Heat transfer and pressure drop in compact plain-fin-and-tube heat exchangers

Pedro R. Vilas Bôas
Gherhardt Ribatski
Department of Mechanical Engineering,
Escola de Engenharia de São Carlos (EESC), University of São Paulo (USP)
Av. Trabalhador SanCarlense 400, Centro, São Carlos, SP, Brazil

José M. Saiz Jabardo
Escola Politécnica Superior, Universidade da Coruña
Mendizabal s/n Esteiro
15403 Ferrol, Coruña, Spain

Abstract: *This work presents a detailed review of studies on the air side performance of plain fin-and-tube heat exchangers. A database from the literature was obtained. This database contains over 650 experimental results and covers compact plain-fin-and-tube heat exchanger having the following geometric characteristics: up to 6 tube rows, collar diameter from from 7,53 to 17,26mm, fin pitch from 1,22 to 8,69mm, and transversal and longitudinal tube spacing from 21 to 40mm and from 12,7 to 34,67mm, respectively. These experiments were performed for collar diameter Reynolds numbers in the range of 270 to 16500. The heat transfer and pressure drop data expressed in terms of the Colburn and friction factors, respectively, were reduced according to the collar diameter Reynolds number and, then, analyzed carefully in order to determine the main heat transfer and pressure drop trends and also any inconsistence among databases from distinct authors. After that, the experimental data were also compared against heat transfer and pressure drop correlations from the literature. By this comparison, it was found that the correlations by Rich (1973) and Wang et al. (2000) provided the best heat transfer predictions and Wang et al. (2000) and Wang et al. (1996) provided the best pressure drop prediction.*

Key-words: *Air-side; Coils; Thermal performance*

1. INTRODUCTION

According to Webb (1994), heat exchangers with surface density greater than 700m^2/m^3 belong to the so called “compact heat exchangers”. Fin-and-tube heat exchangers are classified as compact heat exchangers and are characterized by presenting arrays of tubes perpendicular to plate fins. These devices are typically used when one of the fluids is a gas. They are widely used in engineering applications, like power production, food industries, electronics, waste heat recovery, air-conditioning and space applications. The complex airflow pattern through the fin-tube structure makes prediction of heat transfer and pressure drop very difficult in these heat exchangers. As a result most of the published studies are strictly empirical. The main dimensionless numbers used in these studies related to the pressure drop and heat transfer in the gas side are displayed in Table 1. Generally, plots of friction and Colburn-j factors vs. Reynolds number are used in correlating experimental results. Gas side correlations involve dimensionless numbers based on geometric characteristics of the heat exchanger, the gas mass (or volumetric) flow rate and its transport properties.

The main dimensions of fin-tube type heat exchangers are schematically shown in Figure 1. In this figure, Pl and Pt are the longitudinal and transversal tube spacing, respectively; Fp is the fin pitch; t is the fin thickness, Do is the outside tube diameter. Different characteristic lengths are used in the literature, among them the most commonly found are the transversal fin or tube spacing, the longitudinal tube spacing, the hydraulic and outside tube diameters, and the collar diameter. In the present paper the collar diameter will be used, its definition being as follows (refer to Figure 1):

$$Dc = Do + 2\mathrm{t} \tag{1}$$

An extensive literature survey was carried out in the present study in order to identify the best heat transfer and pressure drop correlations for plain fin-tube heat exchangers. In the present paper experimental data from the literature are compared against predictive methods. Although, plain fins are been systematically replaced by wavy, offset strip and louvered fins, the plain fins performance can be used as a reference in order to evaluate the degree of heat transfer enhancement and pressure drop penalty provided by the structured fins.

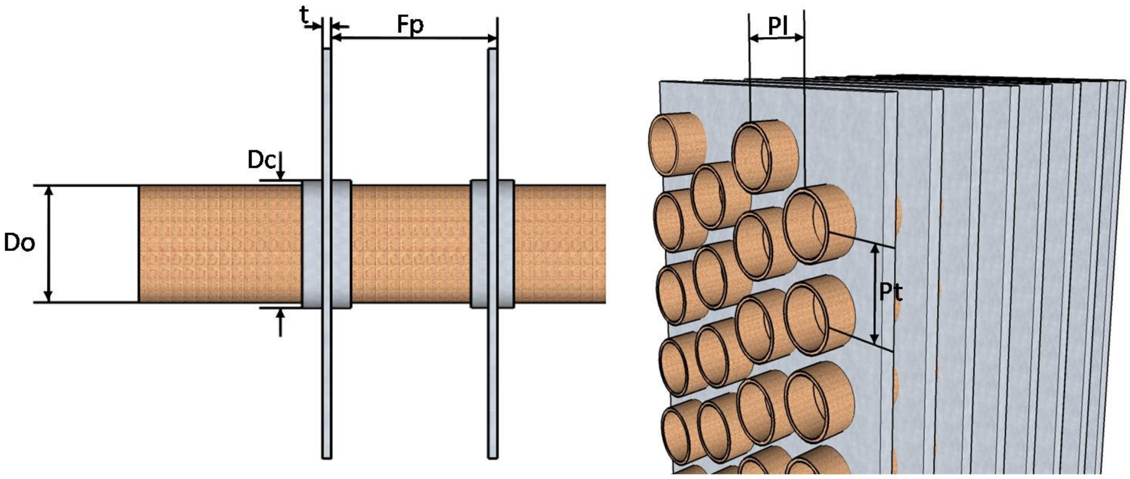

Figure 1. Main characteristic dimensions of fin-and-tube heat exchangers.

Table 1. Dimensionless numbers.

Dimensionless	Definition	Eq. nº	Symbols definition
Reynolds Number	$Re_L = \frac{(\rho V L)}{\mu}$	(2)	ρ: density V: velocity L: characteristic length, Dh; Dc; Pt; etc. μ: dynamic viscosity
Nusselt Number	$Nu = hL/k$	(3)	h: heat transfer coefficient k: thermal conductivity
Graetz Number	$Gz = \frac{(Re_L Pr L)}{(N Pl)}$	(4)	Pr: Prandtl Number N: number of tube rows
Colburn	$j = St\, Pr^{2/3}$	(5)	
Stanton Number	$St = \frac{Nu}{Re_L Pr}$	(6)	
Prandtl Number	$Pr = (cp\, \mu)/k$	(7)	cp: specific heat
Friction factor	$f = (Ac/Ao)(\rho_m/\rho_1)[(2\Delta P \rho_m/Gc^2) - (1+\sigma^2)((\rho_1/\rho_2) - 1)]$	(8)	Ac: minimum flow area Ao: total surface area ρ_m: mean density ρ_1: air side inlet density ΔP: pressure drop Gc: mass flux of the air based on minimum flow area σ: contraction ratio, due to the entrance and exit of air in the heat exchanger ρ_2: air side outlet density

2. DATABASE DESCRIPTION

The objective of the present study is, through a literature survey, to obtain a broad experimental database covering a wide range of geometric parameters involving plate fin-tube heat exchangers. These data were used to evaluate current predictive Colburn-j and friction factor methods for the gas (in general, air) side. Heat transfer data were obtained for 43 different geometries, whereas for pressure drop, experimental results from 22 different geometries were gathered. Detailed description of the air side geometrical characteristics of the surveyed heat exchangers is given in Table 2. A total of 382 heat transfer and 280 pressure drop data points were gathered, their origin being indicated in Table 2. The number of tube rows, N, in the air flow direction, not mentioned so far, have been included in Table 2.

The experimental setup in these studies includes an air loop and a heating circuit. Contrary to most studies, Halıcı et al. (1998) used cooled water and hot air in order to simulate an evaporator condition. Generally, air is driven through the fins and its flow rate and its inlet and outlet temperatures are measured. In the heating circuit, warm water is used as a heating medium. The tube-side fluid flow rate and the inlet and outlet temperatures are also measured. Either LMTD (the logarithmic mean temperature difference method) or ε-NTU methods are used to evaluate the overall heat transfer coefficient. The in-tube heat transfer coefficient is determined from a single-phase forced convection correlation from the literature (either Dittus-Boelter (1930) or Gnielinski (1976)). Thus the external heat transfer coefficient is estimated based on the known overall and internal heat transfer coefficients. The graphical method of Wilson (1915) is also used.

Table 2. Detailed geometric characteristics of the sample plane fin-and-tube heat exchanger covered by the obtained database.

Authors	Dc(mm)	N	Fp(mm)	Pt(mm)	Pl(mm)	t(mm)	Do(mm)	Re_{Dc} range	j factor data points number	f data points number
Rich (1973)	13.34	4	1.23 to 8.69	31.75	27.51	0.15	13.64	638 to 14000	78	80
McQuiston (1978-a)	10.26	4	1.81 to 6.66	25.4	25.4	0.15	9.96	264 to 2211	53	-
Kayansayan (1993)	9.92	4	2.2 to 3.2	25.4	22	0.2	9.52	450 to 16475	36	-
Kayansayan (1993)	16.7	4	2.2 and 3.2	40	34.67	0.2	16.3	450 to 16475	21	-
Wang et al. (1996)	10.23	2 to 6	2.03 to 3.16	25.4	22	0.13	9.97	350 to 7300	30	37
Wang et al. (1996)	10.23	2 to 6	1.77 to 3.21	25.4	22	0.2	9.83	350 to 7300	30	40
Halıcı et al. (1998)	17.26	1 to 4	2.58	40	34.5	0.18	16.9	1500 to 7600	49	-
Wang et al. (2000-b)	7.53	4	1.22 and 1.78	21	12.7	0.12	7	270 to 7870	12	18
Wang et al. (2000-b)	8.51	2 to 4	1.23	25.4	19.05	0.12	7.94	270 to 7870	19	20
Wang et al. (2000-b)	10.23	1 to 4	1.23 to 2.31	25.4	19.05	0.12	9.52	270 to 7870	54	85

to estimate the air side heat transfer coefficient. Friction factors are estimated from Eq. (8), Table 1, based on the air-side pressure drop, generally, measured by differential pressure transducers.

According to Table 2, the surveyed heat exchangers include the following ranges: collar diameter from 7.53 to

17.26mm; up to 6 tube rows; fin pitch from 1.22 to 8.69mm; transversal tube spacing from 21 to 40mm; longitudinal tube spacing from 12.7 to 34.67mm; fin thickness from 0.12 to 0.20mm; collar diameter Reynolds number from 270 to 16500; Colburn-j factors from 0.003 to 0.055; and friction factors from 0.016 to 0.032.

Figure 2 shows the effect of the fin pitch on the Colburn-j and the friction factors according to the experimental data by Wang et al. (2000-b). It is shown that the effect of fin pitch on the Colburn-j factor is almost negligible. Similar behavior was also reported by Rich (1973), Wang et al. (2000-b) and Wang et al. (1996). At low Reynolds number, the friction factor diminishes with the fin pitch. However, at higher Reynolds numbers the fin pitch effect is negligible. Although not shown, it has been determined that at low Reynolds number and reduced fin pitch, the Colburn-j factor diminishes with the number of tube rows. According to Zoghbi Filho (2004) the heat transfer coefficient is no longer affected when the number of the tube rows is higher than 6. This effect can be explained by a mechanism similar to that observed in the entrance region of tubes, though the mechanism in the space between fins is more complex due to the presence of the tubes. Additional arguments in that respect are beyond the scope of the present paper and will not be discussed further.

Figure 3 displays the effect of the collar diameter over the Colburn-j factor this figure is based on data from different authors and different fin pitches. Unfortunately, it has not been possible to devise a clear trend from the available data. Finally, it is worth mentioning that it has not been found any study dealing with the fin thickness, probably due to the apparent result, though an optimization study involving the fin thickness would be in line with either energy or cost reductions.

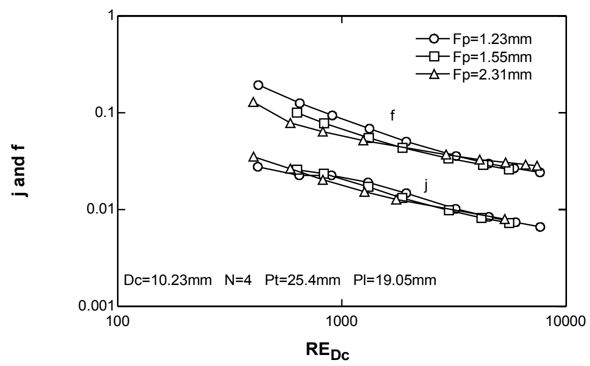

Figure 2. Effect of fin pitch on the friction and Colburn j-factor, Wang et al. (2000-b).

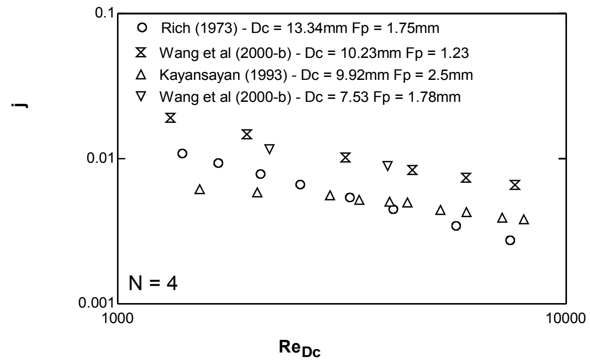

Figure 3. Collar diameter effect on the Colburn-j factor according to data from different authors.

3. CORRELATIONS

Several correlations from the literature for the air side Colburn-j and the friction factor have been investigated aiming at establishing the degree of prediction they provide for the available experimental results. Based on his own database, described in Table 2, Rich (1973) proposed the following correlations:

$$f = 1.70\, Re_{Pt}^{-0,5} \quad (9)$$

$$j = 0.195\, Re_{Pt}^{-0,35} \quad (10)$$

where the Reynolds number is based on the transversal tube spacing. The fin density covered by Rich data varied from 76 to 355 fins/m.

Beecher and Fagan (1987) performed experiments with seven plain fin-and-tube heat exchangers. Contrary to most authors, Beecher and Fagan (1987) used the Nusselt number, Nu_a, and the Graetz number, Gz, to correlate their results. The Graetz number was introduced in order to capture thermal development effects. The Nusselt number, Nu_a, is defined in terms of the convective heat transfer coefficient based on the evaluation of the heat exchanger overall heat transfer coefficient assuming an arithmetic mean temperature difference, instead of the usual logarithmic mean temperature difference. Webb (1990) performed a multiple regression technique to correlate Beecher and Fagan (1987) experimental data obtaining the following expressions:

$$Nu_a = 0.4\mathrm{Gz}^{0.73}(Fp/Dc)^{-0.23}N^{0.23}\text{ , for Gz} \le 25 \tag{11}$$

$$Nu_a = 0.53\mathrm{Gz}^{0.62}(Fp/Dc)^{-0.23}N^{0.31}\text{ , for Gz} > 25 \tag{12}$$

Kayansayan (1993) experimentally evaluated ten different compact heat exchangers with geometric characteristics described in Table 2. He proposed the following heat transfer correlation by correlating his own data by a multiple regression technique:

$$j = 0.15\,Re_{Dh}^{-0.28}\,\Omega^{-0.362} \tag{13}$$

Where the Re_{Dh} is the Reynolds number based in the hydraulic diameter and Ω is defined by the following equation:

$$\Omega = 1 + [4/\pi(Pt/Do)(Pl/Do) - 1](1 - Do/Pt)(Do/[2\mathrm{Fp}(1 - Do/Pt)(1 - t/Fp)]) \tag{14}$$

Wang et al. (1996) performed an extensive experimental study on plain fin-and-tube heat exchangers. They performed experiments with 15 different heat exchangers. Based on their own database, they proposed the following correlations:

$$j = 0.394\,Re_{Dc}(t/Dc)N^{-0.0897}(Fp/Dc) \tag{15}$$

$$f = 1.039\,Re_{Dc}^{-0.418}(t/Dc)N^{-0.0935}(Fp/Dc)^{0.197} \tag{16}$$

where Re_{Dc} is the Reynolds number based on the collar diameter.

By using multiple regression technique and based on their own database and also on data from the literature comprising a total of 47 different fin-and-tube heat exchanger geometries, Kim et al. (1999) proposed the following correlations:

$$j_{N=3} = 0.163\,Re_{Dc}^{-0.369}(Pt/Pl)^{0.106}(Fp/Dc)^{0.0138}(Pt/Dc)^{0.13}\text{ , for N} \ge 3 \tag{17}$$

$$j_{N=1,2}/j_{N=3} = 1.043\left[Re_{Dc}^{0.14}(Pt/Pl)^{-0.564}(Fp/Dc)^{-0.123}(Pt/Dc)^{1.17}\right]^{(3-N)}\text{ , for N} \le 2 \tag{18}$$

A multiple regression technique was also used by Wang et al. (2000-c) to correlate a database including 74 different fin-and-tube heat exchanger geometries. The database included both their data and data from the literature. The Colburn-j factor correlation comprises a total of 27 empirical coefficients and exponents which corresponds to a ratio of almost three heat exchangers for each empirical factor. The number of empirical factors is significant high when compared with those present in other correlations. Wang et al. (2000-c) proposed the following expression:

$$j_{N=1} = 0.108\,Re_{Dc}^{-0.29}(Pt/Pl)^{P1}(Fp/Dc)^{-1.084}(Fp/Dh)^{-0.786}(Fp/Pt)^{P2} \tag{19}$$

The coefficients of Eq. (19) are given by the following set of equations:

$$P1 = 1.9 - 0.23\log_e(Re_{Dc}) \tag{20}$$

$$P2 = -0.236 + 0.126\log_e(Re_{Dc}) \tag{21}$$

$$j_{N\ge 2} = 0.086\,Re_{Dc}^{P3}N^{P4}(Fp/Dc)^{P5}(Fp/Dh)^{P6}(Fp/Pt)^{-0.93} \tag{22}$$

The coefficients of Eq. (22) are determined from the following set:

$$P3 = -0.361 - (0.042\,N)/\log_e(Re_{Dc}) \tag{23}$$

$$P4 = -1.224 - (0.076\,(Pl/Dh)^{1.42})/(\log_e(Re_{Dc})) \tag{24}$$

$$P5 = -0.083 + (0.058\,N)/(\log_e(Re_{Dc})) \tag{25}$$

$$P6 = -5.735 + 1.21\log_e(Re_{Dc}/N) \tag{26}$$

$$f = 0.0267\,Re_{Dc}^{F1}(Pt/Pl)^{F2}(Fp/Dc)^{F3} \tag{27}$$

The coefficients of Eq. (27) are given by:

$$F1 = -0.764 + 0.739(Pt/Pl) + 0.177(Fp/Dc) - (0.00758/N) \tag{28}$$

$$F2 = -15.689 + 64.021/(\log_e(Re_{Dc})) \tag{29}$$

$$F3 = 1.696 - 15.695/\log_e(Re_{Dc}) \tag{30}$$

An empirical correlation for the heat transfer coefficient was also proposed by Yonghan et al. (2004). They tested 22 heat exchangers in a psychrometric chamber, where the air humidity could be controlled. The in-tube cooling fluid was a phase changing refrigerant. Younghan et al. (2004) correlation is given as fallows:

$$j = 0.170\mathrm{N}^{-0.141}(Fp/Do)^{0.384}\,Re_{Dh}^{-0.349} \tag{31}$$

4. RESULTS

The different correlations were evaluated with respect to the available experimental data according to the following parameters: percentage of correlation results within the range of ±30% of the experimental data, the mean error, and the average error. The mean and average errors are, respectively, defined as:

$$\lambda = (1/p)\left(\sum_{1}^{p} value_{calculated} - value_{experimental}/value_{experimental}\right) \tag{32}$$

$$\gamma = (1/p)\left(\sum_{1}^{p} \left|value_{calculated} - value_{experimental}\right|/(value_{experimental})\right) \tag{33}$$

where p is the number of data points.

Table 3. Statistical analysis of the comparison between the predicted and the experimental data.

Deviation	Rich (1973)		Webb (1990)		Kayansayan (1993)		Wang et al. (1996)		Kim (1999)		Wang et al. (2000-c)		Yonghan et al. (2004)	
	j(%)	f(%)	j(%)	f(%)	j(%)	f(%)	j(%)	f(%)	j(%)	f(%)	j(%)	f(%)	j(%)	f(%)
±30%	73.2	18.6	26.2	-	52.3	-	24.3	39.2	68.2	-	55.2	80	26.1	-
Average Deviation (%)	6.9	-42.6	57.2	-	-31.6	-	59.7	-26.3	15.1	-	-9.1	1.4	-41.7	-
Mean Deviation(%)	23.1	42.6	58.7	-	33.0	-	62.4	43.4	24.0	-	34.7	16.3	41.9	-

Table 3 presents the comparison results in terms of the comparison parameters referred to above. According to this table, Rich (1973) and Wang et al. (2000-c) correlations provided the best Colburn-j factor and friction factor predictions, respectively. Figure 4 displays a comparison between calculated and experimental Colburn-j data for the Rich (1973) and Wang et al. (2000-c) correlations, ranked as the first and second best according to present study.

According to Figure 4, the Rich correlation predicts most of the experimental data within ±30%. This figure also reveals that the method proposed by Wang et al. (2000-c) over predicts most of the experimental data. Figure 5 presents the comparison of two of the Wang et al. friction factor correlations, Wang et al. (2000-c) and Wang et al. (1999), with experimental data. It is noted that most of the Wang et al. (2000-c) fall within the ±30% range, whereas the Wang et al. (1999) correlation tends to under predict experimental data.

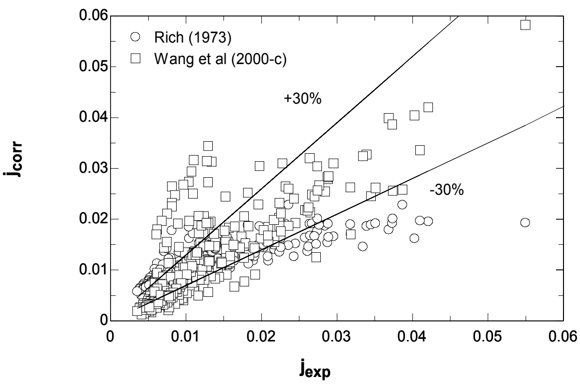

Figure 4. Comparisons of the Rich (1973) and Wang et al. (2000-c) Colburn-j predictive methods against the overall database.

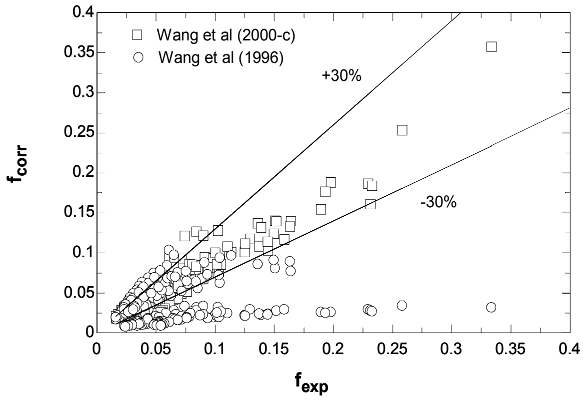

Figure 5. Comparisons of Wang et al. (1996) and Wang et al. (2000-c) friction factor predictive methods against the overall database.

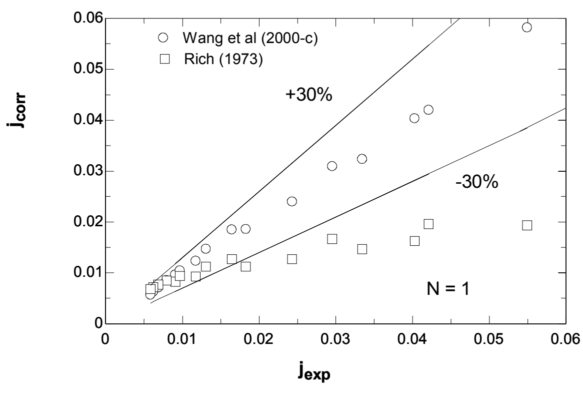

Figure 6. Comparisons of Rich (1973) and Wang et al. (2000-c) Colburn j-factor for heat exchangers having N=1.

Although the Rich (1973) Colburn-j factor correlation is the one that predicts best the overall database, according to Figure 6, and as pointed out by Wang et al. (2000-c), this correlation correlates poorly data corresponding to a number of tube rows lower than 3. The aforementioned good performance by the Rich (1973) correlations seems to be related to the fact that most of the overall experimental data used in the present paper correspond to 4 or more tube rows in the air flow direction. As clearly seen in Figure 6, the Wang et al. (2000-c) correlation is better fitted to one tube row heat exchangers.

5. CONCLUSIONS

A comprehensive literature review on plane fin-and-tube heat exchangers has been performed under the present investigation. A database involving experimental data from different sources has been raised and statistically treated. In addition, the survey has exposed a number of viable proposed correlations for the Colburn-j and friction factors. The correlations performance has been evaluated against the available experimental data. Evaluation results have revealed that the Rich (1973) correlation is the one that fits best the Colburn-j factor as far as the number of tube rows is higher than 3. As for the friction factor, the Wang et al. (2000-c) correlation seems to be the best fitted.

6. ACKNOWLEDGEMENTS

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