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Analysis of fouling characteristic in enhanced tubes using multiple heat and mass transfer analogies☆

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A B S T R A C T

This paper provides a comprehensive analysis on cooling tower fouling data taken from seven 15.54 mm I.D. helically ribbed, copper tubes and a plain tube at $Re = 16000$. There are two key processes during fouling formation: fouling deposition and fouling removal, which can be determined by mass transfer and fluid friction respectively. The mass transfer coefficient can be calculated through three analogies: Prandtl analogy, Von–Karman analogy, and Chilton–Colburn analogy. Based on our analyses, Von–Karman analogy is the optimized analogy, which can well predict the formation of cooling tower fouling. Series of semi-theoretical fouling correlations as a function of the product of area indexes and efficiency indexes were developed, which can be applicable to different internally ribbed geometries. The correlations can be directly used to assess the fouling potential of enhanced tubes in actual cooling tower water situations.

1. Introduction

Fouling is the solid substance deposited on heat transfer surfaces contacting with unclean fluid, which can significantly reduce the heat transfer efficiency and therefore has been attracted many research interests. There are three issues to be addressed in fouling research [1]. The first one is the theoretical and experimental analysis of fouling formation, providing a common and precise predicting model for heat exchanger design. The second is fouling monitor techniques. The last one is the countermeasure of fouling. Many researches had been focused on the theoretical and experimental analysis on fouling mechanisms, which are crucial to the control of fouling formation.

With regard to modeling the fouling data, Kern and Seaton [2] published an analysis which is considered as a landmark in the beginning of modern scientific study of fouling. They assumed two key processes that control the fouling resistance, namely fouling deposition and fouling removal. In principle, they brought forward a model which can embrace fouling involving all forms of growth and removal processes [3]. Several significant investigations addressed the foulant deposition and removal rates in enhanced tubes. Kim and Webb [4] made a detailed research on three internal repeated rib tubes. Heat-mass transfer analogy was utilized to calculate the mass transfer coefficient (K_m) in the diffusion region. Within the test range, the fouling resistance increased as geometric parameter e/D decreased and p/e increased. Li and Webb [5] developed the first long-term fouling model for practical cooling tower water flowing inside the enhanced tubes by using the Chilton–Colburn analogy to calculate mass transfer coefficient. Recently, Jun and Puri [6] applied a 2D fouling model to predict the fouling performance of plate heat exchangers, which coupled a 2D dynamic model with material balance equations. The model showed a strong relation between fouling and the operating conditions, *i.e.* flow rate and deposit formation. Their model can determine the sensor locations that can provide necessary information for fouling formation to monitor not only fouling during thermal process but also cleaning condition. Quan *et al.* [7] conducted an experimental study to investigate the fouling process of calcium carbonate on enhanced heat transfer surface in forced convective heat transfer. The fouling behaviors were examined under different factors including fluid velocity, hardness, alkalinity, solution temperature, and wall temperature. Webb [8] reported heat transfer and friction characteristics of three tubes having a conical, three-dimensional roughness on the inner tube surface with water flow in the tube. The Nusselt number (Nu) of 3-D TC3 truncated cone tube was 3.74 times higher than a plain tube. Accelerated particular fouling data were also provided for this tube and compared to the results of helical-ribbed tubes studied in Li [9]. The results showed

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that the 3-D tube provided the highest heat transfer coefficient but also had the highest asymptotic fouling resistance.

Webb and Li [10] described the experimental results of long-term fouling tests for cooling tower water flowing inside enhanced tubes. Fouling data were measured for the seven helical rib roughened tubes for more than a 2500 h operating period. There exist combined precipitation and particulate fouling (PPF) in cooling tower systems. The test geometries include seven enhanced copper tubes and a plain copper tube. In this study, the main focus is to analyze the experimental results in Ref. [10,11] to quantitatively define the effect of rib height, rib axial pitch, and helix angle on the tube fouling performance.

No analysis has been done to establish the fouling model that can fully reflect the realistic flow conditions. From the previous studies, fouling deposition rate determined by mass transfer coefficient is the key parameter to estimate the fouling resistance. In this study, new models for calculating the deposition rate are developed. Comprehensive analyses about three widely used analogies, *i.e.* the Prandtl analogy, the Von–Karman analogy, and the Chilton–Colburn analogy, are discussed to develop the mass transfer coefficient K_m. Semi-theoretical fouling models based on three analogies have been established to correlate experimental data on fouling. It is revealed that the Von–Karman analogy is the most appropriate for providing the most accurate fouling model to represent the experimental results.

2. Fouling Model

There are four different patterns for fouling formation: the linear increasing pattern, the linear decreasing pattern, the power pattern, and the asymptotic pattern [12]. In this work, the fouling pattern in cooling tower systems is considered to follow the asymptotic pattern:

$$R_f = R_f^*\left(1-e^{-Bt}\right),\quad R_f^* = \frac{K_m P C_b \xi}{\tau_s k_f \rho_f},\quad B = \frac{\tau_s}{\xi}.$$

The asymptotic fouling resistance ratio R_f^*/R_{fp}^* is utilized to obtain the fouling characteristics in the enhanced tubes. For rough surfaces, a fraction of the pressure drop may be due to the profile drag on roughness elements. Webb *et al.* [13] reported the j and f factors for the clean tube, which can be used for characterize the heat transfer and flow performance in the helical-ribbed tubes. They showed that $f \propto Re^{-0.283}$, and the results indicated that the pressure drop was dominated by wall shear stress, and that the pressure drag component was negligible. In this study, the profile drag can be ignored, comparing with the wall shear stress, which can be calculated with $\tau_s = \Delta P A_c/A_w = 0.5 f\rho u^2$. We define $\sigma = (P/P_p)(\xi/\xi_p)$, then the following equation can be obtained:

$$R_f^*/R_{fp}^* = \sigma\left(K_m/K_{mp}\right)/\left(\tau_s/\tau_{sp}\right) \tag{1}$$

where σ is a fouling process index, which is determined from the fouling tests. The asymptotic fouling resistance ratio R_f^*/R_{fp}^* can be obtained after calculating K_m and τ_s.

2.1. Calculation of K_m [14,15]

Three analogies are used to calculate the mass transfer coefficient, however, the Reynolds analogy is not included since it is assumed that $Pr \approx 1$ and $Sc \approx 1$. It is not suitable for the experimental data because of the wide range in the operating conditions.

Prandtl [16] assumed the turbulent boundary can be divided into laminar region and turbulent region, which can amend the Reynolds analogy. The momentum and heat transfer are attributed to molecule transfer, and the thickness of the laminar region is thin. Thus, the temperature and velocity are distributed linearly. Prandtl analogy divides the boundary into laminar region and turbulent region and ignores the influence of the buffer region. This brings the deviation. The equations of Prandtl analogy are listed below:

$$St = j \times Pr^{-2/3} = (f/2)/\left[1+(f/2)^{0.5}\times 5(Pr-1)\right] \tag{2a}$$

$$St' = K_m/u_0 = (f/2)/\left[1+(f/2)^{0.5}\times 5(Sc-1)\right]. \tag{2b}$$

Compared to the Reynolds analogy and the Prandtl analogy, the Von Karman analogy [17] includes buffer region into the model. The usage of the three-region model makes the turbulent heat and mass transfer theory more close to the realistic flow conditions. Fig. 1 shows the temperature distribution and velocity profile of the turbulent flow inside a tube. The deviation of the Von–Karman analogy is primarily caused by neglecting turbulent transfer in laminar region and heat transfer in turbulent region. There are large deviations when Pr number is extra large or small. The equations of the Von–Karman analogy are listed below:

$$\begin{aligned} St &= j \times Pr^{-2/3} \\ &= (f/2)/\left\{1+(f/2)^{0.5}\times[5(Pr-1)+5\ln((5Pr+1)/6)]\right\} \end{aligned} \tag{3a}$$

$$\begin{aligned} St' &= K_m/u_0 \\ &= (f/2)/\left\{1+(f/2)^{0.5}\times[5(Sc-1)+5\ln((5Sc+1)/6)]\right\}. \end{aligned} \tag{3b}$$

Colburn [18] modified the Reynolds analogy based on their experimental data. Their modification make the new analogy not restrict by $Pr \approx 1$ and $Sc \approx 1$, and it is extended to systems having fluid friction. Chilton–Colburn analogy presents:

$$(K_m/u_0)Sc^{2/3} = (h/\rho c_p u_0)Pr^{2/3} \tag{4}$$

The correlations of mass transfer coefficient K_m are obtained through eliminating f factor in above equations:

For Prandtl analogy,

$$K_m/u_0 = 4/\left[b_1^2 m_1^2 + \left(2a_1b_1 - 2b_1^2\right)m + \left(b_1^2 - 2a_1b_1\right)\right] \tag{5}$$

where $m_1 = [1 + 4/(b_1^2 St)]^{0.5}$, $a_1 = 5(Sc - 1)$, $b_1 = 5(Pr - 1)$.

For the Von-Karman analogy,

$$K_m/u_0 = 4/\left[b_2^2 m_2^2 + \left(2a_2b_2 - 2b_2^2\right)m + \left(b_2^2 - 2a_2b_2\right)\right] \tag{6}$$

where $m_2 = [1 + 4/(b_2^2 St)]^{0.5}$, $a_2 = 5(Sc - 1) + 5\ln[(5Sc + 1)/6]$, $b_2 = 5(Pr - 1) + 5\ln[(5Pr + 1)/6]$.

For the Chilton–Colburn analogy,

$$K_m/u_0 = j \times Pr^{2/3} \times Sc^{-2/3}. \tag{7}$$

2.2. Correlation of f and j factors

Webb *et al.* [13] developed the following heat transfer and friction factor equations for the seven tubes tested in their study:

$$f = 0.108Re^{-0.283} n_s^{0.221} (e/D_i)^{0.785} \alpha^{0.78} \tag{8}$$

$$j = 0.00933Re^{-0.181} n_s^{0.285} (e/D_i)^{0.323} \alpha^{0.505}. \tag{9}$$

Average deviation of friction and heat transfer correlations is 4.9% and 3.8%, respectively. For smooth tube, the Seider–Tate correlations $f_p = 0.079Re^{-0.25}$ and $j_p = 0.027Re^{-0.2}$ are used to calculate f and j factors for smooth tubes [5].