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A distributed enthalpy model for the calculation of humid air condensers with long narrow channels using a modified Merkel method

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ABSTRACT

A distributed enthalpy model for flow condensation of humid air in long narrow channels is presented in this paper. The model is based on the Merkel method which treats the simultaneous heat and mass transfer between humid air and wet surfaces, with the condensate interfacial waviness effect modified by several correction methods. Experimental data obtained from condensation tests at constant wall temperature conditions are used to validate the model. Results show that the presented model is capable of predicting the distributions of fluid temperature, specific humidity and condensate flow rate in the channel with adequate accuracy. This distributed enthalpy model is simple for condensation calculations in condensers such as plate-fin heat exchangers.

1. Introduction

For aeronautical engineering or other fields that call for compactness, plate-fin heat exchangers (PFHE) are widely used, such as the condensers in aircraft environmental control systems, and the intercoolers in gas turbine systems. The hot fluids in these PFHEs are usually humid air, which are cooled and dehumidified by cold agents. Although conventional lump-methods [1] can be easily used for the design of these PFHEs, they are incapable of calculating the internal distribution of parameters, such as the fluid temperatures, the vapor mass fraction or the condensate flow rate. Alternatively, there are many theoretical models published in open literature that have been developed for vapor condensation in the presence of non-condensable gases, and accomplish the drawback of the lump-methods. These models are capable of calculating humid air condensation in tubes.

Some of the theoretical models are based on the conservation equations of energy, mass and momentum as well as the diffusion equation either in the boundary layer of the vapor-gas mixture [2–4] or in the whole flow field [5–10]. The solution domain is divided into structured grid of control volumes and the governing equations are then solved using finite volume or finite difference methods. These models are rigorous in theory and are capable of providing accurate solutions. However, they are developed for heat and mass transfer on flat plate or in smooth tubes, and are difficult to be applied to the condensation of vapor-gas mixtures in channels with unsmooth heat transfer surfaces (typical in compact heat exchangers as PFHEs) due to the disturbed boundary layer and to the difficulty in discretization of the solution domain. Other researchers established their models using the serial-thermal-resistance method which is based on the energy balance between the vapor-gas mixture and the cooling surface [11–15]. The thermal resistance at the mixture side is treated as a combination of the sensible and latent thermal resistances, with a latent heat transfer coefficient defined in the same form as that of the sensible heat transfer. Empirical correlations are used in these models to determine the heat and mass transfer coefficients in the mixture, while the heat transfer in the condensate film is addressed using other published methods. These serial-thermal-resistance models are applicable to flow condensation in the channels in PFHEs as long as the methods determining the individual resistances in the energy balance are adaptable to be used accordingly. However, the calculation of the latent heat transfer coefficient introduces complexity to the application of this method, which can be further simplified for humid air condensation.

Compared to the above theoretical models, the Colburn and Hougen type method [16–18] is by far the most simple model to be used for the calculation of vapor-gas flow condensation. It is based on the energy conservation at the interface between the vapor-gas mixture and the condensing liquid, the sensible and latent heat transfer are calculated separately using empirical correlations and the heat and mass transfer analogy. In practical calculation, the entire condenser is divided into several sections, and the

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Nomenclature

A	effective heat transfer area, m^2, $A = \pi dl$
A_c	the minimum cross section area, m^2
c_p	specific heat capacity, J/kg·K
d	specific humidity, kg/kg dry air; or inner diameter of the test tube, m
D	mass diffusivity, m^2/s
D_h	hydraulic diameter, m
f	Fanning friction factor, $f = \Delta p/(2LG^2/\rho D_h)$
f_r	interfacial friction factor
f_g	friction factor of gas flowing alone
f_s	smooth surface friction factor
G	mass velocity, kg/m^2·s, $G = W/A_c$
h	heat transfer coefficient, W/m^2·K
h_{md}	mass transfer coefficient, kg/m^2·s
i	enthalpy, J/kg
j	Colburn j factor, $j = Nu/(Re{\cdot}Pr^{1/3})$
k	index
l	grid length, m
L	length of the channel, m
m	exponent in Eq. (15)
M	molar weight, kg/mol
n	number of grids
Nu	Nusselt number, $Nu = hD_h/\lambda$
p	pressure, Pa
p_v	vapor partial pressure, Pa
Pr	Prandtl number, $Pr = \mu c_p/\lambda$
q	heat flux, W/m^2
r	latent heat of vaporization of water, J/kg
r_0	latent heat of vaporization of water at 0 °C, r_0 = 2500.9 kJ/kg
Re	Reynolds number, $Re = GD_h/\mu$
t	temperature, °C
u	velocity, m/s
W	mass flow rate, kg/s
Y	mole fraction

Greek symbols

α	thermal diffusivity, m^2/s
δ	liquid film thickness, um
λ	thermal conductivity, W/m·K
μ	viscosity, Pa·s
ξ	correction coefficient
ρ	density, kg/m^3
σ	accuracy requirement
τ	interfacial shear force, kg/m·s^2
ϕ	dimensionless constant defined in Eqs. (24) and (25)
Δ	difference

Subscripts

a	dry air
b	bulk flow
cal	calculated value
$cond$	condensate
dew	dew point
ex	exit of channel
exp	experimental value
f	liquid film
g	gas phase flowing alone, smooth pipe value
i	interface
in	inlet of channel
l	latent
m	air-water vapor mixture
$mean$	average value in the channel
new	current iteration
old	previous iteration
r	roughness, interfacial
s	sensible
v	vapor
w	wall
0	initial state of humid air
1	dry air
2	water vapor

Colburn-Hougen method is performed iteratively through all the sections. The Colburn-Hougen method is simple and applicable to heat exchangers such as plate-fin heat exchanger, but the calculation of condensation rate and the interfacial temperature also introduce complexity to the model, and can be further simplified.

In engineering applications, the long narrow channel in the PFHEs may be purely dry, partially dry and partially wet, or completely wet. The heat transfer in the dry part is sensible convection and can be simply addressed by common treatments. While for the wet part, heat and mass transfer occur simultaneously on the surface of the condensate film, the solution of this problem had been studied by Merkel as early as in 1925 for the heat and mass transfer process in cooling towers [19], and can also be seen in some other monographs [20,21]. Till now, the Merkel method is mainly applied to the calculation of heat and mass transfer on the surface of water droplets in cooling towers or on the surface of the condensate film adhered to the outer surface of cooling coils, the application of this method to the condensation of humid air in long narrow channels is not seen in open literature.

The flow condensation in long narrow channels features a continuous wavy liquid film flowing along the internal wall of the channel. The waviness caused by the shear force of the gas phase behaves like rough interface, which makes the heat and mass transfer coefficients different from that on smooth surface. This effect has been considered in many theoretical models mentioned previously [13,15,18], but has not been combined with the Merkel method to address the condensation problem in long narrow channels. Regarding the interfacial waviness, Norris [22] suggested using the ratio of rough surface friction factor to the smooth surface value to account for the roughness effect on heat transfer coefficient, Collier and Thome [23] suggested using a similar method proposed by Wallis [24,25]. In both methods, the essential point is the calculation of the friction factor on the wavy surface of the liquid film. A number of studies have been published on this subject [24–29]. Compared to the methods of Henstock and Hanratty [26], and Asali et al. [27], the Wallis' method [24] was proved to be able to better predict the interfacial friction factor except a little overestimation when the film thickness is very thin [28]. Fore and co-workers also found that a decrease of 0.0015 to the relative film thickness in Wallis' equation would make it well fit their experimental data. The experimental work of Mascarenhas et al. [29] using nitrogen-water mixture validated the correlations of Wallis [24] and Fore et al. [28], the results revealed that both two correlations can give reasonable predictions for the interfacial friction factor but overestimate at higher gas phase Reynolds number (Re_g = 5000–11,000). Based on the experimental data, Mascarenhas et al. proposed their own correlation, which includes the wave height and wavelength of the wavy film surface in the expression. Although the correlation of Mascarenhas et al. can better describe their experimental data, the wave height and wavelength are two