Heat Transfer Coefficients for Particulate Airflow in Shell and Coiled Tube Heat Exchangers

W. Witchayanuwat and S. Kheawhom

Abstract—In this work, we experimentally study heat transfer from exhaust particulate air of detergent spray drying tower to water by using coiled tube heat exchanger. Water flows in the coiled tubes, where air loaded with detergent particles of 43 micrometers in diameter flows within the shell. Four coiled tubes with different coil pitches are used in a counter-current flow configuration. We investigate heat transfer coefficients of inside and outside the heat transfer surfaces through 400 experiments. The correlations between Nusselt number and Reynolds number, Prandtl number, mass flow rate of particulates to mass flow rate of air ratio and coiled tube pitch parameter are proposed. The correlations procured can be used to predicted heat transfer between tube and shell of the heat exchanger.

Keywords—Shell and coiled tube heat exchanger, Spray drying tower, Heat transfer coefficients.

I. INTRODUCTION

HEAT transfer between two fluids that are at different temperatures and separated by a solid wall occurs in various engineering applications. The device used to implement this exchange is termed a heat exchanger, and specific applications may be found in waste heat recovery and chemical processing [1].

The heat transfer enhancement can ameliorate the performance and simultaneously decrease the size of the heat exchanger. Generally, the enhancement techniques can be divided into two categories including active and passive techniques. The active techniques require external forces, e.g., electric field, acoustic and surface vibration. The passive techniques require special surface geometries or fluid additives. Curved tubes have been introduced as one of the passive heat transfer enhancement techniques and are widely used in various industrial applications because of their compact structure and high heat transfer coefficient [2]. Coiled tube is one of curved tubes that have been used in a wide variety of application. One application contemplated in this work is the heat recovery from exhaust air of detergent spray drying tower.

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Normally, the coiled tube heat exchanger used in the heat recovery system encounters deposition of detergent particles on the heat transfer surfaces. Thus, performance of the heat exchanger deteriorates with time [3]. Prabhanjan et al. [4] studied and compared the heat transfer rate between a straight tube heat exchanger and a helically coiled heat exchanger. They reported that use of a helical coil heat exchanger could increase the heat transfer coefficient compared to a similarly dimensioned straight tube heat exchanger. Nuntaphan et al. [3] investigated the thermal resistance due to fly-ash deposition on the heat transfer surface. It was found that the thermal resistance is directly proportional to the dust-air ratio. Salimpour [5] studied heat transfer coefficients for water-water heat transfer in shell and coiled tube heat exchanger. The correlations between Nusselt number and Reynolds number, Prandtl number and coiled tube pitch parameter were proposed.

However, no work has been done on the particulate air-water system. Thus, in this work, we study the characteristics of particulate air and water heat transfer in the shell and coiled tube heat exchanger to determine the correlations between various important parameters.

II. EXPERIMENTAL SYSTEM

The schematic diagram of the experimental system is shown in Fig. 1. Experiments were performed by measuring the heat transfer between water stream flowing inside the tube-side and hot particulate air stream flowing in the shell-side. The experiments were carried out in a counter-current mode operation. The main parts of the experiment unit are coiled tube heat exchanger, blower, electric heater, particles feeder, cyclone, pump, and data logger.

The heat exchanger includes a copper coiled tube and an insulated shell. Blower feeds ambient air through an electric heater to generate hot air stream, where the temperature of hot air stream is under controlled. Detergent particles are then fed to the hot air stream by a particle feeder. The particulate hot air stream generated is similar to that leaving a detergent spray drying tower. The particulate hot air flows through the tube bank and exchanges heat with cold water circulated inside the coiled tube. In this work, the mass flow rate and the inter temperature of the cold water were kept constant at 0.02 kg/s and room temperature, respectively.

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The ranges of operating parameters are given in Table I. The pitches of coiled tube at 4.43 cm, 6.31 cm, 8.28 cm, and 10.41 cm used in this study are shown in Fig. 2.

The average diameter of detergent particles is 43 micrometers. The dimensions of the heat exchangers are depicted in Table II.

Fig. 1 Schematic of experimental system

Fig. 2 Pitches of the coiled tube
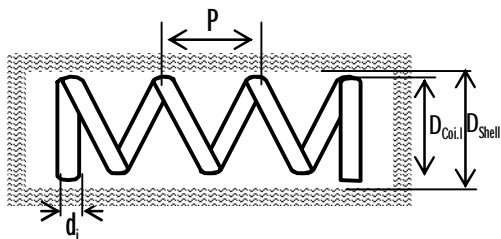
Fig. 3 Coiled tube heat exchanger

TABLE I
OPERATING PARAMETERS

Parameters	Unit	Value
Mass flow rate of water	kg/s	0.02
Inlet temperature of water	°c	room temp.
Mass flow rate of air	kg/s	0.3-0.6
Inlet temperature of air	°c	70 - 100
Mass flow rate of particles	g/s	0.5 - 1.0

TABLE II
DIMENSIONS OF THE HEAT EXCHANGER

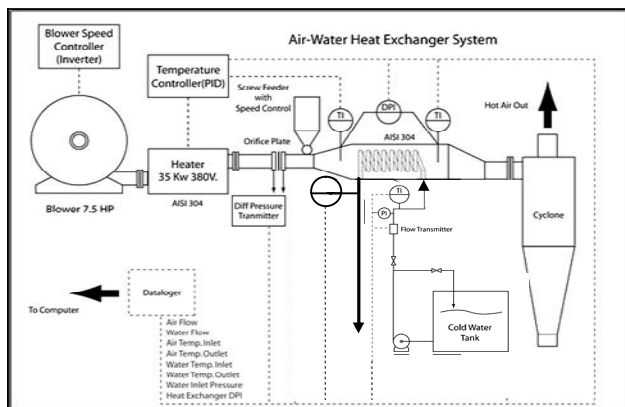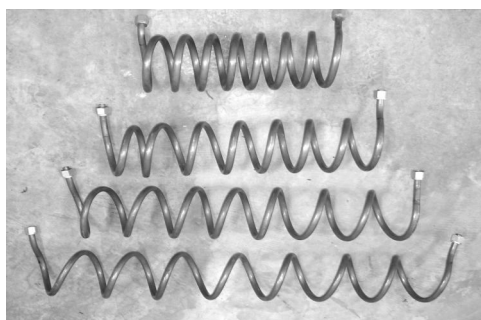| Parameters	Unit	Value
Size of copper tube	inch.	0.5
Inner diameter of shell	cm	20
Diameter of coiled tube	cm	10
Heat exchanger length	cm	150

III. GEOMETRY OF COILED TUBE HEAT EXCHANGER

A typical shell and coiled tube heat exchanger is show in Fig. 3. Where, D_{Shell} is the inner diameter of the shell, D_{Coil} is the diameter of the coil, d_i and d_o are the inner and outer diameter of the tube, respectively. P is the coil pitch. The dimensionless pitch parameter, φ, is defined as $\varphi = P / \pi D_{Coil}$.

Shell-side: Reynolds number (Re_o), Prandtl number (Pr_o), Nusselt number (Nu_o), and mass flow rate of particles to mass flow rate of air ratio (β) are defined as

$$\mathrm{Re}_o = \frac{\rho_o \upsilon_o D_h}{\mu_o}, \; \mathrm{Pr}_o = \frac{C_{p,o}\mu_o}{k_o}, \; Nu_o = \frac{h_o D_h}{k_o},$$

$$\text{and} \quad \beta = \frac{m_{particles}}{m_h}.$$

where, h_o, k_o, and $\quad D_h = \dfrac{D_{Shell} - \pi D_{Coil} d_o^2 \varphi^{-1}}{D_{Shell} - \pi D_{Coil} d_o \varphi^{-1}} \quad$ are

convective heat transfer coefficient of outside, thermal conductivity of outside the heat transfer surfaces, and hydraulic diameter of shell-side, respectively.

Tube-side: Reynolds number (Re_i), Prandtl number (Pr_i), Nusselt number (Nu_i), are defined as

$$\mathrm{Re}_i = \frac{\rho_i \upsilon_i d_i}{\mu_i}, \; \mathrm{Pr}_i = \frac{C_{p,i}\mu_i}{k_i}, \text{ and } \; Nu_i = \frac{h_i d_i}{k_i}.$$

where, h_i and k_i are convective heat transfer coefficient of inside, thermal conductivity of inside the heat transfer surfaces.

IV. ANALYSIS FOR DATA CORRELATION

The heat transfer rate of a heat exchanger ($\dot{Q}_c$) can be calculated from the water side as

$$\dot{Q}_c = \dot{m}_c \, C_{p,c}(T_{c,o} - T_{c,i}) \qquad (1)$$

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where, $\dot{m}_c$ is the mass flow rate of water, $T_{c,i}$ and $T_{c,o}$ are, respectively, the inlet and outlet temperature of water and $c_{p,c}$ is the specific heat of water.

The heat transfer rate of a heat exchanger ($\dot{Q}_h$) can be calculated from the hot air side as

$$\dot{Q}_h = \dot{m}_h\, C_{p,h}(T_{h,o} - T_{h,i}) \tag{2}$$

where, $\dot{m}_h$ is the mass flow rate of hot air, $T_{h,i}$ and $T_{h,o}$ are, respectively, the inlet and outlet temperature of hot air and $c_{p,h}$ is the specific heat of hot air.

The average heat transfer rate of a heat exchanger ($\dot{Q}_{avg}$) is defined as

$$\dot{Q}_{avg} = \frac{\left|\dot{Q}_h\right| + \left|\dot{Q}_c\right|}{2} \tag{3}$$

The corresponding overall heat transfer coefficient (U_o) is defined as

$$\dot{Q}_{avg} = U_o A_o \Delta T_{lm} \tag{4}$$

Then the overall heat transfer resistance can be expressed by

$$\frac{1}{U_o A_o} = \frac{\Delta T_{lm}}{\dot{Q}_{avg}} \tag{5}$$

where, A_o is the total heat transfer area, and ΔT_{lm} is the log mean temperature difference (LMTD) defined as

$$\Delta T_{lm} = \frac{\Delta T_a - \Delta T_b}{\ln\left(\Delta T_a / \Delta T_b\right)} \tag{6}$$

where, $\Delta T_a = T_{h,i} - T_{c,o}$ and $\Delta T_b = T_{h,o} - T_{c,i}$

with subscript "o" and "i" denoting outer and inner surface of coiled tube, respectively.

The overall thermal resistance consists of three parts in series: the convective resistance on inner surface, the coiled tube wall conduction resistance and the convective resistance on outer surface of the coiled tube. That is

$$\frac{1}{U_o A_o} = \frac{1}{h_o A_o} + \frac{\ln\left(d_o / d_i\right)}{2\pi k L} + \frac{1}{h_i A_i} \tag{7}$$

where, d_o is the outer diameter of the tube, d_i is the inner diameter of the tube, k is the thermal conductivity of the wall, and L is the length of the tube.

The inner surface heat transfer coefficient (h_i) is cast into the form

$$h_i = C_i\, \mathrm{Re}_i^{m_i}\, \mathrm{Pr}_i^{n_i}\, \varphi_i^{a_i}\, \frac{k_i}{d_i} \tag{8}$$

where, C_i, m_i, n_i and a_i are constants to be determined.

Equation (8) can be rearranged into the form of Nusselt number as in equation (9).

$$Nu_i = C_i\, \mathrm{Re}_i^{m_i}\, \mathrm{Pr}_i^{n_i}\, \varphi_i^{a_i} \tag{9}$$

or

$$Nu_i = \frac{h_i d_i}{k_i} \tag{10}$$

The outer surface heat transfer coefficient (h_o) can be expressed accordingly as

$$h_o = C_o\, \mathrm{Re}_o^{m_o}\, \mathrm{Pr}_o^{n_o}\, \varphi_o^{a_o}\, \beta^b\, \frac{k_o}{D_h} \tag{11}$$

where, C_o, m_o, n_o, a_o and b are constants to be determined.

Equation (11) can be rewritten into the form of Nusselt number as in equation (12).

$$Nu_o = C_o\, \mathrm{Re}_o^{m_o}\, \mathrm{Pr}_o^{n_o}\, \varphi_o^{a_o}\, \beta^b \tag{12}$$

or

$$Nu_o = \frac{h_o D_h}{k_o} \tag{13}$$

By substituting equations (8) and (11) into equation (7), equation (14) is obtained.

$$\frac{1}{U_o A_o} = \frac{1}{A_o\left(C_o\, \mathrm{Re}_o^{m_o}\, \mathrm{Pr}_o^{n_o}\, \varphi^{a_o}\, \beta^b\left(k_o / D_h\right)\right)} + \frac{\ln\left(d_o / d_i\right)}{2\pi k L}$$
$$+ \frac{1}{A_i\left(C_i\, \mathrm{Re}_i^{m_i}\, \mathrm{Pr}_i^{n_i}\, \varphi^{a_i}\left(k_i / d_i\right)\right)} \tag{14}$$

The value of $1/U_o A_o$ is evaluated by using equation (5) and C_i, C_o, m_i, m_o, n_i, n_o, a_i, a_o and b in equation (14) are

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constants to be determined by experimental data. Based on experimental results the correlations are obtained using least square analysis as

$$E\left(C_o, C_i, m_i, m_o, n_i, n_o, a_i, a_o, b\right) =$$

$$\frac{1}{NP}\sum_{j=1}^{NP}\left[\left(\left(\frac{1}{U_o A_o}\right)_j - \left(\frac{1}{A_o\left(C_o \operatorname{Re}_o^{m_o} \operatorname{Pr}_o^{n_o} \varphi^{a_o} \beta^b \left(k_o / D_h\right)\right)} + \frac{\ln\left(d_o / d_i\right)}{2\pi kL} + \frac{1}{A_i\left(C_i \operatorname{Re}_i^{m_i} \operatorname{Pr}_i^{n_i} \varphi^{a_i}\left(k_i / d_i\right)\right)}\right)_j\right)\right]^2$$

(15)

V. RESULTS AND DISCUSSIONS

The tests were performed for all four coiled tubes in total 400 test runs with operating parameters given in Table I. The correlations between the tube-side Nusselt number and Reynolds number, Prandtl number, and dimensionless of coiled tube pitch parameter are obtained as

$$Nu_i = 0.134\,\operatorname{Re}_i^{0.440}\,\operatorname{Pr}_i^{0.097}\,\varphi_i^{-0.181} \tag{16}$$

for $3,500 \le \operatorname{Re}_i \le 4,100$.

Equation (17) shows the correlation between the shell-side Nusselt number and Reynolds number, Prandtl number, mass flow rate of particulates to mass flow rate of air ratio and dimensionless of coiled tube pitch parameter.

$$Nu_o = 0.608\,\operatorname{Re}_o^{3.083}\,\operatorname{Pr}_o^{3.949}\,\varphi_o^{0.029}\,\beta^{-0.217} \tag{17}$$

for $89,000 \le \operatorname{Re}_o \le 220,000$.

Fig. 5 Comparison of experimental data with present data of 6.31 cm pitches coil

Fig. 6 Comparison of experimental data with present data of 8.28 cm pitches coil
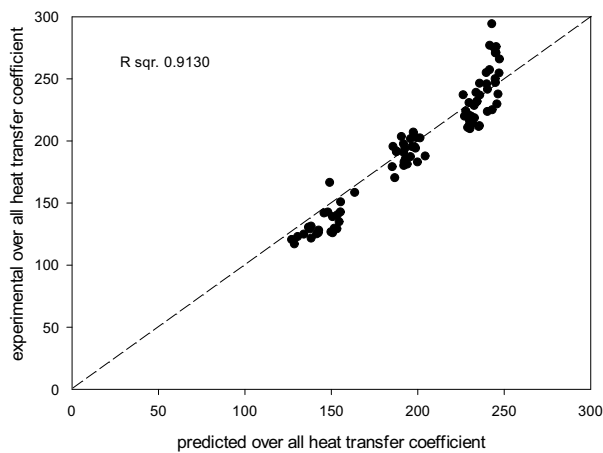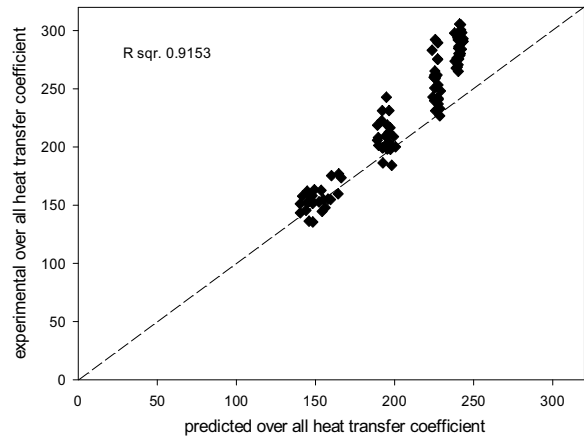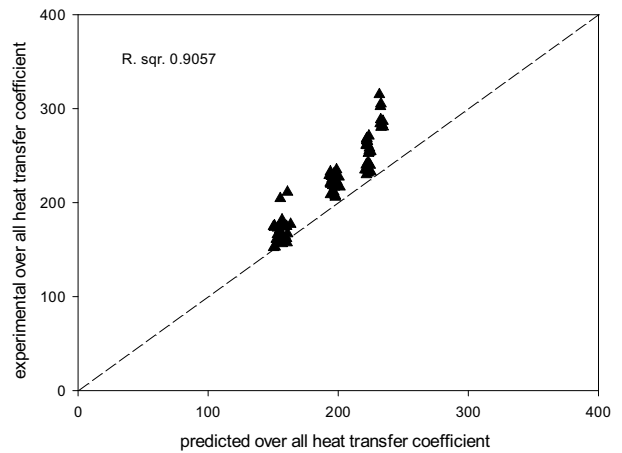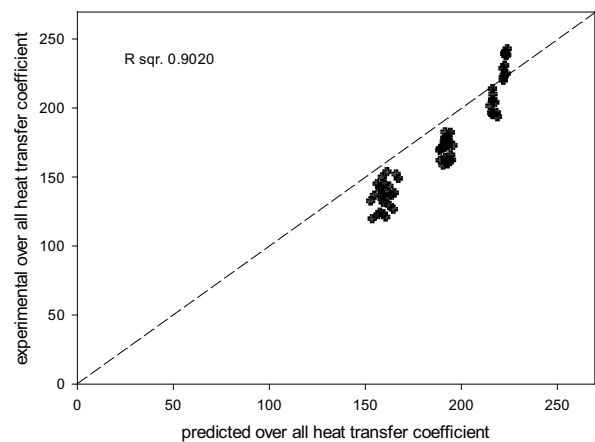

Fig. 4 Comparison of experimental data with present data of 4.43 cm pitches coil

Fig. 7 Comparison of experimental data with present data of 10.41 cm pitches coil

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Figures 4-7 show the comparison between the predicted overall heat transfer coefficients and the experimental overall heat transfer coefficients of 4.43 cm, 6.31 cm, 8.28 cm and 10.41 cm coiled tube pitches, respectively. It was found that the proposed correlations are in good agreement with the present data.
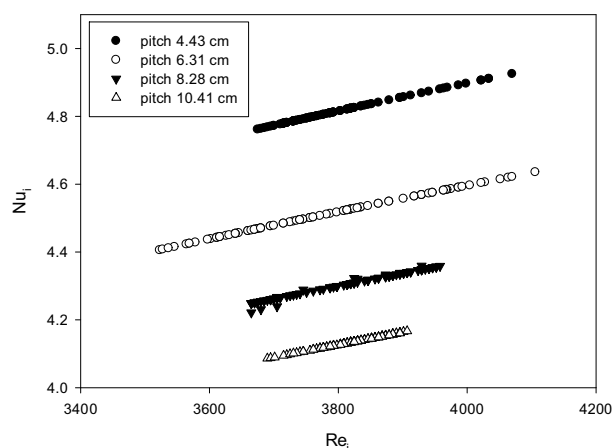

Fig. 8   Inside Nusselt number vs. inside Reynolds number various coiled tube pitch

The relation between inside Nusselt number and inside Reynolds number of each coiled tube pitch is shown in Fig. 8. The increasing of the coiled tube pitch decreases the inside Nusselt number. This matter can be justified as the higher values correspond to loose-coiling conditions. Therefore, the heat transfer coefficients of these coiled tubes are close to that of a similarly dimensioned straight tube heat exchanger.
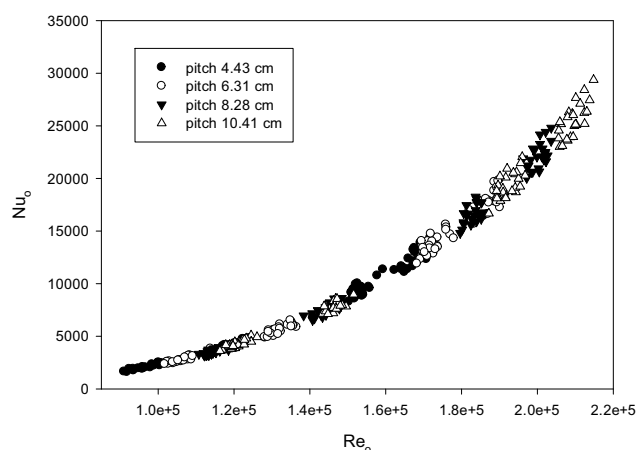

Fig. 9   Outside Nusselt number vs. outside Reynolds number various coiled tube pitch.

Figure 9 presents the variation of outside Nusselt number with outside Reynolds number calculated based on hydraulic diameter of the shell. It was found that the increasing of the outside Nusselt number also increases the outside Reynolds number. Although, it is observed that coil pitches of heat exchanger slightly affect to the outside Nusselt number, because air flow in shell-side is in high Reynolds number regime.

From experimental data, it appears that the increasing of the mass flow rate of particles leads to lower values of shell-side Nusselt number. This may be explained as in higher mass flow rate of particles, the thermal resistance on heat transfer surfaces will increase.

VI. CONCLUSION

In this work, heat transfer from exhaust particulate air of detergent spray drying tower to water by using coiled tube heat exchanger is investigated experimentally. From the results of the present study, it was found that the tube-side heat transfer coefficients of the coils with smaller pitches are higher than those of larger pitches but the pitches of coiled tubes slightly affect the shell-side heat transfer coefficients. The experiments data were used to determine Nusselt number of inside and outside of the coiled tube heat exchanger. Finally, two empirical correlations were developed to predict the inside and outside heat transfer coefficients of the coiled tube heat exchanger for the particulate airflow-water system.

REFERENCES

[1]   Su Thet Mon Than, Khin Aung Lin, Mi Sandar Mon, 2008, Heat exchanger design, Proceedings of world academy of science, Engineering and technology, 36: 619-626.
[2]   P.Naphon, 2006, A review of flow and heat transfer characteristics in curved tubes, Renewable and sustainable energy reviews, 10: 463-490.
[3]   A.nuntaphan, T.Kiatsiriroat, 2007, Thermal behavior of spiral fin and tube heat exchanger having fly ash deposit, Experimental thermal and fluid science, 31: 1103-1109.
[4]   D.G. Prabhanjan, G.S.V. Raghavan, T.J. Rennie, 2002, Comparison of heat transfer rates between a straight tube heat exchanger and a helically coiled heat exchanger, International communications of heat and mass transfer, 29: 185-191.
[5]   M.R. Salimpour, 2008, Heat transfer coefficient of shell and coiled tube heat exchangers, Experimental thermal and fluid science.