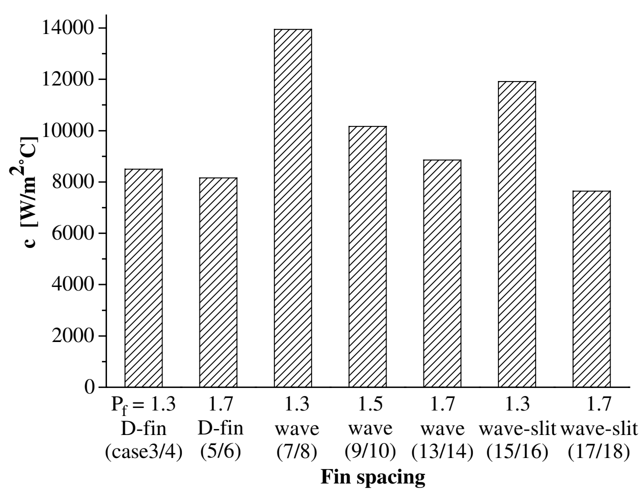

Fig. 6. Thermal contact conductance as to fin spacing in fin–tube heat exchangers with 9.52 mm tube.

The influence of various fin types is depicted in Fig. 7. The thermal contact conductance of wave fin is larger than that of wave-slit fin both in the cases of fin spacings of 1.3 and 1.7 mm because the wave fin is more likely to resist against the expansion of tubes ensuring higher contact pressure between the tubes and the fins. It is reasoned by the fact that the effect of slit is not considered in the calculating process of equivalent length of the wave-slit fin. However, if the other parameters are the same in the cases of wave fin and wave-slit fin, it is clear that thermal resistance of the wave-slit fin is larger than that of the wave fin because the effective heat conduction area through the wave-slit fin is smaller. In addition, the thermal contact conductances are calculated in the cases of the D-fin, but these cannot be compared to the cases of wave fin and wave-slit fin because other parameters are different.

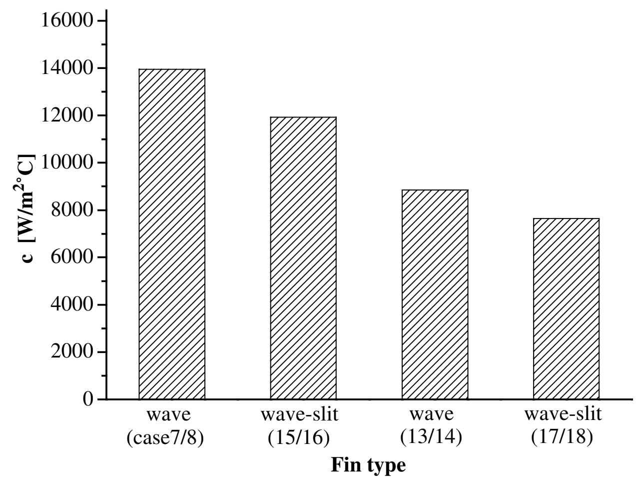

Fig. 7. Thermal contact conductance as to fin type in fin–tube heat exchangers with 9.52 mm tube.

The hydrophilic coating fin has been generally used for the increase of convective heat transfer between the fin surface and the air flow in fin–tube heat exchangers. But, as shown in Fig. 8, the thermal contact conductance of the case with the hydrophilic coating is much small, compared with that of the general fin because the hydrophilic coating interferes with heat transfer at the contact interface. Therefore, the hydrophilic coating must be considered with the thermal contact resistance in the fin–tube heat exchanger.

To evaluate the correlation between several factors and the thermal contact conductance in fin–tube heat exchangers with 9.52 mm tubes, a new parameter of interfacial pressure, p, similar to that used in Eckels (1977) correlation is proposed in Eq. (9). The parameter is different from Eckels (1977) parameter in considering the effects of expansion rate, hydrophilic coating and fin type.

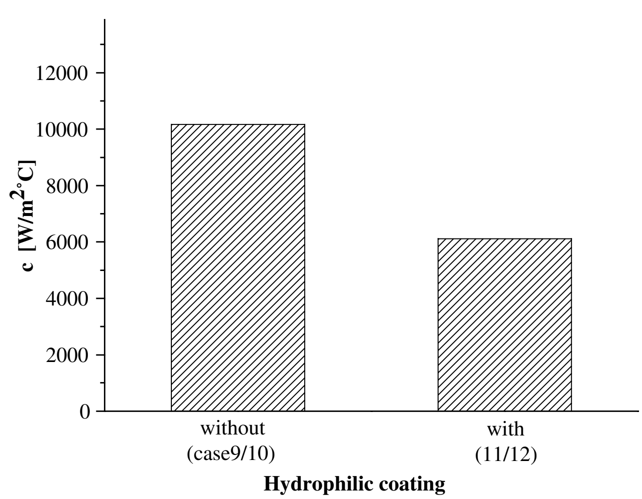

Fig. 8. Thermal contact conductance as to fin coating in fin–tube heat exchangers with 9.52 mm tube.

$$p \propto \frac{t_f}{\left(\frac{1}{n_f t_f} - 1\right)^2 D_o} \cdot E \cdot C_f \cdot S_f \tag{9}$$

Here, C_f and S_f are the constants related to the hydrophilic fin coating and the fin type, respectively, and the specific values are as follows:

$$C_f = \begin{cases} 0.6 & \text{: with hydrophilic coating} \\ 1 & \text{: without hydrophilic coating} \end{cases}$$
$$S_f = \begin{cases} 0.86 & \text{: with slit} \\ 1 & \text{: without slit} \end{cases} \tag{10}$$

As depicted in Fig. 9, the correlation between the thermal contact conductance and the parameter is adequate for the least squares linear fitting by regression analysis. The correlation can be represented in the following equation, and the coefficient of determination (r^2) is 0.92.

$$c(\mathrm{W/m^2\,°C}) = 2790 \cdot \left(\frac{t_f}{\left(\frac{1}{n_f t_f} - 1\right)^2 D_o} \cdot E \cdot C_f \cdot S_f \times 10^4 \right) - 266 \tag{11}$$

In addition, the portions of each thermal resistance to the total thermal resistance are calculated in all cases. Here, the total thermal resistance means only the sum of the radial resistances between hot and cold water, which is R_{total} in Eq. (2). As depicted in Fig. 10, the portions of the thermal conduction resistance of the fin are the largest while the ratios of the thermal conduction resistance of the tube thickness are so small in all cases. The portions of the thermal contact resistance range approximately from 6% to 18%. Like this, the results of the thermal contact resistance in this study are in the same line with other authors' (Eckels, 1977; Sheffield et al., 1989) results although the experimental conditions are different.

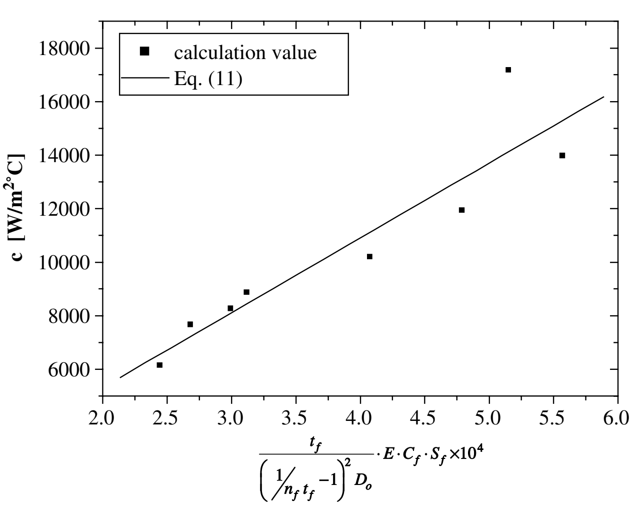

Fig. 9. Effectiveness of the proposed correlation compared with the dispersed data.

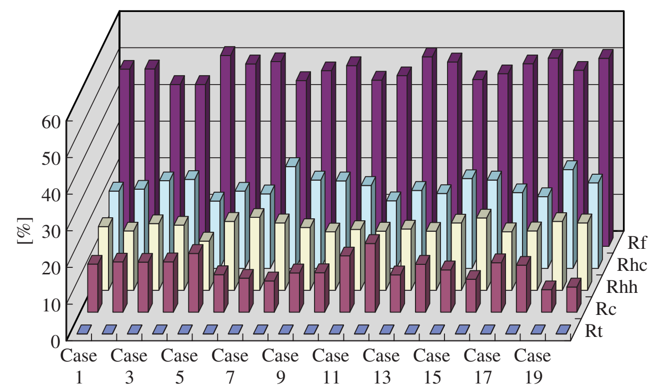

Fig. 10. Composition of thermal resistances in fin–tube heat exchangers with 9.52 mm tube.

4. Conclusion

In this study, the thermal contact conductances have been evaluated using the experimental–numerical method for various fin–tube heat exchangers with 9.52 mm tube and a new correlation between thermal contact conductance and the effective factors such as tube expansion ratios, fin spacings, fin types, and hydrophilic fin coating has been presented. Also, the portions of the thermal resistances have been studied as well.

The results of the present study imply that the thermal contact resistance is closely related to the several factors. That is, the thermal contact conductance increases as the expansion ratio and the number of fin increase. And, the thermal contact conductances in the wave fin and the non-hydrophilic coating fin is larger than that in the wave-slit fin and the hydrophilic coating fin. These prove that hydrophilic fin coating and the contact pressure between the fin and the tube is an essential parameter in the thermal contact resistance. Meanwhile, the portions of the thermal contact resistance to the total thermal resistances range approximately from 6% to 18%, and the thermal contact resistance may not be ignored in the analysis of the fin–tube heat exchanger. Also, A new correlation for the thermal contact conductance can be utilized as a basic data in the design process of the fin–tube heat exchangers. In the future study, further experiments can be carried out with different tube diameters for the optimal fin–tube heat exchanger design.

References

Abuebid, M.A., 1984. A thermal contact conductance correlation for mechanically expanded finned tube heat exchangers. M.S. thesis. University of Missouri-Rolla.

Dart, D.M., 1959. Effect of fin bond on heat transfer. ASHRAE Journal 1 (5), 67–71.

Eckels, P.W., 1977. Contact conductance of mechanically expanded plate finned tube heat exchangers. ASME Paper No. 77-HT-24.

Eckels, P.W., Rabas, T.J., 1987. On the correlation of wet and dry transport processes in plate finned-tube heat exchangers. Journal of Heat Transfer 109, 575–582.

Kim, C.N. et al., 2003. An experimental–numerical evaluation of thermal contact conductance in fin–tube heat exchangers. JSME International Journal, Series B 46 (2), 299–307.

Park, H.B. et al., 1997. Experimental study of heat transfer and pressure drop characteristics for flow of water inside circular smooth and micro-fin tubes. Korean Journal of Air-Conditioning and Refrigeration Engineering 9 (4), 454–461.

Salgon, J.J. et al., 1997. A mechanical and geometrical approach to thermal contact resistance. International Journal of Heat Mass Transfer 40 (5), 1121–1129.

Shah, P.R., 1986. Microscopic and macroscopic fin collar effect in the prediction of finned tube contact conductance. Ph.D. thesis. University of Missouri-Rolla.

Sheffield, J.W., Sauer Jr., H.J., Wood, R.A., 1987. An experimental method for measuring the thermal contact resistance of plate finned tube heat exchangers. ASHRAE Transactions 93 (2), 776–785.

Sheffield, J.W., Wood, R.A., Sauer Jr., H.J., 1989. Experimental investigation of thermal conductance of finned tube contacts. Experimental, Thermal and Fluid Science, 107–121.

Stubblefield, M.A., Pang, S.S., Coundy, V.A., 1996. Heat loss in insulated pipe the influence of thermal contact resistance: a case study. Journal of Composites, Part B 27B (1), 85–93.