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Technical Note

A study on the thermal contact conductance in fin–tube heat exchangers with 7 mm tube

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Abstract

The thermal contact resistance has been frequently neglected in the process of design of heat exchangers because of the difficulty of measurement and the lack of accurate data. However, the thermal contact resistance is one of principal parameters in heat transfer mechanism of fin–tube heat exchangers. The objective of the present study is to investigate new factors such as fin types and manufacturing types of the tube affecting the thermal contact conductance and to find a correlation between the thermal contact conductance and the effective factors in fin–tube heat exchangers with 7 mm tube. The thermal contact conductances in the 22 heat exchangers with 7 mm tube have been investigated through the experimental–numerical method. A numerical scheme has been employed to calculate the thermal contact conductance and the portion of thermal resistances using the experimental data. As a result, the thermal contact conductance has been evaluated quantitatively, and a new correlation including the influence of new factors such as fin types and manufacturing types of the tube has been developed in the fin–tube heat exchanger with 7 mm tube. Also, the portion of each thermal resistance has been evaluated in each case.
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Keywords: Fin–tube heat exchanger; Thermal contact resistance; Thermal contact conductance

1. Introduction

The heat exchanger is a device used to transfer heat from a fluid on one side of a barrier to another fluid on the other side without bringing the fluids into direct contact. According to their peculiarities the heat exchanger can be classified into various types such as shell and tube heat exchanger, plate heat exchanger, fin–tube heat exchanger and etc. The fin–tube heat exchanger has been applied in too many parts such as power stations, chemical plants, refrigerating industries, aircrafts, automobiles and etc. For several decades, many researchers have studied to improve the efficiency of the fin–tube heat exchanger and have developed many heat exchangers. Generally, the fin–tube heat exchanger is manufactured through mechanical expansion of tube to tighten the contact between fins and tubes. The features of heat transfer through interfaces have not been clarified because of the irregular contact of interface. Therefore, the thermal contact resistance has not been investigated deeply and occasionally has been overlooked because of lack of accurate data, difficulties in measurements and complexities of heat transfer through interfaces.

A study on thermal contact resistance in a fin–tube heat exchanger was first attempted by Dart [1]. He installed heat exchangers in an adiabatic chamber to minimize the influence of natural convection. The thermal contact resistance was evaluated in the fin–tube heat exchangers with two passages, which were one for cold and the other for hot water, and was compared to that in soldered fins. Eckels [2] examined the thermal contact conductance varying the

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Nomenclature

A_c	contact area between the fin collar and tube surface [m^2]
A_f	cross-sectional area of the fin [m^2]
A_i	inside area of the tube [m^2]
A_m	log mean area of the tube [m^2]
c	thermal contact conductance [W/m^2 °C]
D	nominal tube diameter [m]
D_{ball}	diameter of expansion ball [m]
D_i	inner diameter of the tube [m]
D_{min}	minimum diameter of the tube [m]
D_o	outer diameter of the tube [m]
E	expansion ratio [%]
h_c	convective heat transfer coefficient of the cold water [W/m^2 °C]
h_h	convective heat transfer coefficient of the hot water [W/m^2 °C]
k_f	thermal conductivity of the fin [W/m °C]
k_t	thermal conductivity of the tube [W/m °C]
l_{eq}	equivalent length of the fin [m]
N_f	fin number [EA]
Nu	Nusselt number
P_f	fin pitch (spacing) [m]
Pr	Prandtl number
R_c	thermal contact resistance [%] or [°C/W]
Re	Reynolds number
R_f	thermal conduction resistance of the fin [%] or [°C/W]
R_{hc}	thermal convection resistance of cold water [%] or [°C/W]
R_{hh}	thermal convection resistance of hot water [%] or [°C/W]
R_t	thermal conduction resistance of the tube [%] or [°C/W]
t_f	thickness of the fin [m]
t_t	thickness of the tube [m]
T_c	temperature of the cold water [°C]
T_h	temperature of the hot water [°C]
ΔT_{ho}	temperature drop of hot water [°C]
U	overall heat transfer coefficient [W/m^2 °C]
w	width of the fin [m]

number of fins, the thickness of fin and the diameter of tube in the wet and dry fin–tube heat exchangers. He tested with empirical method based on Dart's method and derived a functional relationship for the thermal contact conductance in plate-finned tube heat exchangers. Abuebid [3] investigated the thermal contact resistance with plate-finned tube heat exchangers placed in a vacuum. He performed an error analysis similar to the Eckels' method, but the error band was narrower. Shah [4] studied the relation between pressure distribution on the collar and the temperature distribution in the fin and collar. He calculated possible errors in neglecting the influence of the thermal contact conductance and constriction in the fin root. Sheffield et al. [5] considered the contact pressure as the significant factor of the thermal contact resistance. They examined the influence of surface hardness and studied the correlation between contact pressure and expansion interference such as surface hardness and roughness of tube. Stubblefield et al. [6] studied the influence of thermal contact resistance by evaluating heat loss in insulated pipe and presented a simple model to predict the effect of contact resistance. Salgon et. al. [7] theoretically predicted the thermal contact resistance which was computed as a function of contact pressure and was compared to experimental data. But, these previous studies have been insufficient to ascertain factors affecting the thermal contact conductance and have included many errors.

In this study, we have investigated the new factors such as fin types (plate fin, slit fin and wide slit fin) and manufacturing types of the tube (drawn tube and welded tube) affecting the thermal contact conductance and have presented a new correlation between the effective factors and the thermal contact conductance in fin–tube heat exchangers with 7 mm tube. Therefore, the thermal contact conductance has been evaluated quantitatively on various fin–tube heat exchangers with 7 mm tube using experimental-numerical method. Also, each portion of thermal resistances to the total thermal resistance has been calculated in all samples.

2. Theory

2.1. Step 1—experimentation

The experiment is conducted with fin–tube heat exchangers with various tube expansion ratios, fin spacings, fin types (plate fin, slit fin and wide slit-fin) and manufacturing types of the tube (drawn tube and welded tube). The configuration and specifications of the fin–tube heat exchanger with 7 mm tube is presented in Table 1. To improve the reliability of experiment, a pair of heat exchangers of similar specifications have been manufactured and tested respectively, and eleven pairs (total 22 samples) have been tested repeatedly.

As shown in Fig. 1, the experimental apparatus consists of vacuum chamber, vacuum pump, a pair of constant temperature reservoirs, a pair of water pumps, a pair of mass flow meters, pressure gauge, thermo sensors, and etc. The fin–tube heat exchanger is composed of aluminum fins and grooved copper tubes with diameter of 7 mm, and has 12 tubes in a row as depicted in Fig. 2. 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 to minimize the heat transfer from hot to cold

Table 1
Configuration of fin–tube heat exchangers with 7 mm tube

	D (mm)	D_{ball} (mm)	P_f (mm)	Fin type	w (mm)	Manufacturing type of tube	Hydrophilic coating	N_f (EA)	D_o (mm)	D_i (mm)	t_f (mm)
Case 1	7	6.48	1.3	Slit fin	12.7	Drawn	With	308	7.32	6.78	0.11
Case 2	7	6.48	1.3	Slit fin	12.7	Drawn	With	308	7.32	6.78	0.11
Case 3	7	6.48	1.3	Slit fin	12.7	Welded	With	308	7.32	6.78	0.11
Case 4	7	6.48	1.3	Slit fin	12.7	Welded	With	308	7.32	6.78	0.11
Case 5	7	6.53	1.3	Wide slit fin	18.2	Drawn	With	303	7.37	6.83	0.11
Case 6	7	6.53	1.3	Wide slit fin	18.2	Drawn	With	307	7.37	6.83	0.11
Case 7	7	6.53	1.5	Wide slit fin	18.2	Drawn	With	267	7.37	6.83	0.11
Case 8	7	6.53	1.5	Wide slit fin	18.2	Drawn	With	267	7.37	6.83	0.11
Case 9	7	6.53	1.3	Slit fin	12.7	Drawn	With	309	7.37	6.83	0.11
Case 10	7	6.53	1.3	Slit fin	12.7	Drawn	With	307	7.37	6.83	0.11
Case 11	7	6.53	1.3	Plate fin	12.7	Drawn	With	304	7.37	6.83	0.11
Case 12	7	6.53	1.3	Plate fin	12.7	Drawn	With	305	7.37	6.83	0.11
Case 13	7	6.53	1.5	Slit fin	12.7	Drawn	With	267	7.37	6.83	0.11
Case 14	7	6.53	1.5	Slit fin	12.7	Drawn	With	265	7.37	6.83	0.11
Case 15	7	6.53	1.5	Slit fin	12.7	Drawn	Without	271	7.37	6.83	0.11
Case 16	7	6.53	1.5	Slit fin	12.7	Drawn	Without	271	7.37	6.83	0.11
Case 17	7	6.53	1.3	Slit fin	12.7	Welded	With	308	7.37	6.83	0.11
Case 18	7	6.53	1.3	Slit fin	12.7	Welded	With	303	7.37	6.83	0.11
Case 19	7	6.58	1.3	Slit fin	12.7	Drawn	With	308	7.44	6.9	0.11
Case 20	7	6.58	1.3	Slit fin	12.7	Drawn	With	310	7.44	6.9	0.11
Case 21	7	6.58	1.3	Slit fin	12.7	Welded	With	308	7.44	6.9	0.11
Case 22	7	6.58	1.3	Slit fin	12.7	Welded	With	308	7.44	6.9	0.11

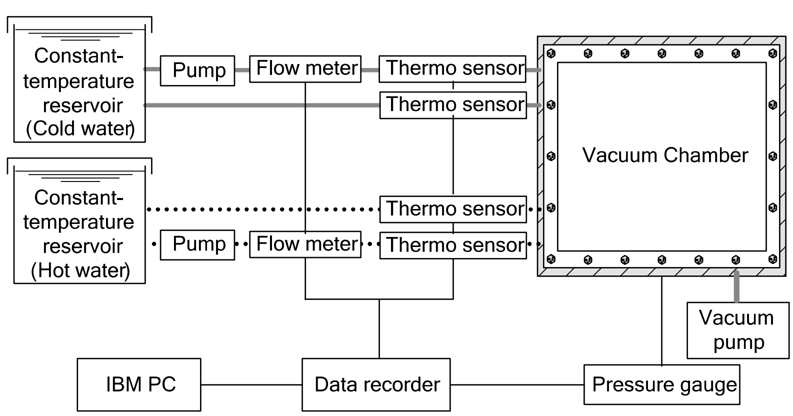

Fig. 1. Schematic diagram of experimental apparatus.

water through the chamber wall. Also, the fin–tube heat exchanger is placed in an insulated vacuum chamber ensuring that the aluminum fins function only as a conduction media with a minimum of natural convection on their surface.

The inlet temperatures of hot and cold water are respectively about 75 and 21 °C, and the flow rates of hot and cold water are about 1.8 [kg/min]. The measurement errors of the temperatures and the flow rates are about ± 0.05 °C and $\pm 0.1\%$, respectively. The differences of heat transfer rates between the hot and the cold water have been estimated to be about below 1%. Therefore, the heat loss by the natural convection and radiative heat transfer inside the chamber and through the wall of chamber can be neglected in this study. The experiment was conducted only for calculating the energy balance and obtaining the input data for the numerical calculation. The more detailed explanation about the procedure of experiment has been suggested in our previous paper [8].

2.2. Step 2—numerical calculation

It is too difficult to calculate the thermal contact conductance analytically because the fin–tube heat exchangers have 12 tubes in a row as shown in Fig. 2. Therefore, a numerical calculation is introduced in this study. Assuming the fin–tube heat exchanger has only two tubes (one for the hot and the other for the cold water) in a row, thermal

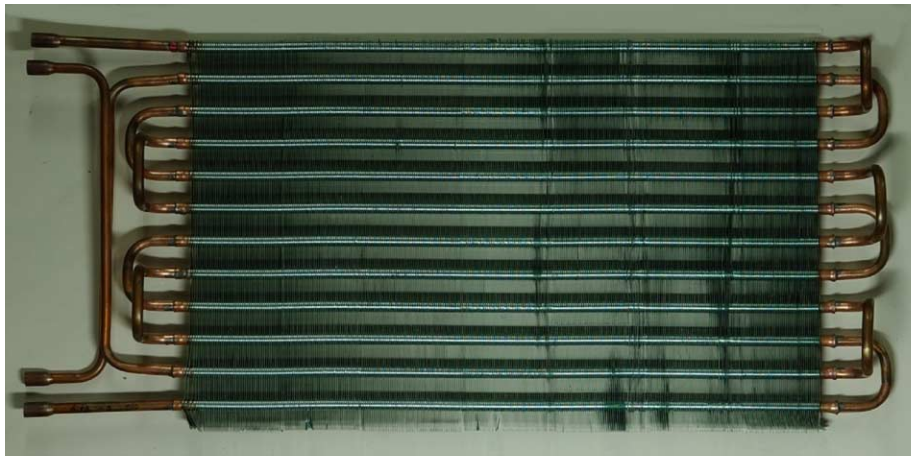

Fig. 2. A fin–tube heat exchanger with 7 mm tube used in the experiment.

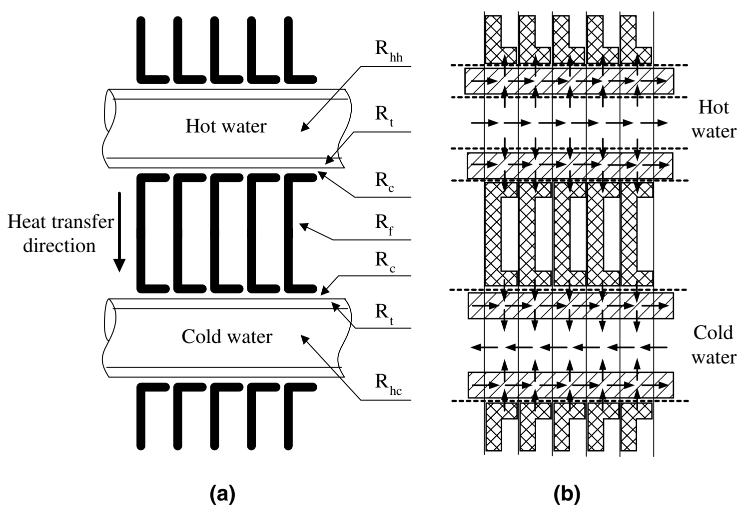

Fig. 3. Thermal resistances and control volumes in a part of fin–tube heat exchanger.

resistances can be expressed as shown in Fig. 3(a), and heat transfer rate ($d\dot{Q}$) through only an infinitesimal area (dA_i) between the hot and cold water is as follows.

$$d\dot{Q} = U(T_h - T_c)\,dA_i \tag{1}$$

$$\frac{1}{U\,dA_i} = R_{total} = R_{hh} + 2R_t + 2R_c + R_f + R_{hc} \tag{2}$$

where

$$R_{hh} = 2/(h_h\,dA_i) \tag{3}$$

$$R_t = 2t_t/(k_t\,dA_m) \tag{4}$$

$$R_c = 2/(c\,dA_c) \tag{5}$$

$$R_f = l_{eq}/(k_f\,dA_f) \tag{6}$$

$$R_{hc} = 2/(h_c\,dA_i) \tag{7}$$

Also, axial conduction through the tubes materials and axial energy flow inside the tubes are considered additively. On the other hand, heat conduction in the hot and cold water is neglected because it is too small compared with the advection. For numerical calculation, the heat exchanger is divided into small elements as shown in Fig. 3(b). The numerical calculation is conducted using the global energy balance scheme that the sum of net energy inflow in every node in 12 tubes should be zero in a steady state. The global energy balance can be achieved by the iterative calculation of heat flow in each local element of a fin–tube heat exchanger.

The correlation for convective heat transfer of water flow in a 7 mm grooved tube was presented by Park et al. [9] as

$$Nu = 0.00211 \times Re^{1.11} \times Pr^{0.3} \quad (3000 < Re < 16{,}000) \tag{8}$$

Therefore, the convective heat transfer coefficients (h_h and h_c) of the hot and cold water can be determined from $h_h = Nu(k_h/D_i)$, $h_c = Nu(k_c/D_i)$ and Eq. (8). In this study, the Reynolds number of hot water and cold water are about 12,000–14000 and 6200–7200, respectively.

To simplify the calculation of the thermal resistance of fin (R_f) as presented in Eq. (6), we introduced an effective conduction length (equivalent length) instead of a real

Table 2
Equivalent length of different fins

D (mm)	w (mm)	l (mm)	l_{eq} (mm)
7 (Plate fin)	12.7	21.00	15.60
7 (Slit fin)	12.7	21.00	15.60
7 (Wide slit fin)	18.2	21.00	18.11

conduction length through a fin. The detailed explanation about the equivalent length (l_{eq}) can be referred to our previous research paper [8]. The length between the centers of the tubes (l) is 21.0 mm in the all fins, but the calculated equivalent length (l_{eq}) is 15.6 mm for the plate and slit fins, and is 18.11 mm for the wide slit fin as presented in Table 2.

In this study, the numerical calculation is used to evaluate quantitatively a thermal contact conductance which allows both the measured and the numerical heat balances to be the same. Therefore, the numerical calculation is repeatedly carried out changing the presumed value of thermal contact conductance until the computed outlet temperatures of the hot and cold water is equal to those of the experiment within a certain error. The detailed explanation about numerical calculation has been suggested in the previous paper of ours [8].

As the results of numerical calculation, the thermal contact conductances range about from 6000 to 11,000 [W/m^2 °C] depending on the features of fin–tube heat exchangers as shown in Table 3. In any pair of heat exchangers of similar specifications, where the difference lies only on the number of fins and the other features are the same, the difference of the thermal contact conductances is below 20%. Therefore, the representative thermal contact conductance has been evaluated in this study with the arithmetic mean value of the thermal contact conductances of the two heat exchangers. Here, the result of Case 4 has been excluded from the evaluation of thermal contact conductance because the quality of the heat exchanger is too poor to be tested as compared with the other heat exchangers.

Table 3
Results of numerical calculation for thermal contact conductance

	c (W/m^2 °C)	Ave. of c (W/m^2 °C)	Dev. of c (%)	R_c (%)
Case 1	5625	6148.5	15.69	25.47
Case 2	6672			22.00
Case 3	7026	7026	50.73	21.44
Case 4	3462 (excluded)			35.93
Case 5	8572	9063.3	10.28	20.56
Case 6	9554			18.88
Case 7	8374	7848.5	12.54	19.48
Case 8	7324			21.65
Case 9	8006	7280.5	18.11	19.24
Case 10	6556			22.59
Case 11	9557	9297.0	2.80	18.00
Case 12	9037			18.82
Case 13	6530	6351.5	5.47	20.86
Case 14	6173			21.78
Case 15	9675	9492.0	1.93	16.21
Case 16	9309			16.72
Case 17	7411	7700.0	7.23	19.95
Case 18	7989			19.38
Case 19	9704	8894	16.69	16.19
Case 20	8084			18.80
Case 21	10,532	10,413	2.26	15.10
Case 22	10,294			15.44

In this study, the major source of the uncertainty of the thermal contact conductance is the measurement error of temperature of the hot and the cold water because the sensitivity of the thermal contact conductance to the temperature drop in the numerical calculation, $\frac{\partial c}{\partial(\Delta T_{ho})}$, is about 2.402×10^4 [W/m^2 °C^2]. Here, the symbol, ∂, means the infinitesimal change in the numerical calculation. Under the condition that the mass flow rate $\dot{m}_h$ is exact, this can be transformed into the ratio of the infinitesimal change in the thermal contact conductance to that in the heat loss from the hot water $\left(\frac{\partial c}{\partial \dot{Q}_h}\right)$.

Now, the heat transfer rate from the hot to cold water, $\dot{Q}_h$, can be determined by the three experimental data; mass flow rate ($\dot{m}_h$) and temperatures of inlet and outlet (T_{hi}, T_{ho}). It is clear that the inaccuracy of heat transfer rate from the hot to cold water, $\delta\dot{Q}_h$, consists of the error caused by uncertainty of the mass flow rate and that originated from the uncertainty of the temperatures at the inlet and outlet. Therefore, the inaccuracy of heat transfer rate from the hot to cold water, $\delta\dot{Q}_h$, can be written as

$$\delta\dot{Q}_h = \sqrt{\left(\frac{\partial\dot{Q}}{\partial(\dot{m}_h)}\delta\dot{m}_h\right)^2 + \left(\frac{\partial\dot{Q}}{\partial(T_{hi})}\delta T_{hi}\right)^2 + \left(\frac{\partial\dot{Q}}{\partial(T_{ho})}\delta T_{ho}\right)^2} \tag{9}$$

with the inaccuracies of 0.1% in the measurement of hot water mass flow rate and of 0.05 °C in the temperature measurement as stated previously. Then the value of $\delta\dot{Q}_h$ is obtained to be about 1% of heat transfer rate from hot to cold water. Also, the inaccuracy of the thermal contact conductance, δc, can be written in the following equation:

$$\delta c \approx \frac{\partial c}{\partial \dot{Q}_h}\delta\dot{Q}_h \tag{10}$$

The estimated inaccuracy (δc) turns out to be about 20% of the values of thermal contact conductance. In the current experiment the instruments of high accuracy have been used.

3. Discussion

The experiment for heat balance has been carried out twice for each of the two heat exchangers of similar specifications. Also, to validate the results the additional experiments have been conducted several times repeatedly in Case 5 and Case 6. The analyses of data have been performed with the confidence level of 95% for the statistical method. As shown in Fig. 4, the reproducibility of the thermal contact conductance has been comparatively good.

To validate the experimental–numerical method, the influence as to the change of flow rate has been investigated

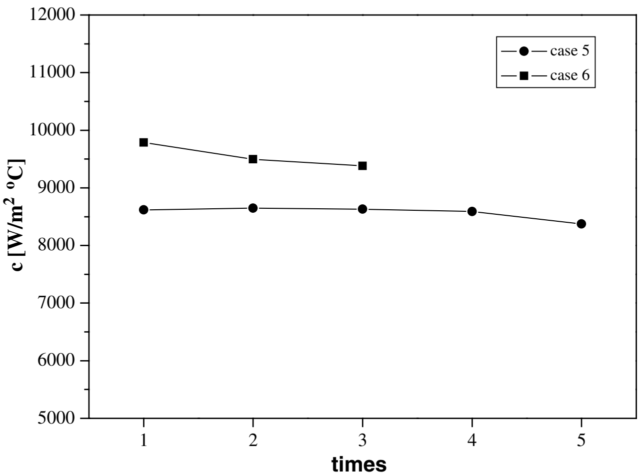

Fig. 4. Reproducibility of thermal contact conductance.

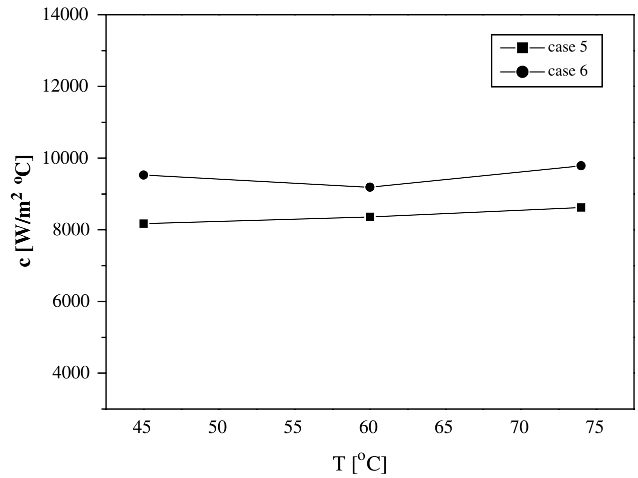

Fig. 6. Thermal contact conductance with inlet temperature.

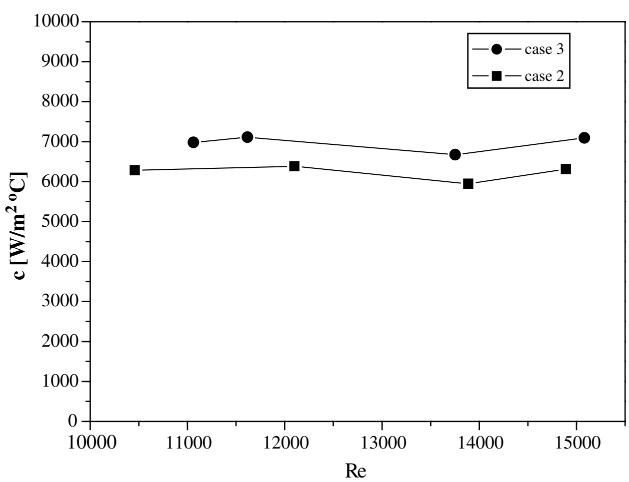

Fig. 5. Thermal contact conductance with flow rate.

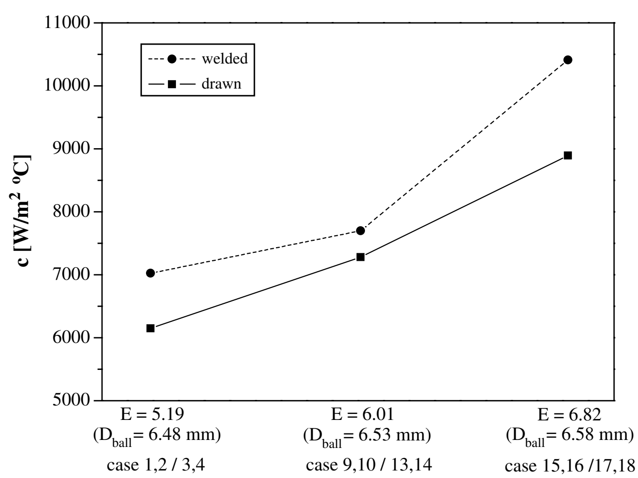

Fig. 7. Effect of tube expansion ratio on thermal contact conductance for different manufacturing types of the tube.

in two cases as depicted in Fig. 5. Here, the thermal contact conductance almost does not change as the Reynolds number (i.e. flow rate) increases, which implies that this experimental–numerical method is adequate for this study. Also, to investigate the effect of temperature the experiment has been conducted with different inlet temperature of hot water in two cases as shown in Fig. 6. The thermal contact conductance does not change much with the change of the inlet temperature of the hot water. But these behaviors do not agree with general prediction that the thermal contact conductance decreases a little because the thermal expansion of aluminum fin is larger than that of copper tube. It is believed that the effect of hydrophilic coating, interfering with heat transfer between the fin and the tube, is much larger so that the effect of thermal expansion of the fin and the tube is not notable here. The effect of the hydrophilic coating of the fin is explained in the following paragraph.

As shown in Fig. 7, the thermal contact conductance increases as the expansion ratio (E) of tube increases both in drawn tube and in welded tube. This behavior can be reasoned by the fact that the contact pressure between the tube and the fin increases as the expansion ratio of tube increases. Here, the tube expansion ratio has been defined as $E = (D_{\text{ball}}/D_{\min} - 1) \times 100$, and they are 5.19, 6.01 and 6.82, respectively.

The tube used in fin–tube heat exchangers can be made by two ways. One way is of drawing process and the other is of welding process. These manufacturing processes of tube can affect the thermal contact conductance. As shown in Fig. 7, the thermal contact conductance in the welded tube is larger than that in drawn tube in each expansion ratio. The reason is that the welded tube undertakes a process of surface treatment after welding of tubes in the manufacturing process, while the drawn tube has no process of surface treatment. Although the surface roughness of the tube is not investigated in this study, this surface treatment may increase the thermal contact conductance of welded tubes compared with that of drawn tubes.

Also, effect of fin spacing has been studied as shown in Fig. 8. The thermal contact conductance in the case of fin

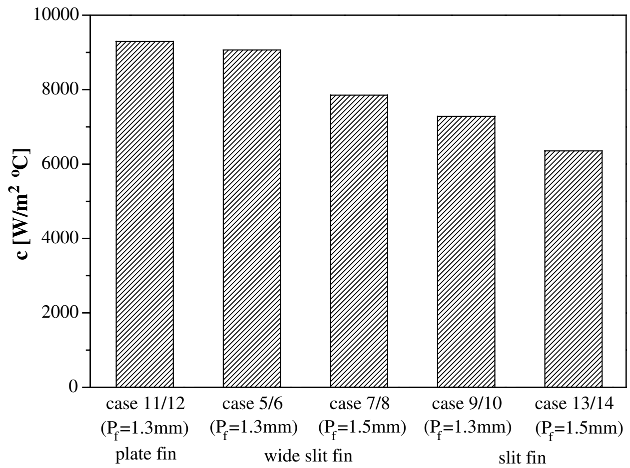

Fig. 8. Effect of fin spacing and fin type on thermal contact conductance.

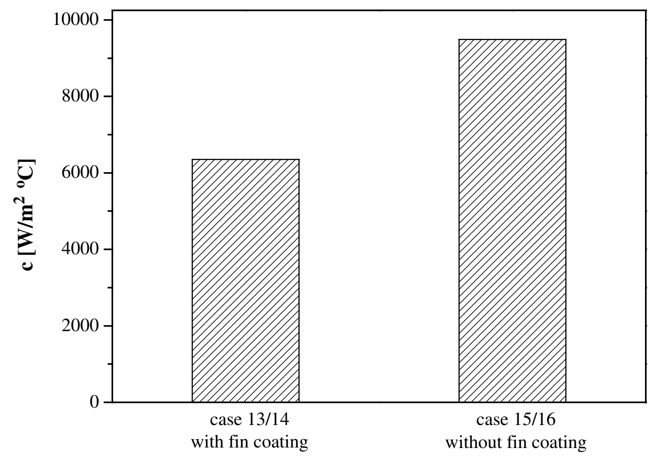

Fig. 9. Effect of hydrophilic fin coating on thermal contact conductance.

spacing of 1.3 mm is larger than that of 1.5 mm both in slit fin and in wide slit fin. This is attributable to the fact that the applied pressure during the process of tube expansion increases as the fin spacing decreases because the 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. This result is in the same line with our previous research result in the fin–tube heat exchanger with 9.52 mm tube [10].

To investigate the influence of fin type, the thermal contact conductance has been evaluated in the cases of plate fin, slit fin and wide slit fin as depicted in Fig. 8. The thermal contact conductance in the case of the wide slit fin is larger than that of the normal slit fin both in the cases of fin spacing of 1.3 and 1.5 mm. Also, the thermal contact conductance in the case of the plate fin is largest of all fin types. The reason is that the plate fin and the wide slit fin are more likely to resist against the expansion of tubes ensuring higher contact pressure between the tubes and the fins.

In the fin–tube heat exchanger, the hydrophilic coating fin has been generally used for the increase of convective heat transfer between the fin surface and the air flow. But, as shown in Fig. 9, the thermal contact conductances of the cases without the hydrophilic coating are much larger, compared with those of the cases with hydrophilic coating fin. The reason is that the hydrophilic coating of the fin interferes with heat transfer at the contact interface between the fin and the tube. Therefore, the hydrophilic coating of the fin in the fin–tube heat exchanger must be carefully considered in association with the thermal contact resistance.

To develop a correlation between several new effective factors and the thermal contact conductance in fin–tube heat exchangers with 7 mm tube, a new parameter of interfacial pressure, p, similar to that used in Eckels [2,11] correlation is proposed in Eq. (11). The parameter is different from Eckels' parameter in the sense that it considers the effects of expansion ratio, the fin type and manufacturing type of the tube.

$$p \propto \frac{t_{\mathrm{f}}^3}{(P_{\mathrm{f}} - t_{\mathrm{f}})^2 D_{\mathrm{o}}} \cdot E \cdot S_{\mathrm{f}} \cdot M_{\mathrm{t}} \cdot C_{\mathrm{f}} \tag{11}$$

Here, S_{f} and M_{t} are the constants related to the fin type (plate fin, slit fin and wide slit fin) and manufacturing type of the tube (drawn tube and welded tube), respectively and C_{f} is the constant related to the hydrophilic coating of the fin (with coating and without coating). The specific values are as follows:

$$\begin{aligned} S_{\mathrm{f}} &= \begin{pmatrix} 1 \ : \ \text{for slit fin} \\ 1.25 \ : \ \text{for wide slit fin} \\ 1.28 \ : \ \text{for plate fin} \end{pmatrix} \\ M_{\mathrm{t}} &= \begin{pmatrix} 1 \ : \ \text{for drawn tube} \\ 1.12 \ : \ \text{for welded tube} \end{pmatrix} \\ C_{\mathrm{f}} &= \begin{pmatrix} 1 \ : \ \text{without coating} \\ 1.11 \ : \ \text{with coating} \end{pmatrix} \end{aligned} \tag{12}$$

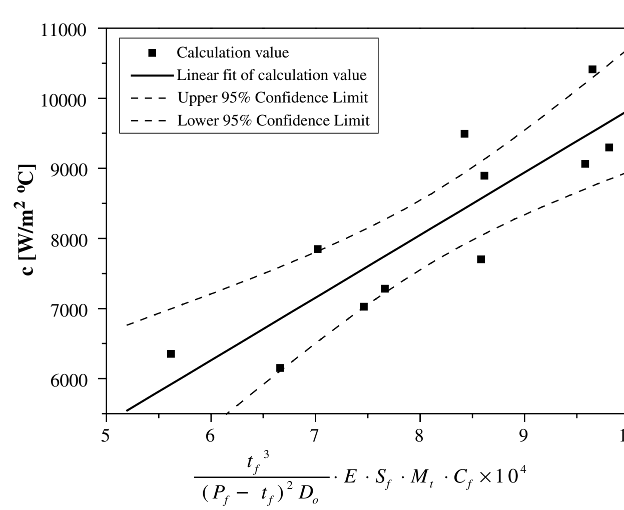

Fig. 10. Comparison of the correlation and dispersed data.

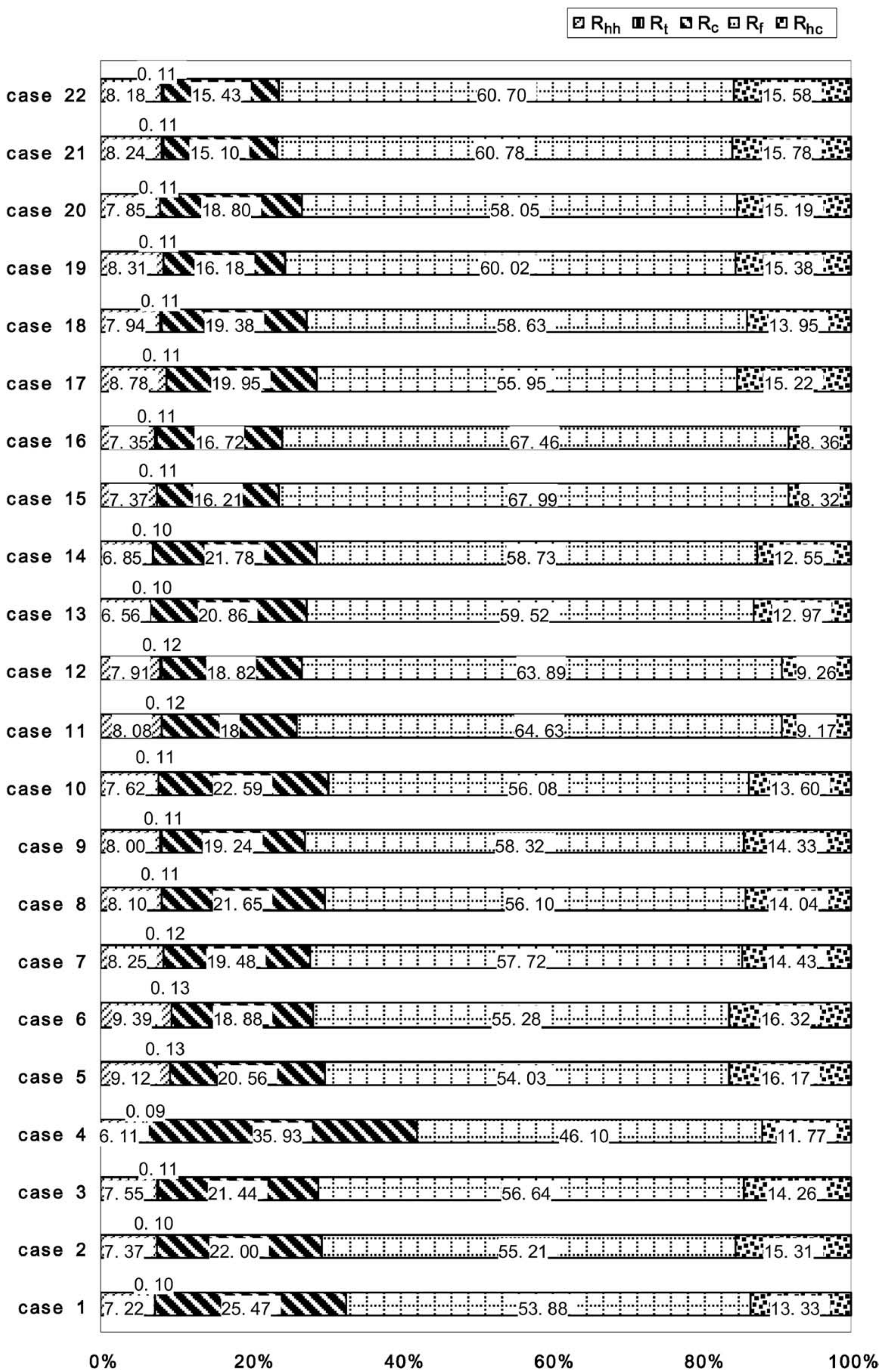

Fig. 11. Composition of thermal resistances in fin–tube heat exchangers.

The least square linear fitting for the regression of the experimental data has been carried out to obtain a correlation for the thermal contact conductance as given in Eq. (13), with the coefficient of determination R^2 to be 0.76. Here, the coefficient of determination is the quantitative index, which shows the degree of the suitability of the least square line to the dispersed data. If R^2 is equal to 1, it means that the least square line can represent the dispersed data perfectly. The comparison of the proposed correlation and the dispersed data is shown in Fig. 10.

$$c\ [\mathrm{W/m^2\,^\circ C}] = 893.7 \cdot \left(\frac{t_f^3}{(P_f - t_f)^2 D_o} \cdot E \cdot S_f \cdot M_t \cdot C_f \times 10^4 \right) + 899.0 \tag{13}$$

The portions of each thermal resistance to the total thermal resistance have been evaluated in all cases. Here, the total thermal resistance means only the sum of resistances R_{total} in the Eq. (2) neglecting axial resistances in the tube and air-side convection resistance for tubes of hot and cold water. As shown in Fig. 11, the portions of thermal conduction resistance of fin are the largest while the portions of the thermal conduction resistance of tube are so small in all cases. The portions of thermal contact resistance are about 15–25%. It can be noted that these large portions of thermal contact resistance are partially attributable to the hydrophilic coating of fins because the hydrophilic coating interferes with the heat transfer between the fins and the tubes as presented in the previous research paper of ours [8].

4. Conclusion

The thermal contact conductances have been investigated using the experimental–numerical method in the fin–tube heat exchangers with 7 mm tubes. Here, the thermal contact conductances have been evaluated quantitatively for the variances of the new factors such as fin type and manufacturing type of the tube. In this study, the new correlation for thermal contact conductance considering the new effective factors has been developed in fin–tube heat exchanger with 7 mm tube. Also, the portions of each thermal resistance to the total thermal resistance have been evaluated in all cases.

Consequently, it has been revealed that the factors such as fin type, manufacturing type of the tube and etc. have a large effect on the thermal contact resistance in fin–tube heat exchanger with 7 mm tube. That is, the thermal contact conductance increases with the increase of the tube expansion ratio and the number of fin, and the thermal contact conductance in the case of wide slit fin is larger than that of normal slit fin and that in the case of plate fin is largest of all fin types. Also, the thermal contact conductance in the case with welded tube is larger than that with drawn tube and that in the case without hydrophilic coating is larger than that with hydrophilic coating. These behaviors closely relates to the contact pressure as mentioned in the previous section. The portions of the thermal contact resistance are about 15–25% in cases of the fin–tube heat exchanger with 7 mm tube, and it implies that the thermal contact resistance may not be ignored in the process of design of the fin–tube heat exchanger. If the validity of our new correlation is profoundly verified by many sample tests in the future study, it will be able to play an important role in the development of fin–tube heat exchangers with higher performance.

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