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Analysis of heat-transfer performance of cross-flow fin-tube heat exchangers under dry and wet conditions

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ABSTRACT

A three-dimensional analysis procedure for the detailed phenomenon in a fin-tube heat exchanger has been developed and applied to predict the heat/mass transfer characteristics of the wave-fin heat exchangers. The continuity, Navier–Stokes and energy equations together with the species equation for the air–vapor mixture are solved in a coupled manner, so that the inter-dependence between the temperature and the humidity can be properly taken into account, by using the SIMPLE-type finite volume method. Having validated the procedure, calculations have been carried out for various frontal-velocity and inlet-humidity conditions. It has been shown that the flow characteristics, such as the temperature and humidity fields, along with the local heat flux and the condensation rate, can be successfully captured. The numerical results reveal that the existing correlations considerably underestimate the fin efficiency especially for multi-row heat exchangers. For dehumidifying cases, the sensible heat-transfer rate seems insensitive to the inlet-humidity change. The rate changes mostly in the narrow band of partially wet regime between 25% and 40% of inlet relative humidity. The range of the frontal velocity that gives the best performance for various numbers of rows is also estimated. The analogy between the heat and mass transfer on the fin surface is also examined.

1. Introduction

It is of practical importance to determine the fin efficiency accurately in estimating the overall heat transfer of a heat exchanger. The solution generally is not known for various fin types and configurations, and numerous mathematical approximations have been suggested in the literature. Most such studies, however, are based on one-dimensional analysis together with the assumption that the fin-to-air heat-transfer coefficient is uniform. Improved methods have been suggested [1–7] by taking the variations of the heat-transfer coefficient or the air temperature into consideration by approximating them as functions of the distance from the tube. Haung and Shah [2] carried out comparative study of these methods for a simple fin shape, but no clear conclusion as to which is most accurate. Besides, all these approaches show limited success as the heat-transfer coefficient is substantially larger near the leading edge of the fin compared to the rest of the region and make it impossible to model by simple analytic functions.

Estimation of heat transfer under dehumidifying condition is even more complex. Various effects of design parameters have been explored. To tackle the problems of practical interest such as optimization of fin pitch, wave depth, positions of slits or louvers, etc., however, one needs to have detailed local information which normally is beyond the reach of experimental study. CFD techniques may provide the accurate and detailed heat-transfer characteristics of heat exchangers. Tsai et al. [8], Perotin and Clodic [9] and Tao et al. [10] carried out three-dimensional conjugate heat-transfer analyses for dry condition and obtained the local heat-transfer characteristics and/or the heat-transfer enhancement for various fin types. Few in the literature, however, provide a methodology that takes the water condensation/evaporation into account. The rate of mass transfer may be estimated by the Chilton–Colburn [11] analogy assuming that the mass transfer can be decoupled from the heat transfer. Although this gives a straight-forward way of predicting the mass-transfer rate from the known heat-transfer performance, it may not be appropriate as the condensation and the temperature are interdependent as seen in the psychrometric chart. Recently, Comini et al. [12] developed an analysis procedure, in which the cooling air is treated as two-component air–vapor mixture, and successfully estimated the temperature and condensate distributions, and the heat-transfer coefficient for rectangular-finned heat exchangers.

The objective of this study is to develop a fully coupled three-dimensional heat/mass-transfer analysis procedure under dry and wet conditions. The procedure is then applied to analyze the detailed heat/mass transfer characteristics of a wave fin of complex shape and the multi-row heat-exchanger performance.

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Nomenclature

A	heat exchange area [m^2]
A_a	total surface area [m^2]
A_{fr}	frontal area [m^2]
A_{min}	minimum free flow area [m^2]
C_p	specific heat [J/kg °C]
D	diffusion coefficient [m^2/s]
d_h	hydraulic diameter [m]
d_o	outer tube diameter [m]
h_a, h_s	air-side and sensible heat-transfer coefficients [W/m^2 K]
h_m	mass-transfer coefficient
i_a	enthalpy of humid air [J/kg]
i_r	enthalpy of saturated water vapor at mean air temperature [J/kg]
k	thermal conductivity [W/m K]
L	length of the fin [m]
N	number of rows of heat exchanger
p	pressure [Pa]
P_T, P_L	transversal and longitudinal tube pitch [m], (see Fig. 1)
q_{lm}	latent heat of water [J/kg]
q''	heat flux [W/m^2]
Q	overall heat-transfer rate [W]
Q_n	heat-transfer rate of nth row [W]
Q_{max}	maximum heat-transfer rate [W]: $m_a(i_{a,in} - i_{r,in})$
R_v	gas constant of air [J/kg K]
Re	Reynolds number
T	temperature [°C; K]
t	fin thickness [m]
u_i	flow velocity in x_i direction [m/s]
v_{fr}	frontal velocity [m/s]
w	humidity [kg/kg air]
Greek symbols	
η_f	fin efficiency
μ	viscosity [N s/m^2]
ρ	density [kg/m^3]
σ	standard deviation
Subscripts	
a	air
b	fin base
f	fin
s	saturated state
t	tube
w	fin surface

These numerical results may help evaluate the commonly invoked data reduction practices: The fin efficiency is estimated and compared with existing correlations based on various approximations. The effects of inlet relative humidity, and the heat and mass transfer analogy are also examined and/or assessed.

2. Equations and solution procedure

The humid air can be taken as a mixture of air and water vapor. Since the mass fraction of the water vapor is very low, it has little effect on the air flow. Taking air as the carrier fluid and water vapor as a dilute species, the laminar humid air flow can be described by the equations of continuity, momentum, energy, and species transport. The effects of water film on the overall heat transfer or the thermal resistance for the condensate of the droplet shape are known to be very small and thus are neglected in the present analysis.

Introducing the hydraulic diameter d_h and the characteristic velocity u_{max} as the characteristic length and velocity scales:

$$d_h = \frac{4A_{min}L}{A_a} \tag{1}$$

$$u_{max} = \frac{A_{fr}}{A_{min}} v_{fr} \tag{2}$$

with

$$A_{min} = Min(A_1, 2A_2) \tag{3}$$

where A_1 and A_2 are the areas between two adjacent tubes shown in Fig. 1 and v_{fr} the frontal velocity, the dimensionless governing equations can be written as

Continuity:

$$\frac{\partial(\hat{\rho}\hat{u}_j)}{\partial \hat{x}_j} = 0 \tag{4}$$

Momentum:

$$\frac{\partial}{\partial \hat{x}_j}(\hat{u}_i\hat{u}_j) = -\frac{\partial \hat{p}}{\partial \hat{x}_i} + \frac{1}{\mathrm{Re}}\frac{\partial^2 \hat{u}_i}{\partial \hat{x}_j \partial \hat{x}_j} \tag{5}$$

Energy:

$$\frac{\partial}{\partial \hat{x}_j}(\hat{u}_j\hat{T}) = \frac{1}{\mathrm{Re}\cdot\mathrm{Pr}}\frac{\partial^2 \hat{T}}{\partial \hat{x}_j \partial \hat{x}_j} \tag{6}$$

Species:

$$\frac{\partial}{\partial \hat{x}_j}(\hat{u}_j w) = \hat{D}\frac{\partial^2 w}{\partial \hat{x}_j \partial \hat{x}_j} \tag{7}$$

where the carets denote dimensionless variables and

$$\hat{\rho} = \frac{\rho}{\rho_{in}}, \quad \hat{p} = \frac{p}{\rho_{in}u_{max}^2}, \quad \hat{T} = \frac{T - T_t}{T_{in} - T_t}, \quad \mathrm{Re} = \frac{\rho_{in}u_{max}d_h}{\mu}, \quad \mathrm{Pr} = \frac{\mu C_p}{k}, \quad \hat{D} = \frac{D}{u_{max}d_h} \tag{8}$$

Here u_i is the velocity component in the x_i direction, p the pressure, T the temperature, w the humidity, D the mass diffusivity, ρ the density, μ the dynamic viscosity, k the thermal conductivity, and C_p the specific heat. The flow is incompressible, but the density variation due to the temperature or the humidity is taken into consideration as was done in O'Connell [13].

$$\rho = \frac{p}{RT\sum_k(m_k/M_k)} \tag{9}$$

where m_k and M_k are the mass fraction and the molecular weight of species k, respectively. The saturated humidity, which is identified by the saturation line in the psychrometric chart, may be described by the following formula:

$$w_s(T_w) = \frac{p_s(T_w)}{\rho R_v T_w} \tag{10}$$

The fin configuration of the heat exchanger is taken to be that of the experimental settings [14] and the boundary conditions for air and water temperature, and humidity are also matched to those of the experiment. The schematic of the heat exchanger under consideration together with its dimensions is shown in Fig. 2. It may be assumed periodic in the transverse direction and it suffices to consider only half of a unit module indicated in the figure. The