PERGAMON

International Journal of Heat and Mass Transfer 43 (2000) 2237–2243

International Journal of HEAT and MASS TRANSFER

www.elsevier.com/locate/ijhmt

Technical Note

A generalized friction correlation for louver fin geometry

Yu-Juei Chang[a], Kuei-Chang Hsu[b], Yur-Tsai Lin[b], Chi-Chuan Wang[a,*]

[a]Energy and Resources Laboratories, Industrial Technology Research Institute, D500 ERL/ITRI, Bldg. 64, 195~6 Section 4, Chung Hsing Road, Chutung 310, Hsinchu, Taiwan

[b]Department of Mechanical Engineering, Yuan-Ze University, Tauyuan, Taiwan

Received 6 June 1999; received in revised form 14 September 1999

1. Introduction

This study is a continuation of a previous study by Chang and Wang [1] who presented a general heat transfer correlation for louver fin geometry based on 91 louvered fin heat exchangers having flat tube configuration (Fig. 1). The database of the 91 samples was collected from Davenport [2] (30 samples, Fig. 1, Type A, 529 data points), Tanaka et al. [3] (one sample, Fig. 1, Type C, 6 data points), Achaichia and Cowell [4] (15 samples, Fig. 1, Type B, 193 data points), Webb [5] (five samples, Fig. 1, Type C, 33 data points), Sunden and Svantesson [6] (six samples, Fig. 1, Type C, 63 data points), Webb and Jung [7] (six samples, Fig. 1, Type C and Type E, 36 data points), Rugh et al. [8] (1 sample, Fig. 1, Type D, 10 data points), and Chang and Wang [9] (27 samples, Fig. 1, Type C, 239 data points). The objective of this study is to propose a friction correlation that can correlate the results of 1109 data points.

2. The data bank

Detailed geometrical dimensions of the 91 samples can be found from previous study by Chang and Wang [1]. Relevant definitions of the geometric parameter for the present louver fin geometry are shown in Fig. 2. The frictional performance is in terms of Fanning friction factor f, i.e.

$$f=\frac{A_c\rho_m}{A\rho_1}\left[\frac{2\rho_1\Delta P}{G_c^2}-(K_c+1-\sigma^2)-2\left(\frac{\rho_1}{\rho_2}-1\right)+(1-\sigma^2-K_e)\frac{\rho_1}{\rho_2}\right] \quad (1)$$

where $G_c=\rho_m V_c$ and V_c is the maximum velocity in the core of the heat exchanger. Note that the core entrance and exit losses were subtracted. The entrance and exit loss coefficients of K_c and K_e (the abrupt contraction and expansion coefficients) were evaluated from Fig. 5-4 of Kays and London [10] at $Re_{Dh}=\infty$. The database for friction factors are shown in Fig. 3. As seen, the frictional performance varies considerably from different data sources.

3. The proposed correlation

After a trial and error process, the final equation form of the friction factor is given as follows:

$$f=f1^* f2^* f3 \quad (2)$$

where

* Corresponding author. Tel.: +886-3-5916294; fax: +886-3-5820250.

E-mail address: ccwang@itri.org.tw (C.-C. Wang).

PII: S0017-9310(99)00289-6

Nomenclature

A	total surface area [m^2]
A_c	minimum flow area [m^2]
D_h	hydraulic diameter of fin array [mm]
D_m	major tube diameter [mm]
f	Fanning friction factor dimensionless
$f1, f2, f3$	correlation parameter
F_d	fin depth [mm]
F_l	fin length [mm]
F_p	fin pitch [mm]
F_t	fin thickness [mm]
G_c	$\rho_m V_c$, mass flux at minimum flow area [kg m^{-2}]
K_c	abrupt contraction coefficient, dimensionless
K_e	abrupt expansion coefficient, dimensionless
L_h	louver height [mm]
L_l	louver length [mm]
L_p	louver pitch [mm]
M	number of test data point
Re_{Dh}	Reynolds number based on hydraulic diameter, dimensionless
Re_{Lp}	Reynolds number based on louver pitch, dimensionless
T_h	$T_p - D_m$ [mm]
T_p	tube pitch [mm]
T_d	tube depth [mm]
V_c	maximum velocity [m s^{-1}]
ΔP	pressure drop [Pa]
ρ_1	inlet air density [kg m^{-3}]
ρ_2	outlet air density [kg m^{-3}]
ρ_m	mean density [kg m^{-3}]
σ	contraction ratio, dimensionless
θ	louver angle [deg]
Subscripts	
exp	experimental value
pred	prediction value by the proposed correlation.

$$f1 = \begin{cases} 14.39 Re_{Lp}^{(-0.805 F_p/F_l)} (\log_e(1.0 + (F_p/L_p)))^{3.04} & Re_{Lp} < 150 \\ 4.97 Re_{Lp}^{0.6049 - 1.064/\theta^{0.2}} (\log_e((F_t/F_p)^{0.5} + 0.9))^{-0.527} & 150 < Re_{Lp} < 5,000 \end{cases} \quad (3)$$

$$f2 = \begin{cases} (\log_e((F_t/F_p)^{0.48} + 0.9))^{-1.435} (D_h/L_p)^{-3.01} (\log_e(0.5 Re_{Lp}))^{-3.01} & Re_{Lp} < 150 \\ ((D_h/L_p) \log_e(0.3 Re_{Lp}))^{-2.966} (F_p/L_l)^{-0.7931(T_p/T_h)} & 150 < Re_{Lp} < 5,000 \end{cases} \quad (4)$$

$$f3 = \begin{cases} (F_p/L_l)^{-0.308} (F_d/L_l)^{-0.308} (e^{-0.1167 T_p/D_m}) \theta^{0.35} & Re_{Lp} < 150 \\ (T_p/D_m)^{-0.0446} \log_e(1.2 + (L_p/F_p)^{1.4})^{-3.553} \theta^{-0.477} & 150 < Re_{Lp} < 5,000 \end{cases} \quad (5)$$

Fig. 4 shows the comparison of the experimental data with Eq. (2). It is shown that 83.14% of the data of friction factor are correlated within ±15%, and the present correlation gives a mean deviation of 9.21%.

4. Tests of the various correlations against the data bank

In addition to the correlation proposed in this paper, several other correlations were tested against the data. These correlations include those by Davenport [2] and Achaichia and Cowell [4]. The f-correlation by Davenport [2] is

$$f = 5.47 Re_{Lp}^{-0.72} (L_t/F_l)^{0.89} F_l^{0.23} L_p^{0.2} L_h^{0.37} \quad (\text{for } 70 < Re_{Dh} < 900) \quad (6)$$

$$f = 0.494 Re_{Lp}^{-0.39} (L_h/L_p)^{0.33} (L_l/F_l)^{1.1} F_l^{0.46} \quad (\text{for } 1000 < Re_{Dh} < 4,000) \quad (7)$$

The correlation by Achaichia and Cowell [4] is

$$f = 10.4 Re_{Lp}^{-1.17} F_p^{0.05} L_p^{1.24} T_p^{0.83} F_l^{0.25} \quad (\text{for } Re_{Lp} < 150) \quad (8)$$

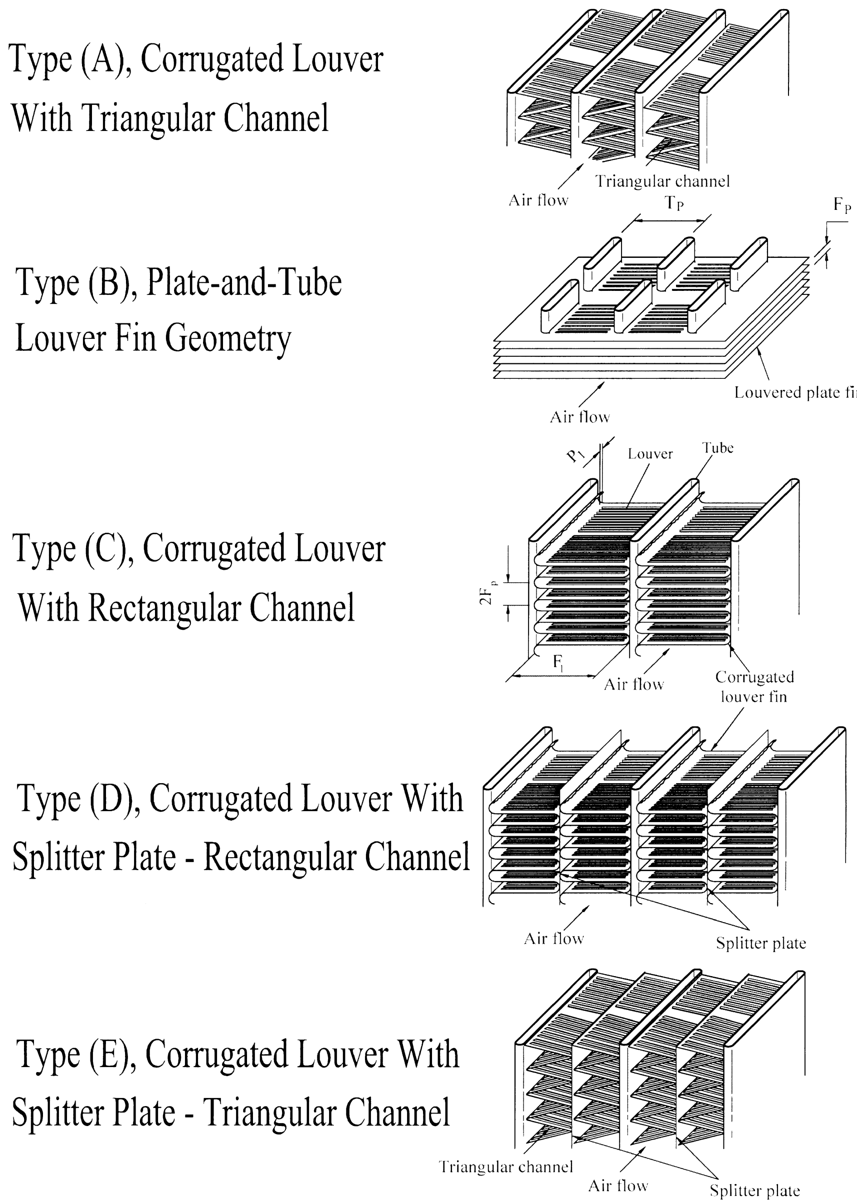

Fig. 1. Type of louvered fin heat exchangers.

$$f = 0.895 f_A^{1.07} F_p^{-0.22} L_p^{0.25} T_p^{0.26} F_l^{0.33} \quad (\text{for } 150 < Re_{Lp} < 3,000) \tag{9}$$

where

$$f_A = 596 Re_{Lp}^{(0.318 \log_{10}(Re_{Lp}) - 2.25)} \tag{10}$$

Notice the above-mentioned equations (Eqs. (6)–(9)) are dimensional equations (units in mm). The results of the comparison of the Davenport correlation [2] and Achaichia and Cowell correlation [4] to the database are shown in Table 1. As seen, the mean deviation of the present correlation, the Davenport correlation and the Achaichia and Cowell [4] correlation, are 9.21, 17.5, and 102.5%, respectively. The Achaichia and Cowell [4] correlation shows significant overpredictions. This is probably due to the fact that their database were of type B (Fig. 1) which have multiple numbers of tube rows. Hence periodic contraction and expansion of the airflow within the heat exchanger may result in higher pressure drops.

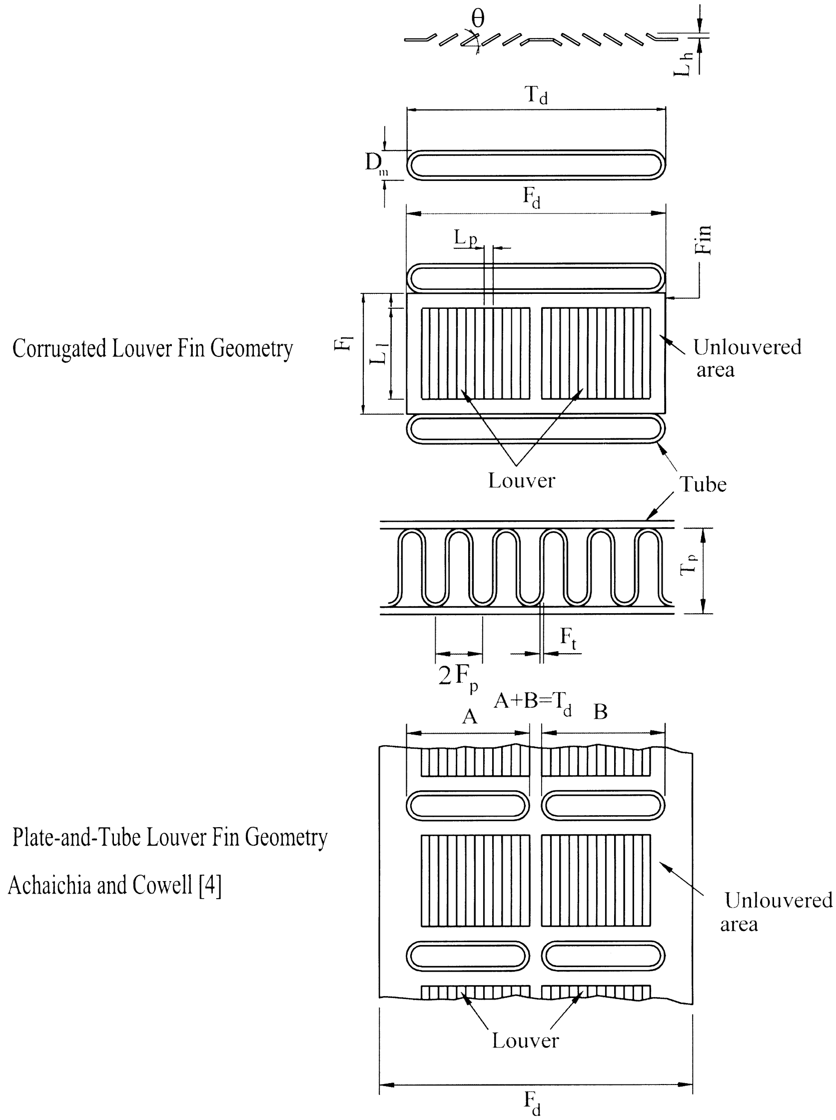

Fig. 2. Definition of various geometric parameters.

Table 1
Comparison of the correlation with all the experimental data[a]

Deviation	Present correlation	Davenport [2]	Achaichia and Cowell [4]
± 10%	68.35%	45.49%	14.97%
± 15%	83.14%	54.48%	19.12%
± 20%	90.89%	64.05%	21.73%
± 25%	94.86%	70.81%	24.17%
Average deviation	0.027%	−11.94%	100.83%
Mean deviation	9.21%	17.50%	102.48%

[a] Average deviation$= \frac{1}{M}(\sum_1^M \frac{f_{\mathrm{pred}}-f_{\mathrm{exp}}}{f_{\mathrm{exp}}}) \times 100\%$; mean deviation$= \frac{1}{M}(\sum_1^M \frac{|f_{\mathrm{pred}}-f_{\mathrm{exp}}|}{f_{\mathrm{exp}}}) \times 100\%$; M: number of data points.

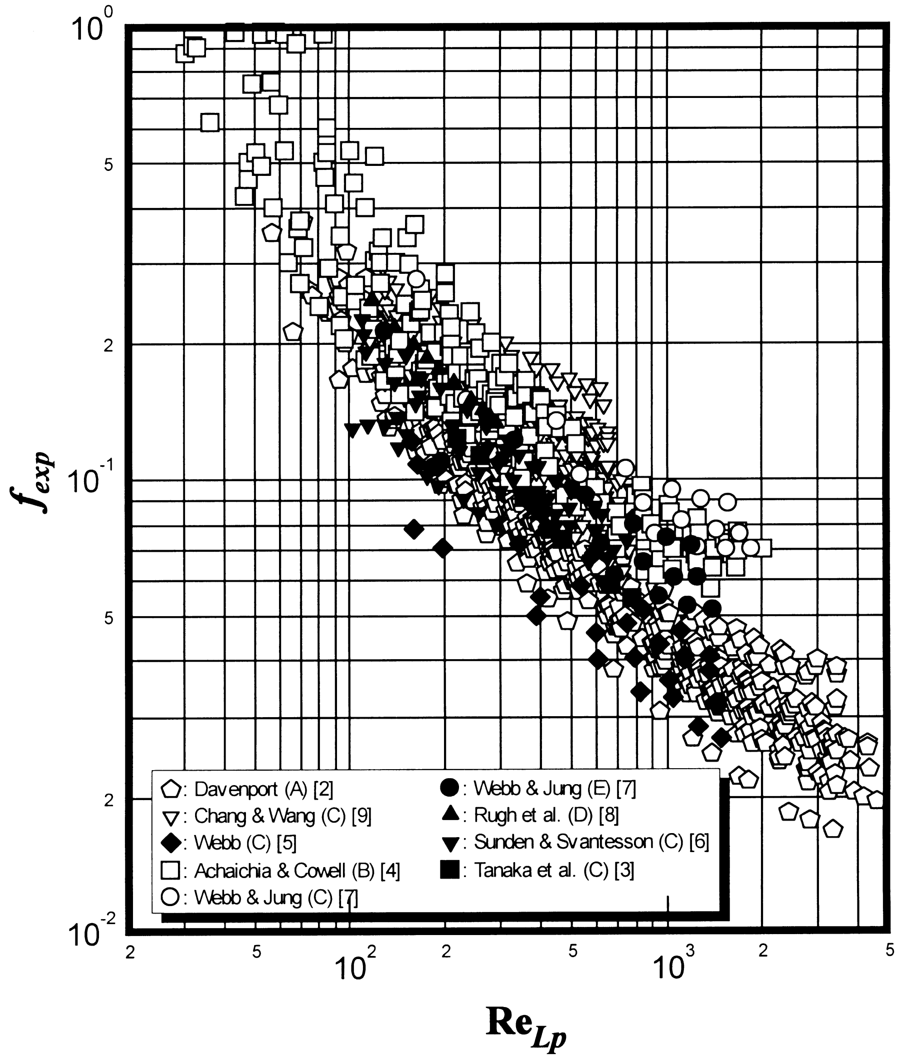

Fig. 3. f vs Re_{LP} for all the test database.

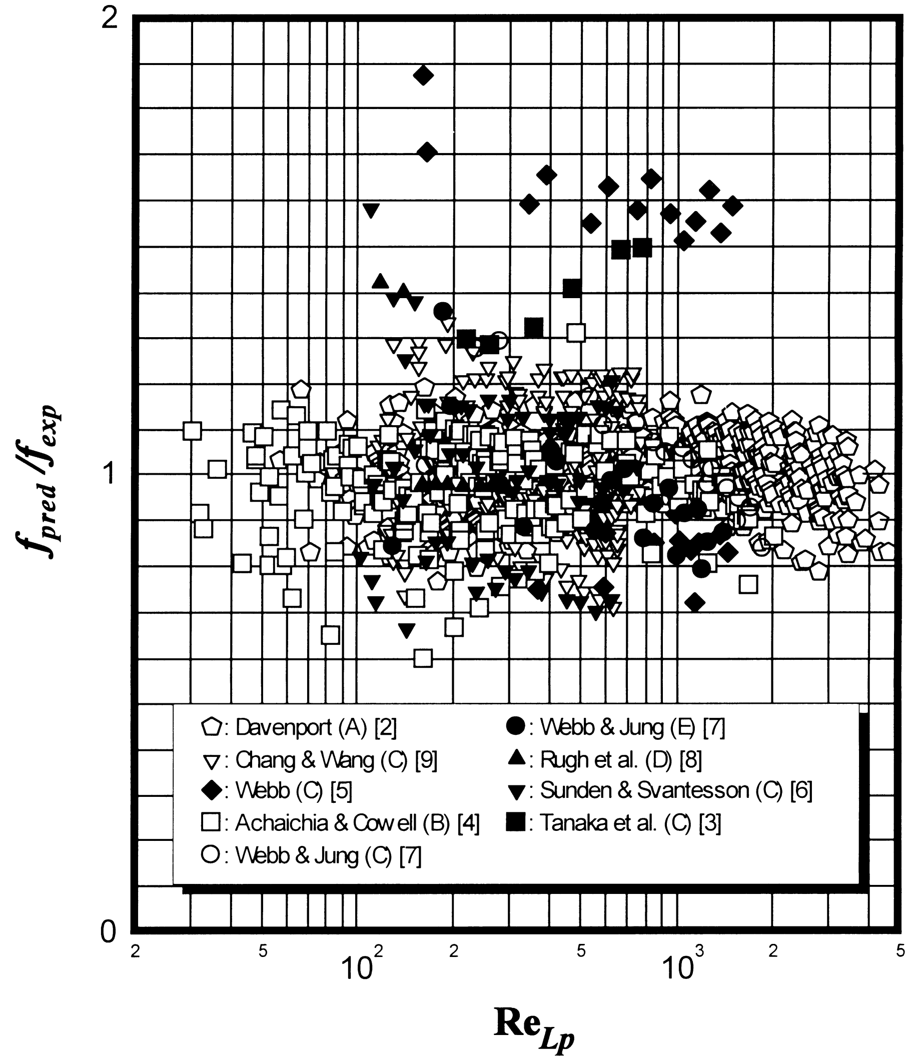

Fig. 4. Heat transfer error plots for the all louver fin samples.

5. Conclusions

A generalized frictional correlation for louver fin geometry is developed in the present study. A total of 91 samples of louver fin heat exchangers are used in the regression analysis. The proposed correlation gives a mean deviation of 9.21%, it is shown that 83.14% of the frictional data can be correlated within ±15%.

Acknowledgements

The authors would like to express gratitude for the Energy R&D foundation funding from the Energy Commission of the Ministry of Economic Affairs, Taiwan. The authors are indebted to Prof. Ralph Webb for providing valuable suggestions and PSU radiator data.

References

[1] Y.J. Chang, C.C. Wang, A generalized heat transfer correlation for louver fin geometry, Int. J. of Heat and Mass Transfer 40 (3) (1997) 533–544.

[2] C.J. Davenport, Correlation for heat transfer and flow friction characteristics of louvered fin, AIChE Symposium Series 79 (25) (1983) 19–27.

[3] T. Tanaka, M. Itoh, M. Kudoh, A. Tomita, Improvement of compact heat exchangers with inclined louvered fins, Bulletin of JSME 27 (224) (1984) 219–226.

[4] A. Achaichia, T.A. Cowell, Heat transfer and pressure

drop characteristics of flat tube and louvered plate fin surfaces, Exp. Thermal and Fluid Sci. 1 (1988) 147–157.
[5] R.L. Webb, PSU radiators test data, unpublished data for five radiators, (1988).
[6] B. Sunden, J. Svantesson, Correlation of *j*- and *f*-factors for multilouvered heat transfer surfaces, in: Proceedings of Third UK National Heat Transfer Conference, 1992, pp. 805–811.
[7] R.L. Webb, S.H. Jung, Air-side performance of enhanced brazed aluminum heat exchangers, ASHRAE Transactions 98 (2) (1992) 391–401.
[8] J.P. Rugh, J.T. Pearson, S. Ramadhyani, A study of a very compact heat exchanger used for passenger compartment heating in automobiles, in: Compact Heat Exchangers for Power and Process Industries, ASME Symp. Ser. HTD-Vol. 201 (1992) 15–24.
[9] Y.J. Chang, C.C. Wang, Air side performance of brazed aluminum heat exchangers, J. of Enhanced Heat Transfer 3 (1) (1996) 15–28.
[10] W.M. Kays, A.L. London, Compact Heat Exchangers, 3rd ed., McGraw-Hill, New York, 1984.