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Experimental Investigation of Heat Transfer and Friction Characteristics in L-Footed Spiral Fin-Tube Banks

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Abstract

In this experimental study, air side heat transfer and flow characteristics of L-footed spiral fin-and-tube heat exchanger have been analyzed. There were four rows of tubes in the air flow direction and four tubes per row in the heat exchanger. The hot water was flowed through the tubes while the ambient air was flowed cross flow over the tubes. The results were presented as plots of friction factor f and Colburn j factor against Reynolds number. The air-side heat transfer coefficient and pressure drop were also presented against frontal air velocity. Additionally, The correlations for j and f factors were also obtained from the experimental data.

Keywords: L-footed spiral fin, Heat exchanger, Extended surface, Heat transfer

L-Ayaklı Spiral Kanatlı-Boru Demetlerinde Isı Transfer ve Sürtünme Karakteristiklerinin Deneysel İncelenmesi

Öz

Bu deneysel çalışmada, L-ayaklı spiral kanatlı boru tipi bir ısı değiştiricinin hava tarafındaki ısı transferi ve akış özellikleri analiz incelenmiştir. Isı değiştiricide hava akışı yönünde dört sıra boru ve her sıradada dört adet boru vardır. Çevre hava borular üzerinden çapraz olarak geçerilirken, sıcak su ise boruların içerisinden geçirilmiştir. Sonuçlar, Reynolds sayısına karşı sürtünme faktörü f ve Colburn j faktörünün grafikleri olarak sunulmuştur. Hava tarafındaki ısı taşınım katsayısı ve basınç düşüşünün de hava hızına karşı değişimi gösterilmiştir. Bunlara ek olarak, j ve f faktörlerinin korelasyonları da deneysel verilerden elde edilmiştir.

Anahtar kelimeler: L-ayaklı spiral kanat, Isı değiştirici, Genişletilmiş yüzey, Isı transferi

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1. Introduction

Heat exchangers are widely used in many engineering applications. In heat exchangers, when one of the fluids is a gas which is passed in cross flow over tubes, liquid is passed through the tubes. The dominant thermal resistance in an air cooled heat transfer between the hot and cold fluids is usually seen on the air side. Consequently, usage of finned surface on the air side has been a wide application in terms of improving the total thermal performance of the system. Nuntaphan et al. (2005a and 2005b) have analyzed the air-side performance of crimped spiral fin-and-tube heat exchangers with cross-flow for wet and dry surface conditions. Pongsoi et al. (2011 and 2012a) have experimentally analyzed effect of fin pitches and the air-side performance of the crimped spiral fin-and-tube heat exchangers. Kwaguchi et al. (2006a and 2006b) have analyzed the heat transfer and pressure drop characteristics for forced convection in spiral finned tube bundles and in serrated finned tube bundles. Tang et al. (2009) have experimentally researched the air-side heat transfer and friction characteristics of the heat exchanger in which five different types of fins were used. Pongsoi et al. (2012b and 2013) have analyzed the air-side performance of the L-footed spiral fin-and-tube heat exchanger with multipass parallel and counter cross-flow. Fa Jiang et al. (2012) have experimentally analyzed the heat transfer and flowing resistance characteristics of the cross-flow over spiral fin-and-tube heat exchangers for different fin height, pitch. Ma et al. (2012) have analyzed the influence of the fin density on the thermal hydraulic performance of serrated finned tube heat exchanger. The air-side performance in serrated welded spiral fin tube heat exchangers was experimentally researched for Z-flow type and high Reynolds (Kiatpachai et al., 2015).

2. Experimental Set-Up

Schematic representation of the experimental setup used in the study is shown in Figure 1 (Kırtepe, 2014).

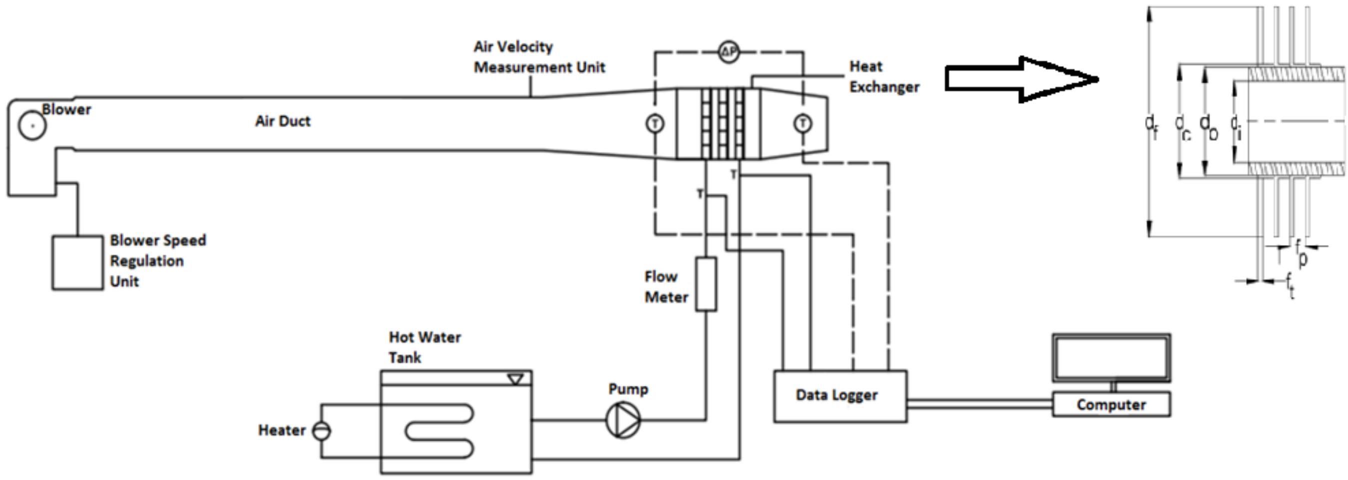


Figure 1. Schematic drawing of the experimental set-up

The experimental set-up consisted of heat exchanger, air duct, blower, blower speed regulation unit, hot water tank, water pump, temperature control system (Proportional Integral Derivative), velocity and temperature measuring devices and data collection unit. The tubes in the heat exchanger were placed as four lines each as shifted rows in both parallel and perpendicular directions to the air flow (Figure 2). The heat exchanger dimensions were 262.5 mm x 400 mm x 262 mm. Tubes were in staggered arrangement in the heat exchanger.

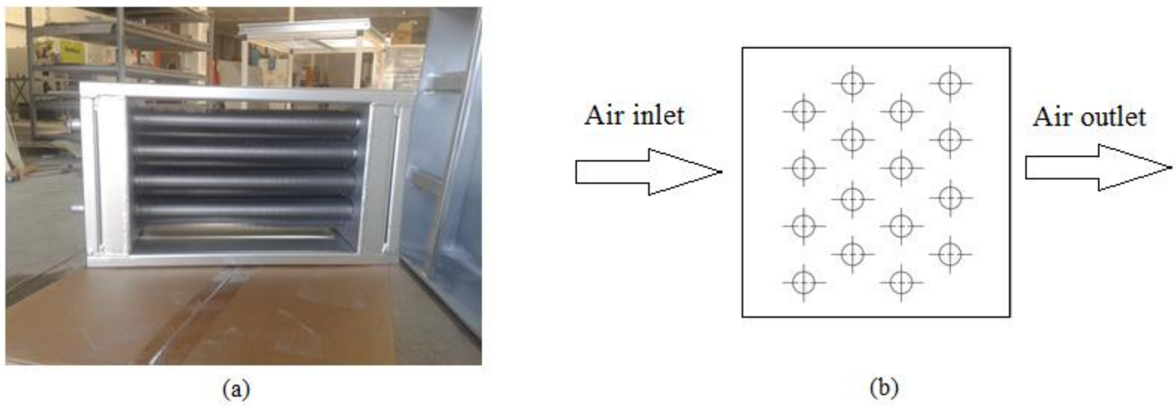


Figure 2. Heat exchanger a) Photo b) Schematic drawing

Table 1. Structure parameters of spiral finned tube

Inner diameter of tube (d_i)	16.1 mm
Outer diameter of tube (d_o)	21.3 mm
Outer diameter of fin collar (d_c)	22.3 mm
Outer diameter of fin (d_f)	45.3 mm
Fin thickness (f_t)	0.5 mm
Fin pitch (f_p)	3.3 mm
Spacing between the tubes which are at cross direction of the tube bundle	55 mm
Diagonal distance between the tubes in the tube bundle	55 mm
Spacing between the tubes which are at axial direction of the tube bundle	47.63 mm

In the experiments, the frontal air velocity of the ambient air passing in cross-flow over the tubes was kept in six different values (0.52, 1.56, 2.56, 3.60, 4.50, 5,44 m/s) while the inlet temperature of the water passing through the pipes was kept in four different values (40, 50, 60, 70 °C) and the flow rate of the water was fixed to 3.83 X 10^{-5} m^3/s.

The frontal air velocity was measured at a point in the heat exchanger before the fluid enters the tube banks. For measuring the frontal air velocity, telescopic air velocity measuring sensor with the operating range between 0 and 10 m/s and between -20 and 70 °C temperature was used. Accuracy of the sensor in terms of temperature and velocity was ±0.5 °C and ±0.03 m/s, respectively. For measuring the pressure drop, differential pressure meter with operating range between 0 and 500 Pascal and with accuracy of ±%2 was used. For measuring the water flow rate, an electromagnetic flow meter with operating range between 0 °C and 80 °C and with accuracy of ±%0.002 was used. J Type thermocouple with operating range between -200 and +800 °C and with accuracy of ±0.5 °C was used for measuring the air temperature. PT-100 with accuracy of ±0.01 °C was used for measuring the water temperature.

3. Material and Method

In this study, *P-NTU* method was used to find out the air side performance of L-footed spiral fin and tube heat exchanger with cross flow.

3.1Temperature Effectiveness – Number of Transfer Units (P-NTU) Method

The heat transfer rate from the hot fluid to the cold fluid in this method is defined below.

$$Q = P_1(\dot{m}c_p)_1\Delta T_{max} = P_2(\dot{m}c_p)_2\Delta T_{max} \tag{1}$$

where P indicates the temperature effectiveness. 1 and 2 indexes on the other hand indicate the fluids (regardless of being hot or cold) (Shah and Sekulic, 2003).

The temperature effectiveness (P_1 and P_2) for every fluid is:

$$P_1 = \frac{T_{1,o}-T_{1,i}}{T_{2,i}-T_{1,i}} = \frac{T_{1,o}-T_{1,i}}{\Delta T_{max}} \quad \text{and} \quad P_2 = \frac{T_{2,i}-T_{2,o}}{T_{2,i}-T_{1,i}} = \frac{T_{2,i}-T_{2,o}}{\Delta T_{max}} \tag{2}$$

The numbers of transfer units (NTU_1 and NTU_2) are:

$$NTU_1 = \frac{UA}{(\dot{m}_1 c_{p1})} = \frac{UA}{C_1} \quad \text{and} \quad NTU_2 = \frac{UA}{(\dot{m}_2 c_{p2})} = \frac{UA}{C_2} \tag{3}$$

The heat capacity rate ratio (R_1 and R_2) is defined as:

$$R_1 = \frac{\dot{m}_1 C_{p1}}{\dot{m}_2 C_{p2}} = \frac{C_1}{C_2} \text{ and } \quad R_2 = \frac{\dot{m}_2 C_{p2}}{\dot{m}_1 C_{p1}} = \frac{C_2}{C_1} \tag{4}$$

The temperature effectiveness, P, is a function of *NTU*, *R* and flow arrangement. In the crossflow heat exchanger in which fluid 1 is mixed while fluid 2 is unmixed, the temperature effectiveness (P_1) and the number of transfer unit (NTU_1) of the fluid 1 are given below (Shah and Sekulic, 2003).

$$P_1 = 1 - exp\left[-\frac{1-\exp(-R_1 NTU_1)}{R_1}\right] \tag{5}$$

$$NTU_1 = \frac{1}{R_1}\ln\left[\frac{1}{1+R_1\ln(1-P_1)}\right] \tag{6}$$

3.2 Calculation of the Heat Transfer Coefficient

The heat transfer rates are calculated for both air-side and water-side as follows:

$$Q_a = \dot{m}_a c_{P,a}\Delta T_a \quad \text{and} \quad Q_w = \dot{m}_w c_{P,w}\Delta T_w \tag{7}$$

The total rate of heat transfer used in performance calculations is the average of the heat transfer rate of air-side and water-side. Air-side heat transfer coefficient h_i is found with the below mentioned equation through the usage of equations 2, 3, 5 and 6.

$$\frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{\ln(d_o/d_i)}{2\pi k_t L} + \frac{\ln(d_c/d_o)}{2\pi k_f L} + \frac{1}{\eta_o h_o A_o} \tag{8}$$

The water-side heat transfer coefficient h_o is calculated with the formula suggested by Sieder and Tate (Kraus et al., 2001)

$$h_i = \frac{k_s}{d_i} 1{,}86\left(\frac{Re_i Pr_s d_i}{L}\right)^{1/3}\left(\frac{\mu}{\mu_y}\right)^{0{,}14} \tag{9}$$

The overall surface efficiency is:

$$\eta_o = 1 - \frac{A_f}{A_o}(1-\eta) \tag{10}$$

where $A_o = A_b + A_f$ and A_o indicates the total surface area of the finned tube whereas A_b= unfinned heat transfer area and A_f= finned heat transfer area. Unfinned and finned heat transfer areas are obtained with the below presented equation (Pongsoi et al., 2012b):

$$A_b = N_T N_L\left[\pi d_c L - \left(\sqrt{f_p^2 + (\pi d_c)^2}\right) \times f_t\left(\frac{L}{f_p}\right)\right] \tag{11}$$

$$A_f = N_T N_L\left(\frac{L}{f_p}\right)\left[0.5\pi d_f^2 - 0.5\pi d_c^2 + \pi d_f f_t\right] \tag{12}$$

Fin efficiency was found with the formula suggested by Gardner (Pongsoi et al., 2012b; Kraus et al., 2001).

$$\eta = \frac{2\psi}{\phi(1+\psi)}\frac{I_1(\phi R_f)K_1(\phi R_o) - I_1(\phi R_o)K_1(\phi R_f)}{I_0(\phi R_o)K_1(\phi R_f) + I_1(\phi R_f)K_0(\phi R_o)} \tag{13}$$

where, ϕ parameter is defined below:

$$\phi = (r_f - r_o)^{3/2}\left(\frac{2h_o}{k_f A_p}\right)^{1/2} \tag{14}$$

where A_p is the profile area of fin:

$$A_p = f_t(r_f - r_o) \tag{15}$$

$R_f = 1/(1-\psi)$, $R_o = \psi/(1-\psi)$ functions are defined with the usage of ratio of the diameters $\psi = d_o/d_f$.

The air-side heat transfer coefficient (h_o) was calculated with the above mentioned equations.

The air-side heat transfer characteristics of the L-footed spiral fin-and-tube heat exchanger were explained on the basis of Colburn-j factor (Nuntaphan et al., 2005a; Kiatpachai et al., 2015; Wang et al., 1996).

$$j = St\, Pr^{2/3} = \frac{h_o}{\rho_a V_{a,max}\, c_{P_a}} Pr^{2/3} \tag{16}$$

Flow characteristics given in Fanning friction factor are obtained from the equation suggested by Kays and London. In this equation, inlet and outlet pressure losses are considered as well

(Nuntaphan et al., 2005a; Kiatpachai et al., 2015; Wang et al., 1996).

$$f = \frac{A_{min}}{A_o}\left(\frac{\rho_m}{\rho_i}\right)\left[\frac{2\ \Delta P\ \rho_i}{G_c^2} - (1+\sigma^2)\left(\frac{\rho_i}{\rho_o}-1\right)\right] \quad (17)$$

where, σ is the ratio of minimum free flow area to the frontal area, A_{min} is the minimum free flow area, A_o is the total heat transfer area and G_c is the mass velocity.

$$G_c\ (kg/sm^2) = \frac{\dot{m}\ (kg/s)}{A_c\ (m^2)} = \frac{\rho U_\infty A_{fr}}{A_c} \quad (18)$$

where, $\dot{m}$ is the mass flow rate (kg/s), U_∞ is the frontal air velocity (m/s), A_{fr} is the frontal area (m^2). U_∞ was measured, G_c was calculated.

Because of the variation of the air side temperature is low, so inlet and outlet effects are ignored, namely when considered as $\rho_i=\rho_o$ and $\rho_m=(\rho_i+\rho_o)/2$, Fanning friction factor can be obtained as presented below:

$$f = \frac{A_{min}\rho_m}{A_o}\left[\frac{2\Delta P}{G_c{}^2}\right] \quad (19)$$

4. Results and Discussion

Figure 3 shows the variation of the air side heat transfer coefficient with the frontal air velocity for the different water inlet temperatures. Figure 4 shows the variation of the pressured drop with the frontal air velocity. The air-side heat transfer coefficient rapidly increases with the frontal air velocity. It is observed that water inlet temperature does not have a significantly effect on the air-side heat transfer coefficient. As the frontal air velocity increases, the pressure drop occurring at the air-side of the heat exchanger also increases.

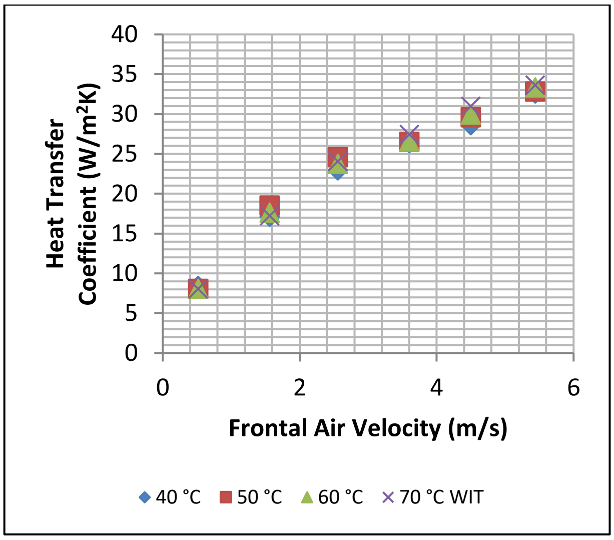


Figure 3. Variation of the air side heat transfer coefficient with the frontal air velocity for different water inlet temperatures (WIT)

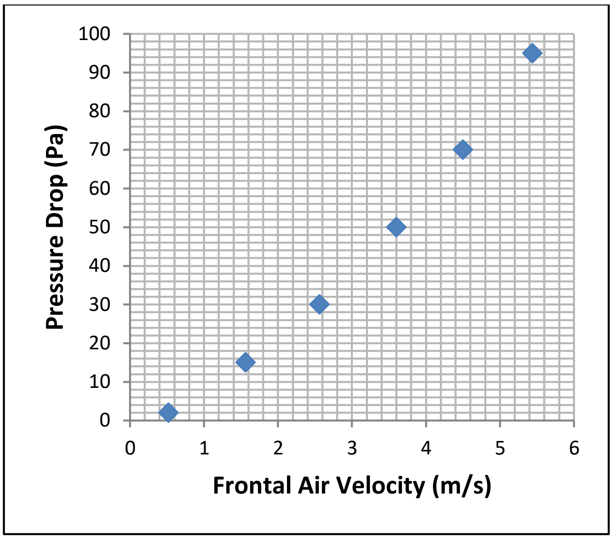


Figure 4. Variation of the pressure drop with the frontal air velocity

The basic surface characteristics of the heat exchangers are generally presented in a dimensionless form as Colburn *j* factor and Fanning friction factor *f*. In Figure 5, changes in Colburn *j* factor and Fanning friction factor *f* with the Reynolds number based on the outer diameter of fin collar can be seen. As the Reynolds number increases, both Colburn *j* factor and Fanning friction factor *f* decrease.

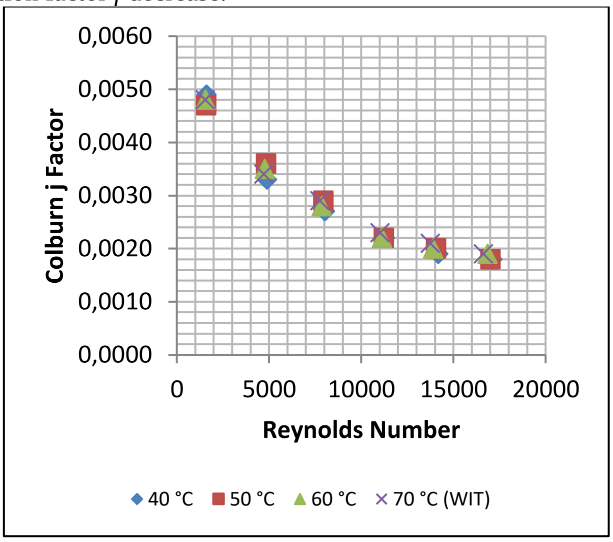


Figure 5a. Variation of Colburn *j* factor with Reynolds number for different water inlet temperatures

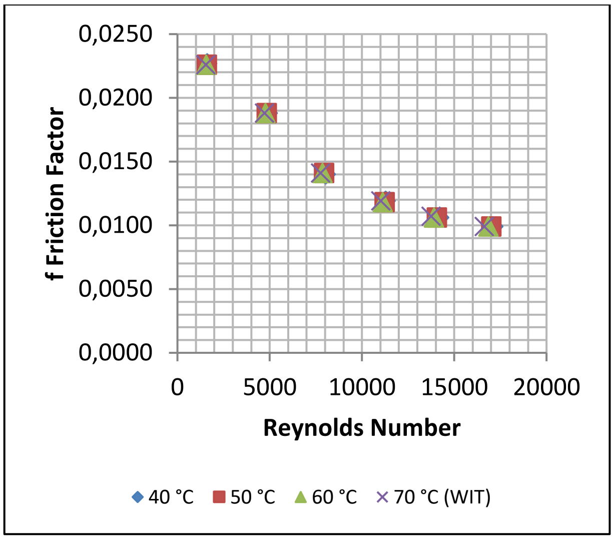


Figure 5b. Variation of *f* Fanning friction factor with Reynolds number for different water inlet temperatures

Colburn *j* factor and Fanning friction factor *f* were obtained with the usage of experimental data and depending upon air-side Reynolds Number as $j = aRe^b$ve $f = mRe^n$ form. Here; a, b, m and n were obtained from the experimental data. Formulas suggested for heat transfer performance and friction performance are presented below:

$$j = 0.08287Re^{-0,3838} \text{ ve } f = 0.2684Re^{-0,3307} \tag{20}$$

$$MeanDeviation = \frac{1}{n}\left(\sum_{1}^{n}\frac{\left|\varphi_{corr} - \varphi_{exp}\right|}{\varphi_{exp}}\right)\times 100\% \tag{21}$$

The correlation coefficient was one of the primary criterion for selecting the best equation. In addition to the correlation coefficient, the various statistical parameters such as; mean bias error (MBE), root mean square error (RMSE) were used to determine the quality of the fit.

$$CC = \frac{\sum_{i=1}^{N}(y_i-\bar{y})(x_i-\bar{x})}{\sqrt{[\sum_{i=1}^{N}(y_i-\bar{y})^2][\sum_{i=1}^{N}(x_i-\bar{x})^2]}} \tag{22}$$

Ideally CC should be 1.

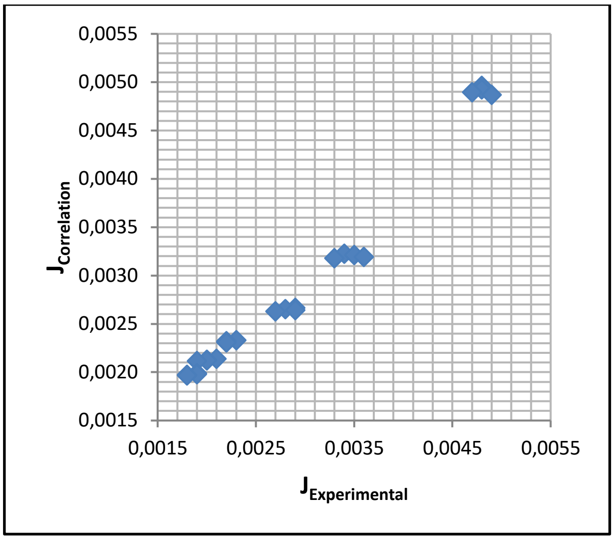
$$MBE = \sum_{i=1}^{N}(y_i - x_i)/N \tag{23}$$

Ideally a zero value of MBE should be obtained.

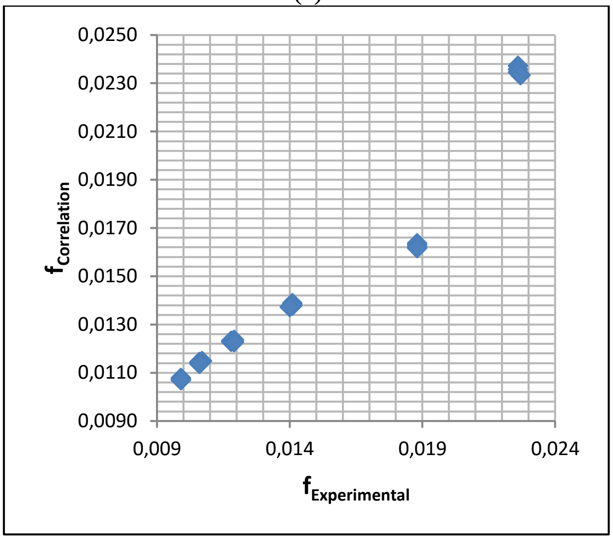
$$RMSE = \sqrt{\sum_{i=1}^{N}(y_i - x_i)^2/N} \tag{24}$$

Ideally a zero value of RMSE should be obtained.

For a model to be good estimator, the mean values of all the errors terms MBE, RMSE should be small. In this study, the statistical tests results were computed as CC=0.965, MBE=2.9x10^{-5}, RMSE= 1.22x10^{-3} for friction factor, CC=0.986, MBE=4.51x10^{-6}, RMSE= 1.73x10^{-4} for Colburn factor.



(a)

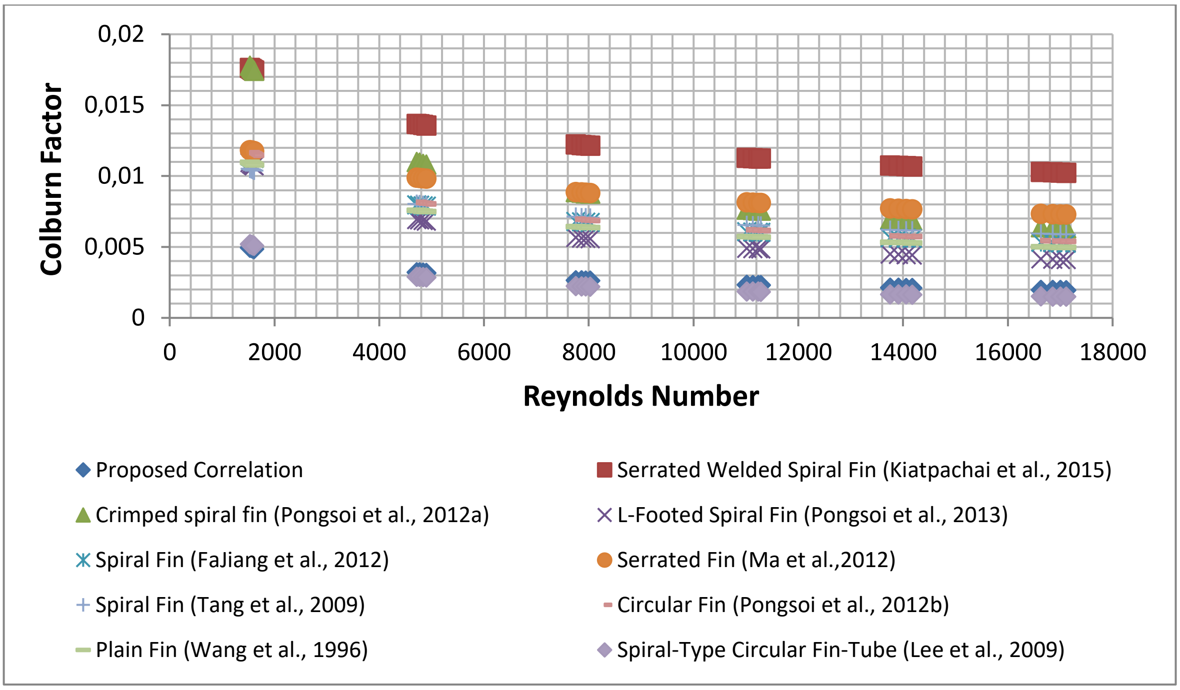


(b)

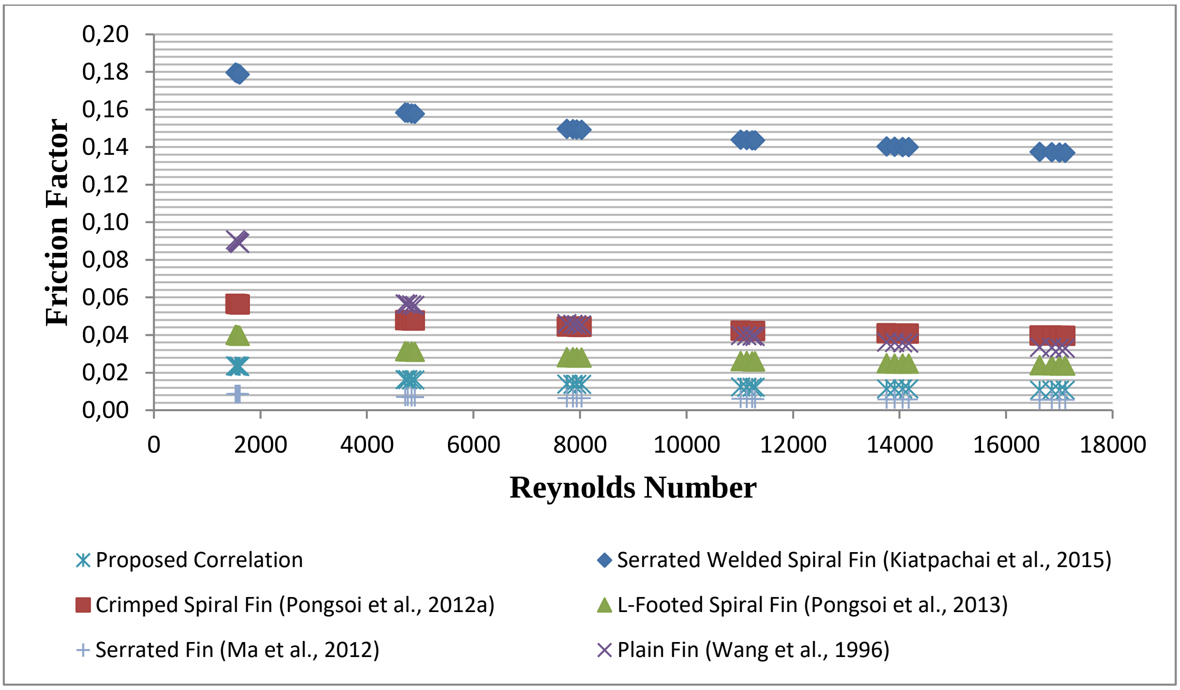
Figure 6. Comparison of the proposed correlations with the experimental data a) Colburn factor b) Fanning friction factor

In Figure 6, the comparison between the proposed correlations with the experimental data is seen. Mean deviation values of the proposed correlations for Colburn *j* factor and Fanning friction factor *f* were found as 5.58% and 6.53%, respectively. Comparison between the proposed correlations for Colburn *j* factor and Fanning friction factor *f* in the literature and the formula developed through the usage of experimental data can be seen in Figure 7. The formula suggested for both Colburn *j* factor and Fanning friction factor *f* at high Reynolds Numbers and the proposed correlations for other fin types present similar changes. It was found that serrated welded spiral fin-tube banks gave a higher Colburn *j* factor and *f*-friction factor than other fin types. But, it will cause higher pressure drop.

It was found that serrated welded spiral fin gave a higher (77.9%) Colburn *j* factor than the L-footed spiral fin, but the L-footed spiral fin gave a lower (90.5%) *f*-friction factor than the serrated welded spiral fin.



(a)



(b)

Figure 7. Comparison of the proposed correlations with the correlations of several fin types a) Colburn factor b) Fanning friction factor

5.Conclusions

In this study, the heat transfer and flow characteristics of the air-side of the L-foot spiral fin-and-tube heat exchanger were analyzed at high Reynolds Number (1500-17500) determined according to the outside diameter of the fin collar. Through the experimental data, mean deviation values of the formulas derived for Colburn *j* factor and Fanning friction factor *f* were found as 5.58% and 6.53%, respectively.

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