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A computational study of mixed convective heat and mass transfer from a shrouded vertical non-isothermal fin array during dehumidification process

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A B S T R A C T

A computational study is made on simultaneous heat and mass transfer from a shrouded vertical non-isothermal finite thickness fin array representing dehumidification process undergoing mixed convection. Governing equations involve number of parameters, namely, dimensionless fin spacing ($0.1 \leqslant S^* \leqslant 0.5$), dimensionless clearance ($0 \leqslant t_c^* \leqslant 0.2$), non-dimensional fin conductance parameter ($2830.18 \leqslant \Omega \leqslant 5660.37$), thermal Grashof number ($1.04 \times 10^5 \leqslant Gr_t \leqslant 4.82 \times 10^5$), mass Grashof number ($2.9 \times 10^4 \leqslant Gr_m \leqslant 1.36 \times 10^5$), and dimensionless inlet mixed convection velocities ($1123 \leqslant W_{in,mix} \leqslant 2879$). Induced velocity is decoupled from the mixed convection velocity and correlated with the governing parameters. Fin temperature distribution shows significant variation along the fin height as well as along the axial direction, especially near the entrance. Local thermal Nusselt number and local condensing Nusselt number show monotonic decreasing trend along the axial direction. Finally induced velocity, pressure drop, overall thermal Nusselt number, and overall condensing Nusselt number is correlated with the governing parameters.

1. Introduction

The process of dehumidification/humidification finds numerous applications in refrigeration and air-conditioning, chemical process industries, cryogenic, thermal power plant, etc. Dehumidification can be achieved by elevating a surface to a temperature lower than the dew point temperature of the surrounding. On the other hand, humidification may be accomplished by passing hot gas over a liquid film maintained at a temperature lower than the hot gas being passed, provided gas is under saturated with the vapor of liquid undergoing evaporation. The aforesaid processes involve simultaneous heat and mass transfer, which make the topic very complicated, but is worth to study. Therefore, better understanding of the dehumidification/humidification process is a topic of interest by many researchers. It is a common practice to use extended surface to augment heat and mass transport, since extended surfaces provide supplementary surface area. Thus, addition of extended surface in the process of dehumidification will provide a resort to the efficient dehumidification.

McQuiston [1] analytically examined moisture condensation on a fin with one dimensional heat conduction model by assuming of a linear relationship between specific humidity and dry bulb temperature and obtained a closed form expression of fin efficiency in terms of geometric and thermo-physical parameters of fin. Mass transfer coefficient is estimated from Lewis relation (i.e., $h_m = h/C_p$). Results indicate that fin efficiency decreases with the relative humidity. Following McQuiston [1], Elmahdy and Biggs [2] analytically studied moisture condensation on annular disc fin assigning linear variation of specific humidity with temperature assuming one-dimensional fin conduction model. No closed form solution is obtained. Therefore, simple one dimensional equation is solved numerically, which provides a data base of fin efficiency. It is found that fin efficiency decreases with relative humidity corroborating the work of McQuiston [1]. Later Wu and Bong [3], examined the performance of rectangular fin undergoing simultaneous heat and mass transfer assigning mass transfer to Chilton-Colburn analogy. Model accommodates both wet fin and partially wet fin. However, following McQuiston's model, Wu and Bong [3] assumed that driving potential of mass transfer (i.e., specific humidity) is related linearly with the dry bulb temperature. Noticeable difference in fin efficiency is observed between McQuiston model with that obtained from Wu and Bong [3]. Result of fin efficiency agrees close to Threlkeld [4] and indicates fin efficiency variation with relative humidity relatively lower than that obtained from McQuiston [1]. Recently, Sharqawy and Zubair

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Nomenclature

A_c	cross-sectional area of fin geometry, $S(H+t_c)+t_f\times t_c$
$c_{p,ma}$	constant pressure specific heat of moist air (J/kg K)
$c_{p,w}$	constant pressure specific heat of water vapor (J/kg K)
D	mass diffusivity (m^2/s)
g	gravitational acceleration (m/s^2)
H	fin height (m)
Gr_t	thermal Grashof number, $g\beta_t(T_0-T_w)H^3/\nu^2$ (dimensionless)
Gr_m	mass Grashof number, $g\beta_m(\omega_0-\omega_w)H^3/\nu^2$ (dimensionless)
h	heat transfer coefficient (W/m^2 K)
h_{fg}	enthalpy of condensation (J/kg)
h_m	mass transfer coefficient (kg/m^2 s)
Ja	Jacob number, $c_{p,w}(T_0-T_w)/h_{fg}$ (dimensionless)
k_{fin}	fin thermal conductivity (W/m K)
k	fluid thermal conductivity (W/m K)
L	fin length (m)
L^*	dimensionless fin length (L/H)
M_{da}	relative molecular mass of dry air (=28.9645 kg/kmol)
M_w	relative molecular mass of water vapor (=18.01527 kg/kmol)
Nu	Nusselt number (dimensionless)
p	local pressure defect, p_s-p_0 (Pa)
p_s	static pressure (Pa)
p_0	ambient pressure, $\int_0^Z \rho_0 g dz$ (Pa)
P^*	dimensionless axial pressure defect, $pH^2/\rho_0\nu^2$
Pr	Prandtl number, ν/α (dimensionless)
R_b	Buoyancy ratio, Gr_m/Gr_t (dimensionless)
r_c	Temperature ratio, $T_0/\Delta T$ (dimensionless)
Re	Reynolds number, $W_{in,force}H/\nu$ (dimensionless)
S	fin spacing (m)
S^*	dimensionless fin spacing, S/H
Sc	Schimdt Number, ν/D (dimensionless)
T	temperature (K)
t_c	fin tip to shroud clearance (m)
t_c^*	dimensionless tip clearance, t_c/H
t_f	fin thickness (m)
t_f^*, t_{f1}^*	dimensionless fin thickness and half fin thickness respectively, t_f/H, $0.5t_f/H$
u, v, w	velocity components in x, y and z directions (m/s)
U, V, W	dimensionless velocities in X, Y and Z directions, uH/ν, vH/ν, wH/ν
x, y, z	cross stream and axial coordinates (m)
X, Y, Z	dimensionless cross stream and axial coordinates, $x/H, y/H, z/H$

Greeks

α	thermal diffusivity (m^2/s)
β_m	solutal volumetric expansion coefficient, $-(1/\rho_0)(\partial\rho_{ma}/\partial\omega)\approx M_{da}/M_w-1$ (dimensionless)
β_t	thermal volumetric expansion coefficient, $-(1/\rho_0)(\partial\rho_{ma}/\partial T)=1/T_0$ (1/K)
ΔT	scaling temperature difference, (T_0-T_w) (K)
$\Delta\omega$	scaling mass fraction difference, $(\omega_w-\omega_0)$ (dimensionless)
ρ	density/partial density (kg/m^3)
θ	dimensionless temperature, $(T_0-T)/(T_0-T_w)$
ν	momentum diffusivity (m^2/s)
χ	scaled mass fraction difference, $(\omega_0-\omega)/(\omega_0-\omega_w)$
ω	mass fraction of water vapor, ρ_w/ρ_{ma}
Ω	non-dimensional fin conductivity, k_{fin}/k

Subscripts

b	bulk
c	condensate
d	dry
da	dry air
f	fin
ma	moist air
0	ambient/reference
s	shroud

[5] visited the problem of straight fin with rectangular profile, triangular profile, convex parabolic profile, and concave parabolic profile assuming mass transfer coefficient to obey Chilton-Colburn analogy following Wu and Bong [3]. In all cases, closed form solutions are obtained. All the aforesaid articles consider linear relationship between specific humidity and temperature. Recalling psychrometric chart, it may easily be identified that specific humidity is hardly linear. Therefore, results in the articles [1–5] cannot be used for the large temperature range. However, in small temperature range, this result may be considered useful. To overcome this difficulty, Kundu [6] recently examined the performance of wet fin with a polynomial relationship between humidity ratio and temperature. However, Chilton-Colburn analogy is used to relate heat and mass transfer coefficient. Introduction of non-linear relationship of specific humidity with temperature, fin conduction equation becomes non-linear, which is then solved Adomian decomposition method. Results of Kundu [6] find deviation from other authors [1–5], but it is more close to Sharqawy and Zubair [5]. Xu et al. [7] reexamined the McQuiston model including the effect of condensate film moving on fin surface. Results indicate significant deviation whenever condensate rate is high. Kilic and Onat [8] observe optimum thickness of fin by an analytical means, in which mass transfer coefficient is assumed constant, while specific humidity responsible mass transfer on the fin surface is evaluated from relation. Toner et al. [9] made a comparison of rectangular fin with triangular fin under condensing condition. It uncovers that triangular fin provides better heat transport characteristics presumably due to increased surface area available in triangular fin.

Analytical studies mentioned above all assumed constant heat transfer coefficient. Thus, it does not talk about fluid flow condition (i.e., laminar or turbulent flow) over fin surface. Further, it is found in literature [10] that fin heat transfer coefficient varies significantly along the fin height. Therefore the assumption of constant heat transfer coefficient is questionable. Karvinen et al. [11] made a balance between conductive heat transfer through fin with convective heat flux over the fin surface, where convective heat transfer is estimated by a relation provided in [12] for a isothermal surface. Thus, fin conduction equation becomes non-linear, which is solved by numerical technique. Numerical results predict relatively better than closed form analytical model, when results are compared with the experiment. However, results differ considerably with the experimental finding whenever relative humidity is high. Authors [11] also developed analytical model including radiation effect assuming constant heat and mass transfer coefficient. But the analytical result deviates more from experimental results. Lin et al. [13] reported experimental finding of wet rectangular plate fin. Dropwise condensation is noted in the flow visualization. However, dropwise condensation is prominent near the boundary of dry-wet region of fin. Away from the boundary of dry-wet